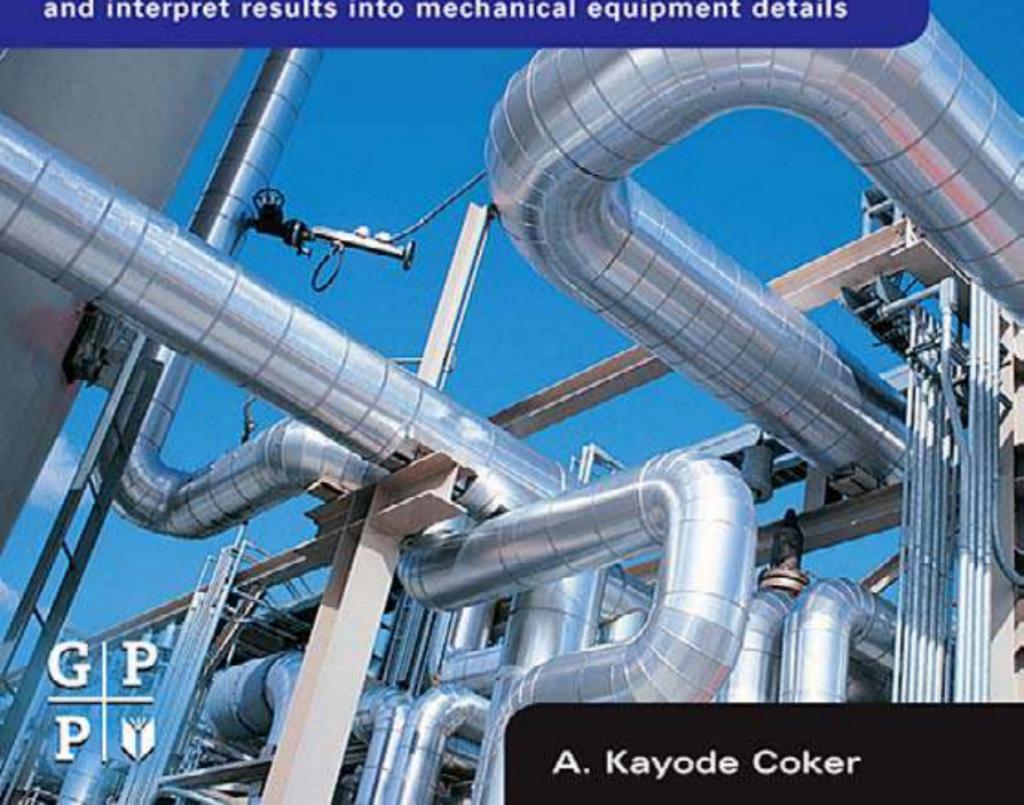


VOLUME 1 | FOURTH EDITION

# Ludwig's Applied Process Design for Chemical and Petrochemical Plants

Emphasizes how to apply techniques of process design  
and interpret results into mechanical equipment details



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P V

A. Kayode Coker

# **Ludwig's Applied Process Design for Chemical and Petrochemical Plants**

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# **Ludwig's Applied Process Design for Chemical and Petrochemical Plants**

*Volume 1*

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Fourth Edition

**A. Kayode Coker**



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Dedication and Love

In Memory of Ernest E. Ludwig

(A Great Chemical Engineer)

and

In Loving Memory of my dear Mother

Modupe Ajibike Coker

To my wife, Victoria

Love and thanks

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## Preface to the Fourth Edition

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This complete revision of *Applied Process Design for Chemical and Petrochemical Plants*, Volume 1, builds upon the late Ernest E. Ludwig's classic text to further enhance its use as a chemical engineering process design manual of methods and proven fundamentals with supplemental mechanical and related data, nomographs, and charts (some in the expanded appendices). Significantly expanded and thoroughly updated, fourth edition contains new topics that will assist the engineer in examining and analyzing problems and finding design methods and mechanical specifications to secure the proper mechanical hardware to accomplish a process objective.

This latest edition includes improved techniques and fundamental design methodologies to guide the engineer in designing process equipment and applying chemical processes to the properly detailed hardware (equipment), because without properly sized and internally detailed hardware, the process will not achieve its unique objective. The various derived and proven equations have been employed in actual plant equipment design, and some of the most reasonable available to inexperienced and experienced engineers (excluding proprietary data and design methods).

This book further provides both fundamental theories where applicable and directs application of these theories to applied equations essential in the design effort. A conscious effort has been made to offer guidelines of sound engineering judgment, decisions, and selections with available codes (e.g. ASME, API, ANSI, TEMA, ASTM, NFPA, BS) and specifications, as some of these are illustrated as problems in the text. This approach at presentation of design information serves well for troubleshooting plant operation problems and equipment/system performance analysis.

This fourth edition presents many developed and executable computer programs and Excel spreadsheet programs, which are readily available to assist the engineer in his or her design problems. Additionally, there are nearly 50 process data sheets in either Excel spreadsheet format or hard copies that can be readily accessed. This book can be used as a classroom text for senior and graduate level chemical plant design courses at the university level. For the first time, the appendices provide with examples of various numerical methods that prove useful for undergraduate and graduate students.

The text material assumes that the reader is at least an undergraduate engineer with a sound knowledge of the fundamentals of the profession. The book will provide the reader with design techniques to actually design as well as mechanical detail for construction. The aim of the process engineer is to ensure that results of his or her process calculations for equipment are specified in terms of something that can be economically constructed or selected from the special designs of manufacturers. This edition follows the format of previous editions in emphasizing the mechanical codes

and some requirements that can be important in the specifications as well as the actual specific design details. An added feature is professional ethics incorporating codes of conduct from the Institution of Chemical Engineers and the American Institute of Chemical Engineers.

Economic evaluations are essential in determining the viability of any project, so the chemical or process engineer must be able to ascertain the economic impact of a new or existing chemical process and judge whether the project will provide a return on investment for his company. The techniques of economic analysis are used to assess the profitability of projects involving both capital expenditures and yearly operating costs.

Because the engineer should have the ready access to essential physical property data of compounds for design calculations of process equipment, this revised volume includes more than 20 physical property data for liquids and gases in Excel spreadsheet format and hardcopy with example problems. Such data are usually found in specialized texts or simulation design packages, or obtained by conducting experiments to measure the properties of individual substances or of mixtures, which may exhibit nonideal behavior, but this is often time-consuming and expensive. Thermodynamic data of this type are required in most calculations such as sizing vessels, process pipe lines, separation of multicomponents, gas absorption and chemical reactor design, and so are now included in this volume.

The techniques of applied chemical plant process design continue to improve as the science of chemical engineering develops new and better interpretations of the fundamentals of chemistry, physics, metallurgical, mechanical, and polymer/plastic science. Accordingly, this fourth edition presents additional reliable design methods based on sound experimental data, proven techniques developed by companies and individuals and groups considered competent in their subjects and who are supported by pertinent data. In breaking with tradition of previous editions, the fourth edition has incorporated the use of S.I. units in some of its design calculations. The text also provides useful conversion table in electronic format to aid the engineer when required. Every chapter has been expanded and updated with new materials. The appendix contains many process flow diagrams and piping and instrumentation diagrams (P & IDs) of some chemical processes that should assist the designer for comparison.

For further information, and for supplementary materials, please visit <http://books.elsevier.com/companions/9780750677660>. In addition, all figures and diagrams from this text will also be available online, as well as additional material.

A. Kayode Coker, C.Eng

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## Preface to the Third Edition

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This volume of *Applied Process Design* is intended to be a chemical engineering process design manual of methods and proven fundamentals with supplemental mechanical and related data and charts (some in the expanded Appendix). It will assist the engineer in examining and analyzing a problem and finding a design method and mechanical specifications to secure the proper mechanical hardware to accomplish a particular process objective. An expanded chapter on safety requirements for chemical plants and equipment design and application stresses the applicable Codes, design methods, and the sources of important new data.

This manual is not intended to be a handbook filled with equations and various data with no explanation of application. Rather, it is a guide for the engineer in applying chemical processes to the properly detailed hardware (equipment), because without properly sized and internally detailed hardware, the process very likely will not accomplish its unique objective. This book does not develop or derive theoretical equations; instead, it provides direct application of sound theory to applied equations useful in the immediate design effort. Most of the recommended equations have been used in actual plant equipment design and are considered to be some of the most reasonable available (excluding proprietary data and design methods) that can be handled by both the inexperienced as well as the experienced engineer. A conscious effort has been made to offer guidelines of judgment, decisions, and selections, and some of this will also be found in the illustrative problems. My experience has shown that this approach at presentation of design information serves well for troubleshooting plant operation problems and equipment/systems performance analysis. This book also can serve as a classroom text for senior and graduate level chemical plant design courses at the university level.

The text material assumes that the reader is an under-graduate engineer with one or two years of engineering fundamentals or a graduate engineer with a sound knowledge of the fundamentals of the profession. This book will provide the reader with design techniques to actually design as well as mechanically detail and specify. It is the author's philosophy that the process engineer has not adequately performed his or her function unless the results of a process calculation for equipment are specified in terms of something that can be economically built or selected from the special designs of manufacturers and can by visual or mental techniques be *mechanically* interpreted to actually perform the process function for which it was designed. Considerable emphasis in this book is placed on the mechanical Codes and some of the requirements that can be so important in the specifications as well as the actual specific design details. Many of the mechanical and metallurgical specifics that are important to good design practice are not usually found in standard mechanical engineering texts.

The chapters are developed by *design function* and not in accordance with previously suggested standards for unit operations. In fact, some of the chapters use the same principles, but require different interpretations that take into account the *process* and the *function* the equipment performs in the process.

Because of the magnitude of the task of preparing the material for this new edition in proper detail, it has been necessary to

omit several important topics that were covered in the previous edition. Topics such as corrosion and metallurgy, cost estimating, and economics are now left to the more specialized works of several fine authors. The topic of static electricity, however, is treated in the chapter on process safety, and the topic of mechanical drivers, which includes electric motors, is covered in a separate chapter because many specific items of process equipment require some type of electrical or mechanical driver. Even though some topics cannot be covered here, the author hopes that the designer will find design techniques adaptable to 75 percent to 85+ percent of required applications and problems.

The techniques of applied chemical plant process design continue to improve as the science of chemical engineering develops new and better interpretations of the fundamentals of chemistry, physics, metallurgical, mechanical, and polymer/plastic sciences. Accordingly, this third edition presents additional reliable design methods based on proven techniques developed by individuals and groups considered competent in their subjects and who are supported by pertinent data. Since the first and second editions, much progress has been made in standardizing (which implies a certain amount of improvement) the hardware components that are used in designing process equipment. Much of the important and basic standardization has been incorporated in this latest edition. Every chapter has been expanded and updated with new material.

All of the chapters have been carefully reviewed and older (not necessarily obsolete) material removed and replaced by newer design techniques. It is important to appreciate that not all of the material has been replaced because much of the so-called "older" material is still the best there is today, and still yields good designs. Additional charts and tables have been included to aid in the design methods or explaining the design techniques.

The author is indebted to the many industrial firms that have so generously made available certain valuable design data and information. Thus, credit is acknowledged at the appropriate locations in the text, except for the few cases where a specific request was made to omit this credit.

The author was encouraged to undertake this work by Dr. James Villbrandt and the late Dr. W. A. Cunningham and Dr. John J. McKetta. The latter two as well as the late Dr. K. A. Kobe offered many suggestions to help establish the usefulness of the material to the broadest group of engineers and as a teaching text.

In addition, the author is deeply appreciative of the courtesy of The Dow Chemical Co. for the use of certain noncredited materials and their release for publication. In this regard, particular thanks is given to the late N. D. Griswold and Mr. J. E. Ross. The valuable contribution of associates in checking material and making suggestions is gratefully acknowledged to H. F. Hasenbeck, L. T. McBeth, E. R. Ketchum, J. D. Hajek, W. J. Evers, and D. A. Gibson. The courtesy of the Rexall Chemical Co. to encourage completion of the work is also gratefully appreciated.

Ernest E. Ludwig, P.E.

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## Foreword

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Since the publication of the previous edition of Volume 1 in 1995 there have been significant developments in process design as applied to chemical and petrochemical plant.

All areas of safety are of importance because of ethical, legal and loss prevention constraints. Lower levels of fugitive emissions are now tolerable and health hazard control is a major concern. Hence an increased proportion of design effort now tends to be spent on safety studies.

The competitive environment in which plants operate necessitates that equipment will perform reliably to specification. But no design can be developed in isolation from economic factors. Some degree of standardization of equipment can result in both capital cost and operative cost savings. When the accuracy of design procedures can be relied upon, "lean design" which avoids unjustified allowances for contingencies can be advantageous. However experience has also shown that in some cases "over design" may be beneficial in allowing extra capacity to be extracted economically.

Probably the major changes in design procedures however have been those made possible by the increased application, and sophistication of computer programs.

This volume has therefore been extensively revised and contains a new chapter covering the physical properties of liquids and gases, with tables of properties as an appendix. Cost estimation and economic evaluation procedures are dealt with in a new

chapter. Hazard identification and analysis, and additional topics in design for safety including concepts in the introduction of inherent safety, are now covered. It is also timely to include material on engineering ethics.

Design engineers will find the inclusion of numerous computer programs of particular value. These are supplemented by appendices containing mathematical functions, analytical and numerical analysis with examples of the use of Fortran and Excel programs to solve selected mathematical problems in chemical process engineering design.

It would be impracticable to deal in one volume with all the techniques and design procedures which engineers now rely upon. As in previous editions, specific topics e. g. metallurgical considerations, associated fire protection systems are excluded. But with the fourth up-date this volume serves as a guide to a majority of design techniques. Moreover it is extensively referenced to other sources of information and data up to 2006.

Easier access to information and the revolutionary developments in computing have proved of immense benefit to designers. But applied process design remains a complex and demanding task. This volume will be of considerable assistance to all those involved in it.

Dr. C. J. Mumford

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## Acknowledgments

Emulating the work of the late Ernest E. Ludwig was a considerable challenge, as he provided the chemical engineering communities with classic works that have withstood the test of time. I hope that this volume at least measures up to his works. Chemical engineering even in the petroleum and petrochemical industries is always evolving with new developments correlations, techniques, data, and nomographs. Examples of these new techniques are in fluid flow (Chapter 4), involving the works of Professor Ron Darby (3-K Method for determining loss coefficient for pipe fittings and valves) and Mr. Trey Walters for compressible fluid flow. Many of these new works and techniques are found in various chapters of the text. Chapter 9 provides extensive work on two-phase runaway reactions pioneered by Dr. Hans Fauske.

I wish to express my profound gratitude to the following for giving their time in proofreading various sections and making valuable comments and suggestions; to many others who have provided many new materials in enhancing and improving this latest edition of *Applied Process Design*. To Professor Ron Darby for his critical review and suggestions on Fluid flow chapter. His expertise and suggestions are greatly appreciated. I am grateful to Mr Lee Partin for reviewing and providing valuable comments on Chapter 1, Dr. C. J. Mumford for his encouragement, comments, and suggestions on Chapter 2, to Mr. Trey Walters for his valuable comments and suggestions on Chapter 4, to Dr. Hans Fauske for his comments and suggestions on Chapter 9, to Mr. Mohammad El-Doma for his time and effort in providing valuable materials, which have greatly enhanced the various chapters, Mr. Joseph Rivera

for his dedication, professionalism, and effort in providing many detailed drawings in the text. Joseph, well done. To Mr. Ahmed Mutawa for developing the conversion table software for the book. Thank you, Ahmed.

Sincere gratitude to many institutions and companies that have provided various materials, which have been included in the text. In particular, Saudi Aramco Shell Refinery (SASREF); Crane Co.; Fauske & Associates, LLC.; Armfield, U.K.; Chemineer, Pfaudler-Balfour Inc.; Krebs Engineers; Envision Systems, Inc.; Trident Computers Resources; Institution of Chemical Engineers, U.K.; American Institute of Chemical Engineers; Hydrocarbon Processing; Chemical Engineering. I am indebted to the many industrial firms that have so generously made available certain valuable design data and information. Thus, credit is acknowledged at the appropriate locations in the text, except for the cases where a specific request was made to omit this credit.

I was encouraged to undertake this project by Mr. Phil Carmical, formerly of Elsevier. Sincere thanks to Phil for first proposing this project and his guidance during the early part of the project; to Mr. Tim Calk, former editor of Gulf Publishing Company, for his encouragement and guidance; to Ms. Priyaa Menon of Integra Software Services for her guidance and suggestions during the production of the text. To production and editorial staff of Elsevier and in particular to Ms. Andrea Sherman for her editing, support, and understanding, Ms. Melinda Ritchie for her excellent production work of the book and to Mr. Theron Shreve, former editor of Elsevier.

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## Biography

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A. Kayode Coker is Chairman of Chemical & Process Engineering Technology department at Jubail Industrial College in Saudi Arabia. He is both a chartered scientist and a chartered chemical engineer for more than 15 years. He is a fellow of the Institute of Chemical Engineers in the U.K., and a senior member of the American Institute of Chemical Engineers and a member of the American Chemical Society. Prior experience includes process engineering for H & G Engineering in Glasgow, Scotland, Davy Energy and Environmental Ltd., U.K., and Shell Petroleum Development Co. of Nigeria, research and development for Blue Circle Industry in the U.K., and a consultant for A.K.C. Technology, U.K. He

holds a B.Sc. honors degree in chemical engineering, an M.Sc. in process analysis and development, and a Ph.D. in chemical engineering, all from Aston University, Birmingham, U.K., and a Teachers' certificate in Education at University of London. He has directed and conducted short courses for blue chip companies in the U.K. and for SABIC industries in Saudi Arabia. He has published several articles in international journals, and is an author of Fortran Programs for Chemical Process Design, Analysis and Simulation, Gulf Publishing Co., Modeling of Chemical Kinetics and Reactor Design, Butterworth-Heinemann, and a book chapter in Encyclopedia of Chemical Processing and Design, vol. 61., Marcel Dekker.

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## **Disclaimer**

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The material in this book was prepared in good faith and carefully reviewed and edited. The author and publisher, however, cannot be held liable for errors of any sort in these chapters. Furthermore, because the author has no means of checking the reliability of some of the data presented in the public literature, but can only examine them for suitability for the intended purpose herein, this

information cannot be warranted. Also because the author cannot vouch for the experience or technical capability of the user of the information and the suitability of the information for the user's purpose, the use of the contents and the software must be at the best judgment of the user.

# Using the Software and Excel Spreadsheet Programs

Microsoft Excel spreadsheets and a high-level programming language (e.g. Fortran) have been employed in this book. Spreadsheets have improved to the point that they are powerful tools for solving engineering problems. Additionally, the author has employed spreadsheets for Microsoft® Excel because of its widespread availability. Powerful packages such as MathCAD®, Mathematical®, Maple®, TK Solver®, Polymath® and others could be used, but none has the widespread availability as of spreadsheets.

The author has developed various codes in the text using the Fortran language, and some software using the screen-handling format under the DOS environment. The use of the high-level programming is still essential for more advanced topics, which cannot readily be handled by a spreadsheet. The author has provided executable codes for all the Fortran source codes in the text. A readme text is provided for the screen-handling software, which is user-friendly.

## USING FORTRAN PROGRAMS

The executable files can be run from Windows by double-clicking on them from the Explorer. However, some programs may find input data files in the same directory. A text file using any text editor (e.g. notepad) can be used to create a data file. It is recommended to run them from the DOS prompt or from any Fortran compiler. The Abssoft® Fortran compiler is a native 32-bit application designed for Microsoft® Windows 95/98, Windows NT™ and Windows XT. Appendix I shows steps in compiling and running the Abssoft compiler.

The executable codes of the Fortran codes are provided in relevant examples with an executable extension as \*.EXE. The data file has the extension \*.DAT, and the result file has the extension \*.RES. The screen-handling software uses the ANSI.SYS device driver in the CONFIG.SYS file. Each screen-handling software uses a batch file that incorporates the ANSI.SYS device driver. The program can be run from DOS using C:\ prompt from the command line. For example, TWOPHASE.EXE in Chapter 4 is run as follows:

C:\>TWOPHASE.BAT (ENTER)

A title is displayed and the user then presses any key from the PC console to continue. Once the screen is cleared, a list of eight options is displayed as follows:

- |                    |                                    |
|--------------------|------------------------------------|
| <b>1. OPEN</b>     | Open a pre-existing file           |
| <b>2. CREATE</b>   | Create a new data file             |
| <b>3. SAVE</b>     | Save the data file you just edited |
| <b>4. CALCS</b>    | Start calculating                  |
| <b>5. PRT RES</b>  | Print the result file              |
| <b>6. SAVE RES</b> | Save the result file               |
| <b>7. FINISH</b>   | Next problem?                      |
| <b>8. QUIT</b>     | Exit to DOS                        |

TWOPHASE is user-friendly and the procedure in running the program is as follows:

1. Create a data file (option 2). Assign the value to any fitting type that is not included in the calculation. Press a return key or a downward arrow key to continue.
2. Calculate the input data (option 4).
3. Print the results onto a printer (option 5). The results are also viewed on the screen.
4. Quit the program by typing N (option 8) to return to DOS (Disk Operating System).
5. Press FINISH (option 7) before opening or creating a data file.

Options 1, 3, 6 and 7 can be further used after the data file has been created.

## USING MICROSOFT EXCEL SPREADSHEET

Start the Excel spreadsheet by double-clicking on the icon. To open a file name, click on “FILE” on the menu bar, then choose “Open . . .”. To open an existing worksheet or file name from the folder, double-click on it. This will open the document (and will start the Excel spreadsheet as well if it was not already running). To print output, you can choose “Printer Setup . . .” to specify a printer, “Page Setup . . .” to specify margins, automatic scaling, etc., and “Print preview . . .” to preview output. To quit the Excel spreadsheet and return to Windows, choose “File . . .”, “Exit”. To exit Windows, repeat the same commands from the File Manager screen.

# Chapter 0

## RULES OF THUMB: SUMMARY

### COMPRESSORS, FANS, BLOWERS AND VACUUM PUMPS

1. *Fans* are used to raise the pressure by about 3% [12 in. (30 cm) water], *blowers* raise to less than 2.75 barg (40 psig), and *compressors* to higher pressures, although the blower range is commonly included in the compressor range.

2. For vacuum pumps use the following:

Reciprocating piston Type	down to 133.3 Pa (1 torr)
Rotary piston type	down to 0.133 Pa (0.001 torr)
Two lobe rotary type	down to 0.0133 Pa (0.0001 torr)
Steam jet ejectors	1 stage down to 13.3 kPa (100 torr) 3 stage down to 133.3 Pa (1 torr) 5 stage down to 6.7 Pa (0.05 torr)

3. A three-stage ejector needs 100 kg steam/kg air to maintain a pressure of 133.3 Pa (1 torr).
4. In-leakage of air to evacuated equipment depends on the absolute pressure (torr) and the volume of the equipment,  $V$  in  $\text{m}^3$  ( $\text{ft}^3$ ), according to  $W = kV^{2/3} \text{kg/h}$  ( $\text{lb/h}$ ), with  $k = 0.98$  (0.2) when  $P > 90$  torr,  $k = 0.39$  (0.08) when  $P$  is between 0.4 and 2.67 kPa (3 and 20 torr), and  $k = 0.12$  (0.025) at  $p$  less than 133.3 Pa (1 torr).
5. Theoretical adiabatic horsepower

$$\text{THP} = \frac{(\text{SCFM}) T_1}{8130a} \left[ \left( \frac{P_2}{P_1} \right)^a - 1 \right]$$

where  $T_1$  is inlet temperature in Rankine,  $R = {}^\circ\text{F} + 460$  and  $a = (k - 1)/k$ ,  $k = C_p/C_v$ . Theoretical reversible adiabatic power =  $mz_1RT_1[(\{P_2/P_1\}^a - 1)]/a$  Where  $T_1$  is inlet temperature,  $R$  = Gas Constant,  $z_1$  = compressibility factor,  $m$  = molar flow rate,  $a = (k - 1)/k$  and  $k = C_p/C_v$ . Values of  $R = 8.314 \text{J/mol K} = 1.987 \text{Btu/lb mol}^\circ\text{R} = 0.7302 \text{atm ft}^3/\text{lb mol}^\circ\text{R}$ .

6. Outlet temperature for reversible adiabatic process

$$T_2 = T_1 \left( \frac{P_2}{P_1} \right)^a$$

7. To compress air from  $37.8^\circ\text{C}$  ( $100^\circ\text{F}$ ),  $k = 1.4$ , compression ratio = 3, theoretical power required = 62 hp/million  $\text{ft}^3/\text{day}$ , outlet temperature  $152.2^\circ\text{C}$  ( $306^\circ\text{F}$ ).
8. Exit temperature should not exceed  $167$ – $204^\circ\text{C}$  ( $350$ – $400^\circ\text{F}$ ); for diatomic gases ( $C_p/C_v = 1.4$ ), this corresponds to a compression ratio of about 4.
9. Compression ratio should be about the same in each stage of a multistage unit, ratio =  $(P_n/P_1)^{1/n}$ , with  $n$  stages.
10. Efficiencies of reciprocating compressors: 65% at compression ratio of 1.5, 75% at 2.0, and 80–85% at 3–6.
11. Efficiencies of large centrifugal compressors,  $2.83$ – $47.2 \text{ m}^3/\text{s}$  ( $6000$ – $100,000 \text{ acfm}$ ) at suction, are 76–78%.
12. Rotary compressors have efficiencies of 70%, except liquid liner type which have 50%.

### CONVEYORS FOR PARTICULATE SOLIDS

1. *Screw conveyors* are suited to transport of even sticky and abrasive solids up inclines of  $20^\circ$  or so. They are limited to distances of 3.81 m (150 ft) or so because of shaft torque strength. A 304.8 mm (12 in.) diameter conveyor can handle  $28.3$ – $84.95 \text{ m}^3/\text{h}$  ( $1000$ – $3000 \text{ ft}^3/\text{h}$ ), at speeds ranging from 40 to 60 rpm.
2. *Belt conveyors* are for high capacity and long distances (a mile or more, but only several hundred feet in a plant), up inclines of  $30^\circ$  maximum. A 609.6-mm (24 in.) wide belt can carry  $84.95 \text{ m}^3/\text{h}$  ( $3000 \text{ ft}^3/\text{h}$ ) at a speed of  $0.508 \text{ m/s}$  (100 ft/min), but speeds up to  $3.048 \text{ m/s}$  (600 ft/min) are suited to some materials. Power consumption is relatively low.
3. *Bucket elevators* are suited to vertical transport of sticky and abrasive materials. With  $508 \times 508$ -mm (20 × 20-in.) buckets, capacity can reach  $28.3 \text{ m}^3/\text{h}$  ( $1000 \text{ ft}^3/\text{h}$ ) at a speed of  $0.508 \text{ m/s}$  (100 ft/min), but speeds up to  $1.524 \text{ m/s}$  (300 ft/min) are used.
4. *Drag-type conveyors* (Redler) are suited to short distances in any direction and are completely enclosed. Units range in size from  $19.4 \times 10^{-4}$  to  $122.6 \times 10^{-4} \text{ m}^2$  (3–19 in.<sup>2</sup>) and may travel from  $0.15 \text{ m/s}$  (30 ft/min) (fly ash) to  $1.27 \text{ m/s}$  (250 ft/min) (grains). Power requirements are high.
5. *Pneumatic conveyors* are for high capacity, short distance (122 m (400 ft)) transport simultaneously from several sources to several destinations. Either vacuum or low pressure  $0.4$ – $0.8$  barg (6–12 psig) is used with a range of air velocities from  $10.7$  to  $36.6 \text{ m/s}$  (35–120 ft/s); depending on the material and pressure and air requirements,  $0.03$ – $0.2 \text{ m}^3/\text{m}^3$  ( $1$ – $7 \text{ ft}^3/\text{ft}^3$ ) of solid is transferred.

### COOLING TOWERS

1. Water in contact with air under adiabatic conditions eventually cools to the wet bulb temperature.
2. In commercial units, 90% of saturation of the air is feasible.
3. Relative cooling tower size is sensitive to the difference between the exit and the wet bulb temperatures:

$\Delta T, {}^\circ\text{F}$	5	15	25
Relative volume	2.4	1.0	0.55

4. Tower fill is of a highly open structure so as to minimize pressure drop, which is in standard practice a maximum of  $497.6 \text{ Pa}$  (2 in. of water).
5. Water circulation rate is  $48.9$ – $195.7 \text{ L/min m}^2$  ( $1$ – $4 \text{ gpm/ft}^2$ ) and air rate is  $6344$ – $8784 \text{ kg/h m}^2$  ( $1300$ – $1800 \text{ lb/h ft}^2$ ) or  $1.52$ – $2.03 \text{ m/s}$  (300–400 ft/min).
6. Chimney-assisted natural draft towers are hyperboloidally shaped because they have greater strength for a given thickness; a tower 76.2 m (250 ft) high has concrete walls 127–152.4 mm (5–6 in.) thick. The enlarged cross section at the top aids in dispersion of exit humid air into the atmosphere.
7. Countercurrent-induced draft towers are the most common in process industries. They are able to cool water within  $2^\circ\text{F}$  of the wet bulb.
8. Evaporation losses are 1% of the circulation for every  $10^\circ\text{F}$  of cooling range. Windage or drift losses of mechanical draft towers are 0.1–0.3%. Blowdown of 2.5–3.0% of the circulation is necessary to prevent excessive salt buildup.

## CRYSTALLIZATION FROM SOLUTION

1. Complete recovery of dissolved solids is obtainable by evaporation, but only to the eutectic composition by chilling. Recovery by melt crystallization also is limited by the eutectic composition.
2. Growth rates and ultimate sizes of crystals are controlled by limiting the extent of supersaturation at any time.
3. The ratio  $S = C/C_{\text{sat}}$  of prevailing concentration to saturation concentration is kept near the range 1.02–1.05.
4. In crystallization by chilling, the temperature of the solution is kept at most 1–2°F below the saturation temperature at the prevailing concentration.
5. Growth rates of crystals under satisfactory conditions are in the range of 0.1–0.8 mm/h. The growth rates are approximately the same in all directions.
6. Growth rates are influenced greatly by the presence of impurities and of certain specific additives, which vary from case to case.

## DISINTEGRATION

1. Percentages of material greater than 50% of the maximum size are about 50% from rolls, 15% from tumbling mills, and 5% from closed-circuit ball mills.
2. Closed-circuit grinding employs external size classification and return of oversize for regrinding. The rules of pneumatic conveying are applied to the design of air classifiers. Closed circuit is most common with ball and roller mills.
3. Jaw crushers take lumps of several feet in diameter to 102 mm (4 in.). Stroke rates are 100–300/min. The average feed is subjected to 8–10 strokes before it becomes small enough to escape. Gyratory crushers are suited to slabby feeds and make a more rounded product.
4. Roll crushers are made either smooth or with teeth. A 610-mm (24-in.) toothed roll can accept lumps of 356 mm (14 in.) diameter. Smooth rolls affect reduction ratios up to about 4. Speeds are 50–90 rpm. Capacity is about 25% of the maximum, corresponding to a continuous ribbon of material passing through the rolls.
5. Hammer mills beat the material until it is small enough to pass through the screen at the bottom of the casing. Reduction ratios of 40 are feasible. Large units operate at 900 rpm, smaller ones up to 16,000 rpm. For fibrous materials the screen is provided with cutting edges.
6. Rod mills are capable of taking feed as large as 50 mm and reducing it to 300 mesh, but normally the product range is 8–65 mesh. Rods are 25–150 mm in diameter. The ratio of rod length to mill diameter is about 1.5. About 45% of the mill volume is occupied by rods. Rotation is at 50–65% of critical.
7. Ball mills are better suited than rod mills to fine grinding. The charge is of equal weights of 1.5-, 2-, and 3-in. balls for the finest grinding. The volume occupied by the balls is 50% of the mill volume. Rotation speed is 70–80% of critical. Ball mills have a length-to-diameter ratio in the range 1–1.5. Tube mills have a ratio of 4–5 and are capable of very fine grinding. Pebble mills have ceramic grinding elements, used when contamination with metal is to be avoided.
8. Roller mills employ cylindrical or tapered surfaces that roll along flatter surfaces and crush nipped particles. Products of 20–200 mesh are made.

## TOWERS

1. Distillation usually is the most economical method of separating liquids, superior to extraction, adsorption, crystallization, or others.

2. For ideal mixtures, relative volatility is the ratio of vapor pressure,  $\alpha_{12} = P_2/P_1$ .
3. Tower operating pressure is most often determined by the temperature of the available condensing medium, 38–50°C (100–120°F) if cooling water, or by the maximum allowable reboiler temperature, 10.34 barg (150 psig) steam, 186°C (366°F) to avoid chemical decomposition/degradation.
4. Sequencing of columns for separating multicomponent mixtures:
  - a. Perform the easiest separation first, that is, the one least demanding of trays and reflux, and leave the most difficult to the last.
  - b. When neither relative volatility nor feed concentration vary widely, remove the components one by one as overhead products.
  - c. When the adjacent ordered components in the feed vary widely in relative volatility, sequence the splits in the order of decreasing volatility.
  - d. And when the concentrations in the feed vary widely but the relative volatilities do not, remove the components in the order of decreasing concentration in the feed.
5. The economically optimum reflux ratio is about 1.2–1.5 times the minimum reflux ratio  $R_m$ .
6. The economically optimum number of theoretical trays is near twice the minimum value  $N_m$ .
7. The minimum number of trays is found with the Fenske–Underwood equation:

$$N_m = \frac{\log \{[x/(1-x)]_{\text{ovhd}} / [x/(1-x)]_{\text{btms}}\}}{\log \alpha}$$

8. Minimum reflux for binary or pseudobinary mixtures is given by the following when separation is essentially complete ( $x_D \cong 1$ ) and  $D/F$  is the ratio of overhead product to feed rate:

$$\text{when feed is at the bubble point } \frac{R_m D}{F} = \frac{1}{\alpha - 1}$$

$$\text{when feed is at the dew point } \frac{(R_m + 1) D}{F} = \frac{\alpha}{\alpha - 1}$$

9. A safety factor of 10% of the number of trays calculated by the best means is advisable.
10. Reflux pumps are made at least 10% oversize.
11. The optimum value of the Kremser–Brown absorption factor  $A = (L/VK)$  is in the range 1.25–2.0.
12. Reflux drums usually are horizontal, with a liquid holdup of 5 min half-full. A takeoff pot for a second liquid phase, such as water in hydrocarbon systems, is sized for a linear velocity of that phase of 0.15 m/s (0.5 ft/s) minimum diameter of 406.4 mm (16 in.).
13. For towers about 914 mm (3 ft) diameter, add 1219 mm (4 ft) at the top for vapor disengagement and 1829 mm (6 ft) at the bottom for liquid level and reboiler return.
14. Limit the tower height to about 53 m (175 ft) maximum because of wind load and foundation considerations. An additional criterion is that  $L/D$  be less than 30 ( $20 < L/D < 30$  often will require special design).

## TRAY TOWERS

1. For reasons of accessibility, tray spacings are made 0.5–0.6 m (20–24 in.).
2. Peak efficiency of trays is at values of the vapor factor  $F_s = \mu (\rho_v)^{0.5}$  in the range of 1.2–1.5 m/s ( $\text{kg}/\text{m}^3\right)^{0.5}$  [ $1\text{--}1.2 \text{ ft/s (lb}/\text{ft}^3\right)^{0.5}$ ]. This range of  $F_s$  establishes the

## XX RULES OF THUMB: SUMMARY

- diameter of tower. Roughly, linear velocities are 0.6 m/s (2 ft/s) at moderate pressures and 1.8 m/s (6 ft/s) in vacuum.
3. Pressure drop per tray is of the order of 747 Pa (3 in. water) or 689.5 Pa (0.1 psi).
  4. Tray efficiencies for distillation of light hydrocarbons and aqueous solutions are 60–90%; for gas absorption and stripping, 10–20%.
  5. Sieve trays have holes of 6–7 mm (0.25–0.50 in.) diameter, hole area being 10% of the active cross section.
  6. Valve trays have holes of 38 mm (1.5 in.) diameter, each provided with a liftable cap, with 130–150 caps per square meter (12–14 caps per square feet) of active cross section. Valve trays are usually cheaper than sieve trays.
  7. Bubblecap trays are used only when liquid level must be maintained at low turndown ratio; they can be designed for lower pressure drop than either sieve or valve trays.
  8. Weir heights are 50 mm (2 in.), weir lengths are about 75% of trays diameter, and liquid rate a maximum of about 1.2 m<sup>3</sup>/min-m of weir (8 gpm/in. of weir); multi-pass arrangements are used at higher liquid rates.

## PACKED TOWERS

1. Structured and random packings are suitable for packed towers less than 0.9 m (3 ft) when low pressure drop is required.
2. Replacing trays with packing allows greater throughput and separation in existing tower shells.
3. For gas rates of 14.2 m<sup>3</sup>/min (500 ft<sup>3</sup>/min), use 25.4-mm (1-in.) packing; for 56.6 m<sup>3</sup>/min (2000 ft<sup>3</sup>/min) or more use 50-mm (2-in.) packing.
4. Ratio of tower diameter/packing diameter should be >15/1.
5. Because of deformability, plastic packing is limited to 3–4 m (10–15 ft) and metal packing to 6.0–7.6 m (20–25 ft) unsupported depth.
6. Liquid distributors are required every 5–10 tower diameters with pall rings and at least every 6.5 m (20 ft) for other types of dumped packing.
7. Number of liquid distributions should be >32–55/m<sup>2</sup> (3–5/ft<sup>2</sup>) in towers greater than 0.9 m (3 ft) diameter and more numerous in smaller columns.
8. Packed towers should operate near 70% of the flooding rate (evaluated from Sherwood and Lobo correlation).
9. Height Equivalent to a Theoretical Stage (HETS) for vapor–liquid contacting is 0.4–0.56 m (1.3–1.8 ft) for 25-mm (1-in.) pall rings and 0.76–0.9 m (2.5–3.0 ft) for 50-mm (2-in.) pall rings.

10. Generalized pressure drops	Design Pressure Drops (cm of H <sub>2</sub> O/m of packing)	Design Pressure Drops (in. of H <sub>2</sub> O/ft of packing)
Absorbers and Regenerators (non-foaming systems)	2.1–3.3	0.25–0.40
Absorbers and Regenerators	0.8–2.1	0.10–0.25
Atmospheric/Pressure Stills and Fractionators	3.3–6.7	0.40–0.80
Vacuum Stills and Fractionators	0.8–3.3	0.10–0.40
Maximum value	8.33	1.0

## DRIVERS AND POWER RECOVERY EQUIPMENT

1. Efficiency is greater for larger machines. Motors, 85–95%; steam turbines, 42–78%; gas engines and turbines, 28–38%.
2. For under 74.6 kW (100 hp), electric motors are used almost exclusively. They are made for up to 14,900 kW (20,000 hp).
3. Induction motors are most popular. Synchronous motors are made for speeds as low as 150 rpm and are thus suited, for example, for low-speed reciprocating compressors, but are not made smaller than 50 hp. A variety of enclosures are available, from weather-proof to explosion-proof.
4. Steam turbines are competitive above 76.6 kW (100 hp). They are speed-controllable. They are frequently used as spares in case of power failure.
5. Combustion engines and turbines are restricted to mobile and remote locations.
6. Gas expanders for power recovery may be justified at capacities of several hundred hp; otherwise any pressure reduction in a process is done with throttling valves.
7. The following useful definitions are given:

$$\text{shaft power} = \frac{\text{theoretical power to pump fluid (liquid or gas)}}{\text{efficiency of pump or compressor, } \varepsilon_{\text{sh}}}$$

$$\text{drive power} = \frac{\text{shaft power}}{\text{efficiency of drive, } \varepsilon_{\text{dr}}}$$

$$\text{Overall efficiency, } \varepsilon_{\text{ov}} = \varepsilon_{\text{sh}} \cdot \varepsilon_{\text{dr}}$$

## DRYING OF SOLIDS

1. Drying times range from a few seconds in spray dryers to 1 h or less in rotary dryers and up to several hours or even several days in tunnel shelf or belt dryers.
2. Continuous tray and belt dryers for granular material of natural size or pelleted to 3–15 mm have drying times in the range of 10–200 min.
3. Rotary cylindrical dryers operate with superficial air velocities of 1.52–3.05 m/s (5–10 ft/s), sometimes up to 10.67 m/s (35 ft/s) when the material is coarse. Residence times are 5–90 min. Holdup of solid is 7–8%. An 85% free cross section is taken for design purposes. In countercurrent flow, the exit gas is 10–20°C above the solid; in parallel flow, the temperature of the exit solid is 100°C. Rotation speeds of about 4 rpm are used, but the product of rpm and diameter in feet is typically between 15 and 25.
4. Drum dryers for pastes and slurries operate with contact times of 3–12 s, and produce flakes 1–3 mm thick with evaporation rates of 15–30 kg/m<sup>2</sup>·h. Diameters are in the range of 1.5–5.0 ft; and rotation rate is 2–10 rpm. The greatest evaporative capacity is of the order of 1360.7 kg/h (3000 lb/h) in commercial units.
5. Pneumatic conveying dryers normally take particles 1–3 mm diameter but up to 10 mm when the moisture is mostly on the surface. Air velocities are 10–30 m/s. Single-pass residence times are 0.5–3.0 s, but with normal recycling the average residence time is brought up to 60 s. Units in use range from 0.2 m in diameter by 1 m long to 0.3 m in diameter by 38 m long. Air requirement is several SCFM per lb of dry product/h.
6. Fluidized bed dryers work best on particles of a few tenths of a mm in diameter, but particles of up to 4 mm in diameter have been processed. Gas velocities of twice the minimum fluidization velocity are a safe prescription. In continuous operation, drying times of 1–2 min are enough, but batch drying of some pharmaceutical products employs drying times of 2–3 h.
7. Spray dryers: Surface moisture is removed in about 5 s, and most drying is completed in less than 60 s. Parallel flow of

air and stock is most common. Atomizing nozzles have openings 3–3.8 mm (0.012–0.15 in.) and operate at pressures of 21–276 bar (300–4000 psi). Atomizing spray wheels rotate at speeds of 20,000 rpm with peripheral speeds of 76.2–183 m/s (250–600 ft/s). With nozzles, the length-to-diameter ratio of the dryer is 4–5; with spray wheels, the ratio is 0.5–1.0. For the final design, the experts say, pilot tests in a unit of 2 m diameter should be made.

## EVAPORATORS

1. Long tube vertical evaporators with either natural or forced circulation are most popular. Tubes are 19–63 mm (0.75–24.8 in.) in diameter and 3.66–9.14 m (12–30 ft) long.
2. In forced circulation, linear velocities in the tubes are in the range of 4.57–6.09 m/s (15–20 ft/s).
3. Elevation of boiling point by dissolved solids results in temperature differences of 3–10°F between solution and saturated vapor.
4. When the boiling point rise is appreciable, the economic number of effects in series with forward feed is 4–6.
5. When the boiling point rise is small, minimum cost is obtained with 8–10 effects in series.
6. In backward feed the more concentrated solution is heated with the highest temperature steam so that heating surface is lessened, but the solution must be pumped between stages.
7. The steam economy of an *N*-stage battery is approximately 0.8 *N*-lb evaporation/lb of outside steam.
8. Interstage steam pressures can be boosted with steam jet compressors of 20–30% efficiency or with mechanical compressors of 70–75% efficiency.

## EXTRACTION, LIQUID-LIQUID

1. The dispersed phase should be the one that has the higher volumetric rate, except in equipment subject to back-mixing where it should be the one with the smaller volumetric rate. It should be the phase that wets the material of construction less well. Since the holdup of continuous phase is greater, that phase should be made up of the less expensive or less hazardous material.
2. There are no known commercial applications of reflux to extraction processes, although the theory is favorable.
3. Mixer-settler arrangements are limited to at most five stages. Mixing is accomplished with rotating impellers or circulating pumps. Settlers are designed on the assumption that droplet sizes are about 150 µm in diameter. In open vessels, residence times of 30–60 min or superficial velocities of 0.15–0.46 m/min (0.5–1.5 ft/min) are provided in settlers. Extraction-stage efficiencies commonly are taken as 80%.
4. Spray towers as tall as 6–12 m (20–40 ft) cannot be depended on to function as more than a single stage.
5. Packed towers are employed when 5–10 stages suffice. Pall rings 25–38 mm (1–1.5 in.) in size are best. Dispersed-phase loadings should not exceed 10.2 m<sup>3</sup>/min-m<sup>2</sup> (25 gal./min-ft<sup>2</sup>). And HETS of 1.5–3.0 m (5–10 ft) may be realized. The dispersed phase must be redistributed every 1.5–2.1 m (5–7 ft). Packed towers are not satisfactory when the surface tension is more than 10 dyne/cm.
6. Sieve tray towers have holes of only 3–8 mm diameter. Velocities through the holes are kept below 0.24 m/s (0.8 ft/s) to avoid formation of small drops. Re-dispersion of either phase at each tray can be designed for. Tray spacings are 152–600 mm (6–24 in.). Tray efficiencies are in the range of 20–30%.

7. Pulsed packed and sieve tray towers may operate at frequencies of 90 cycles/min and amplitudes of 6–25 mm. In large-diameter tower, HETS of about 1 m has been observed. Surface tensions as high as 30–40 dyne/cm have no adverse effect.
8. Reciprocating tray towers can have holes of 150 mm (9/16 in.) diameter, 50–60% open area, stroke length 190 mm (0.75 in.), 100–150 strokes/min, and plate spacing normally 50 mm (2 in.) but in the range of 25.0–150 mm (1–6 in.). In a 760-mm (30-in.) diameter tower, HETS is 500–650 mm (20–25 in.) and throughput is 13.7 m<sup>3</sup>/min-m<sup>2</sup> (2000 gal./h-ft<sup>2</sup>). Power requirements are much less than those of pulsed towers.
9. Rotating disk contractors or other rotary agitated towers realize HETS in the range of 0.1–0.5 m (0.33–1.64 ft). The especially efficient Kuhni with perforated disks of 40% free cross section has HETS of 0.2 m (0.66 ft) and a capacity of 50 m<sup>3</sup>/m<sup>2</sup>-h (164 ft<sup>3</sup>/ft<sup>2</sup>-h).

## FILTRATION

1. Processes are classified by their rate of cake buildup in a laboratory vacuum leaf filter: rapid, 0.1–10.0 cm/s; medium, 0.1–10.0 cm/min; and slow, 0.1–10.0 cm/h.
2. Continuous filtration should not be attempted if 1/8 in. cake thickness cannot be formed in less than 5 min.
3. Rapid filtering is accomplished with belts, top feed drums, or pusher centrifuges.
4. Medium rate filtering is accomplished with vacuum drums or disks or peeler centrifuges.
5. Slow-filtering slurries are handled in pressure filters or sedimenting centrifuges.
6. Clarification with negligible cake buildup is accomplished with cartridges, precoat drums, or sand filters.
7. Laboratory tests are advisable when the filtering surface is expected to be more than a few square meters, when cake washing is critical, when cake drying may be a problem, and when precoating may be needed.
8. For finely ground ores and minerals, rotary drum filtration rates may be 15,000 lb/day-ft<sup>2</sup> at 20 rev/h and 18–25 in. Hg vacuum.
9. Coarse solids and crystals may be filtered at rates of 6000 lb/day-ft<sup>2</sup> at 20 rev/h and 2–6 in. Hg vacuum.

## FLUIDIZATION OF PARTICLES WITH GASES

1. Properties of particles that are conducive to smooth fluidization include rounded or smooth shape, enough toughness to resist attrition, sizes in the range of 50–500 µm diameter, and a spectrum of sizes with ratio of largest to smallest in the range of 10–25.
2. Cracking catalysts are members of a broad class characterized by diameters of 30–150 µm, density of 1.5 g/ml or so, and appreciable expansion of the bed before fluidization sets in, minimum bubbling velocity greater than minimum fluidizing velocity, and rapid disengagement of bubbles.
3. The other extreme of smoothly fluidizing particles are typified by coarse sand and glass beads, both of which have been the subject of much laboratory investigation. Their sizes are in the range of 150–500 µm, densities 1.5–4.0 g/ml, have small bed expansion and about the same magnitudes of minimum bubbling and minimum fluidizing velocities, and they also have rapidly disengaging bubbles.
4. Cohesive particles and large particles of 1 mm or more do not fluidize well and usually are processed in other ways.
5. Rough correlations have been made of minimum fluidization velocity, minimum bubbling velocity, bed expansion, bed level fluctuation, and disengaging height. Experts recommend, however, that any real design be based on pilot-plant work.

6. Practical operations are conducted at two or more multiples of the minimum fluidizing velocity. In reactors, the entrained material is recovered with cyclones and returned to process. In driers, the fine particles dry most quickly so the entrained material need not be recycled.

## HEAT EXCHANGERS

1. For conservative estimate set  $F = 0.9$  for shell and tube exchangers with no phase changes,  $q = UAF\Delta T_{lm}$ . When  $\Delta T$  at exchanger ends differ greatly then check  $F$ , reconfigure if  $F$  is less than 0.85.
2. Take true countercurrent flow in a shell-and-tube exchanger as a basis.
3. Standard tubes are 19.0 mm ( $\frac{3}{4}$  in.) outer diameter (OD), 25.4 mm (1 in.) triangular spacing, 4.9 m (16 ft) long.
  - A shell of 300 mm (1 ft) diameter accommodates  $9.3 \text{ m}^2$  ( $100 \text{ ft}^2$ );
  - 600 mm (2 ft) diameter accommodates  $37.2 \text{ m}^2$  ( $400 \text{ ft}^2$ );
  - 900 mm (3 ft) diameter accommodates  $102 \text{ m}^2$  ( $1100 \text{ ft}^2$ ).
4. Tube side is for corrosive, fouling, scaling, and high-pressure fluids.
5. Shell side is for viscous and condensing fluids.
6. Pressure drops are 0.1 bar (1.5 psi) for boiling and 0.2–0.62 bar (3–9 psi) for other services.
7. Minimum temperature approach is  $10^\circ \text{C}$  ( $20^\circ \text{F}$ ) for fluids and  $5^\circ \text{C}$  ( $10^\circ \text{F}$ ) for refrigerants.
8. Cooling water inlet temperature is  $30^\circ \text{C}$  ( $90^\circ \text{F}$ ), maximum outlet temperature  $49^\circ \text{C}$  ( $120^\circ \text{F}$ ).
9. Heat-transfer coefficients for estimating purposes,  $\text{W/m}^{20} \text{C}$  ( $\text{Btu/h-ft}^{20} \text{F}$ ): water to liquid, 850 (150); condensers, 850 (150); liquid to liquid, 280 (50); liquid to gas, 60 (10); gas to gas, 30 (5); and reboiler 1140 (200). Maximum flux in reboiler is  $31.5 \text{ kW/m}^2$  ( $10,000 \text{ Btu/h-ft}^2$ ). When phase changes occur, use a zoned analysis with appropriate coefficients for each zone.
10. Double-pipe exchanger is competitive at duties requiring  $9.3\text{--}18.6 \text{ m}^2$  ( $100\text{--}200 \text{ ft}^2$ ).
11. Compact (plate and fin) exchangers have  $1150 \text{ m}^2/\text{m}^3$  ( $350 \text{ ft}^2/\text{ft}^3$ ), and about 4 times the heat transfer per cut of shell-and-tube units.
12. Plate and frame exchangers are suited to high sanitation services and are 25–50% cheaper in stainless steel construction than shell-and-tube units.
13. Air coolers: Tubes are 0.75–1.00 in. OD., total finned surface  $15\text{--}20 \text{ ft}^2/\text{ft}^2$  bare surface,  $U = 450\text{--}570 \text{ W/m}^{20} \text{C}$  ( $80\text{--}100 \text{ Btu/hr-ft}^{20}(\text{bare surface}) \text{F}$ ). Minimum approach temperature =  $22^\circ \text{C}$  ( $40^\circ \text{F}$ ). Fan input power =  $1.4\text{--}3.6 \text{ kW}/(\text{MJ/h})$  [ $2\text{--}5 \text{ hp}/(1000 \text{ Btu/h})$ ].
14. Fired heaters: radiant rate,  $37.6 \text{ kW/m}^2$  ( $12,000 \text{ Btu/h-ft}^2$ ), convection rate,  $12.5 \text{ kW/m}^2$  ( $4000 \text{ Btu/h-ft}^2$ ); cold oil tube velocity =  $1.8 \text{ m/s}$  (6 ft/s); approximately equal heat transfer in the two sections; thermal efficiency, 70–75%; flue gas temperature,  $140\text{--}195^\circ \text{C}$  ( $250\text{--}350^\circ \text{F}$ ) above feed inlet; and stack gas temperature,  $345\text{--}510^\circ \text{C}$  ( $650\text{--}950^\circ \text{F}$ ).

## INSULATION

1. Up to  $345^\circ \text{C}$  ( $650^\circ \text{F}$ ), 85% magnesia is used.
2. Up to  $870\text{--}1040^\circ \text{C}$  ( $1600\text{--}1900^\circ \text{F}$ ), a mixture of asbestos and diatomaceous earth is used.
3. Ceramic refractories at higher temperatures.
4. Cryogenic equipment – $130^\circ \text{C}$  ( $-200^\circ \text{F}$ ) employs insulations with fine pores of trapped air, for example, Perlite™.

5. Optimum thickness varies with temperature: 12.7 mm (0.5 in.) at  $95^\circ \text{C}$  ( $200^\circ \text{F}$ ), 25.4 mm (1.0 in.) at  $200^\circ \text{C}$  ( $400^\circ \text{F}$ ), 32 mm (1.25 in.) at  $315^\circ \text{C}$  ( $600^\circ \text{F}$ ).
6. Under windy conditions, 12.1 km/h (7.5 miles/h), 10–20% greater thickness of insulation is justified.

## MIXING AND AGITATION

1. Mild agitation is obtained by circulating the liquid with an impeller at superficial velocities of  $30.48\text{--}60.9 \text{ mm/s}$  (0.1–0.2 ft/s), and intense agitation at  $213.4\text{--}304.8 \text{ mm/s}$  (0.7–1.0 ft/s).
2. Intensities of agitation with impellers in baffled tanks are measured by power input,  $\text{hp}/1000 \text{ gal.}$ , and impeller tip speeds:

<b>Operation</b>	<b>hp/1000 gal.</b>	<b>Tip speed (ft/min)</b>	<b>Tip speed (m/s)</b>
Blending	0.2–0.5		
Homogeneous reaction	0.5–1.5	7.5–10	0.038–0.051
Reaction with heat transfer	1.5–5.0	10–15	0.051–0.076
Liquid-liquid mixtures	5	15–20	0.076–0.10
Liquid-gas mixtures	5–10	15–20	0.076–0.10
Slurries	10		

3. Proportions of a stirred tank relative to the diameter  $D$ : liquid level =  $D$ ; turbine impeller diameter =  $D/3$ ; impeller level above bottom =  $D/3$ ; impeller blade width =  $D/15$ ; four vertical baffles with width =  $D/10$ .
4. Propellers are made with a maximum of 457.2-mm (18-in.) turbine impellers to 2.74 m (9 ft).
5. Gas bubbles sparged at the bottom of the vessel will result in mild agitation at a superficial gas velocity of  $0.0051 \text{ m/s}$  (1 ft/min), severe agitation at  $0.02 \text{ m/s}$  (4 ft/min).
6. Suspension of solids with a settling velocity of  $0.009 \text{ m/s}$  (0.03 ft/s) is accomplished with either turbine or propeller impellers, but when the settling velocity is above  $0.05 \text{ m/s}$  (0.15 ft/s) intense agitation with a propeller is needed.
7. Power to drive a mixture of a gas and a liquid can be 25–50% less than the power to drive the liquid alone.
8. In-line blenders are adequate when a second contact time is sufficient, with power inputs of 0.1–0.2 hp/gal.

## PARTICLE SIZE ENLARGEMENT

1. The chief methods of particle size enlargement are compression into a mold, extrusion through a die followed by cutting or breaking to size, globulation of molten material followed by solidification, agglomeration under tumbling or otherwise agitated conditions with or without binding agents.
2. Rotating drum granulators have length-to-diameter ratios of 2–3, speeds 10–20 rpm, pitch as much as  $10^\circ$ . Size is controlled by speed, residence time, and amount of binder; 2–5 mm diameter is common.
3. Rotary disk granulators produce a more nearly uniform product than drum granulators: fertilizer, 1.5–3.5 mm diameter; iron ore 10–25 mm diameter.
4. Roll compacting and briquetting is done with rolls ranging from 130 mm diameter by 50 mm wide to 910 mm diameter by 550 mm wide. Extrudates are made 1–10 mm thick and are broken down to size for any needed processing, such as feed to tabletting machines or to dryers.

5. Tablets are made in rotary compression machines that convert powders and granules into uniform sizes. The usual maximum diameter is about 38.1 mm (1.5 in.), but special sizes up to 101.6 mm (4 in.) diameter are possible. Machines operate at 100 rpm or so and make up to 10,000 tablets/min.
6. Extruders make pellets by forcing powders, pastes, and melts through a die followed by cutting. A 203.2-mm (8-in.) screw has a capacity of 907.2 kg/h (2000 lb/h) of molten plastic and is able to extrude tubing at 0.76–1.52 m/s (150–300 ft/min) and to cut it into sizes as small as washers at 8000/min. Ring pellet extrusion mills have hole diameters of 1.6–32 mm. Production rates are in the range of 30–200 lb/h-hp.
7. Prilling towers convert molten materials into droplets and allow them to solidify in contact with an air stream. Towers as high as 60 m (196.9 ft) are used. Economically the process becomes competitive with other granulation processes when a capacity of 200–400 tons/day is reached. Ammonium nitrate prills, for example, are 1.6–3.5 mm diameter in the 5–95% range.
8. Fluidized bed granulation is conducted in shallow beds 304.8–609.6 mm (12–24 in.) deep at air velocities of 0.1–2.5 m/s or 3–10 times the minimum fluidizing velocity, with evaporation rates of 0.005–1.0 kg/m<sup>2</sup>s. One product has a size range 0.7–2.4 mm diameter.

## PIPING

1. Line velocities ( $v$ ) and pressure drops ( $\Delta P$ ): (a) For a liquid pump discharge,  $v = (5 + D/3)$  ft/s and  $\Delta P = 0.45$  bar/100 m (2.0 psi/100 ft); (b) For liquid pump suction,  $v = (1.3 + D/6)$  ft/s,  $\Delta P = 0.09$  bar/100 m (0.4 psi/100 ft); (c) for steam or gas flow:  $v = 20D$  ft/s and  $\Delta P = 0.113$  bar/100 m (0.5 psi/100 ft),  $D$  = diameter of pipe in inches.
2. Gas/steam line velocities = 61 m/s (200 ft/s) and pressure drop = 0.1 bar/100 m (0.5 psi/100 ft).
3. In preliminary estimates set line pressure drops for an equivalent length of 30.5 m (100 ft) of pipe between each of piece of equipment.
4. Control valves require at least 0.69 bar (10 psi) pressure drop for good control.
5. Globe valves are used for gases, control and wherever tight shut-off is required. Gate valves are for most other services.
6. Screwed fittings are used only on sizes 38 mm (1.5 in.) or less, flanges or welding used otherwise.
7. Flanges and fittings are rated for 10, 20, 40, 103, 175 bar (150, 300, 600, 900, 1500, or 2500 psig).
8. Approximate schedule number required = 1000  $P/S$ , where  $P$  is the internal pressure psig and  $S$  is the allowable working stress [about 690 bar (10,000 psi)] for A120 carbon steel at 260°C (500°F). Schedule (Sch.) 40 is most common.

## PUMPS

1. Power for pumping liquids:  $kW = (1.67)[\text{Flow (m}^3/\text{min})][\Delta P(\text{bar})]/\varepsilon[\text{hp}] = \text{Flow (gpm)} [\Delta P (\text{psi})/(1,714)(\varepsilon)]$ . ( $\varepsilon$  = fractional efficiency).
2. Net positive suction head (NPSH) of a pump must be in excess of a certain number, depending upon the kind of pumps and the conditions, if damage is to be avoided. NPSH = (pressure at the eye of the impeller-vapor pressure)/( $\rho g$ ). Common range is 1.2–6.1 m (4–20 ft) of liquid.
3. Specific speed  $N_s = (\text{rpm})(\text{gpm})^{0.5}/(\text{head in ft})^{0.75}$ . Pump may be damaged if certain limits of  $N_s$  are exceeded, and efficiency is best in some ranges.
4. Centrifugal pumps: Single stage for 0.057–18.9 m<sup>3</sup>/min (15–5000 gpm), 152 m (500 ft) maximum head; multistage

- for 0.076–41.6 m<sup>3</sup>/min (20–11,000 gpm), 1675 m (5500 ft) maximum head. Efficiency: 45% at 0.378 m<sup>3</sup>/min (100 gpm), 70% at 1.89 m<sup>3</sup>/min (500 gpm), and 80% at 37.8 m<sup>3</sup>/min (10,000 gpm).
5. Axial pumps for 0.076–378 m<sup>3</sup>/min (20–100,000 gpm), 12 m (40 ft) head, 65–85% efficiency.
  6. Rotary pumps for 0.00378–18.9 m<sup>3</sup>/min (1–5000 gpm), 15,200 m (50,000 ft) head, 50–80% efficiency.
  7. Reciprocating pumps for 0.0378–37.8 m<sup>3</sup>/min (10–10,000 gpm), 300 km (1,000,000 ft) maximum head. Efficiency: 70% at 7.46 kW (10 hp), 85% at 37.3 kW (50 hp), and 90% at 373 kW (500 hp).

## REACTORS

1. The rate of reaction in every instance must be established in the laboratory, and the residence time or space velocity and product distribution eventually must be found from a pilot plant.
2. Dimensions of catalyst particles are 0.1 mm (0.004 in.) in fluidized beds, 1 mm in slurry beds, and 2–5 mm (0.078–0.197 in.) in fixed beds.
3. The optimum proportions of stirred tank reactors are with liquid level equal to the tank diameter, but at high pressures slimmer proportions are economical.
4. Power input to a homogeneous reaction stirred tank is 0.1–0.3 kW/m<sup>3</sup> (0.5–1.5 hp/1000 gal.) but three times this amount when heat is to be transferred.
5. Ideal CSTR (continuous stirred tank reactor) behavior is approached when the mean residence time is 5–10 times the length needed to achieve homogeneity, which is accomplished with 500–2000 revolutions of a properly designed stirrer.
6. Batch reactions are conducted in stirred tanks for small daily production rates or when the reaction times are long or when some condition such as feed rate or temperature must be programmed in some way.
7. Relatively slow reactions of liquids and slurries are conducted in continuous stirred tanks. A battery of four or five in series is most economical.
8. Tubular flow reactors are suited to high production rates at short residence times (seconds or minutes) and when substantial heat transfer is needed. Embedded tubes or shell-and-tube constructions then are used.
9. In granular catalyst packed reactors, the residence time distribution is often no better than that of a five-stage CSTR battery.
10. For conversions under about 95% of equilibrium, the performance of a five-stage CSTR battery approaches plug flow.
11. The effect of temperature on chemical reaction rate is to double the rate every 10°C.
12. The rate of reaction in a heterogeneous system is more often controlled by the rate of heat or mass transfer than by the chemical reaction kinetics.
13. The value of a catalyst may be to improve selectivity more than to improve the overall reaction rate.

## REFRIGERATION

1. A ton of refrigeration is the removal of 12,700 kJ/h (12,000 Btu/h) of heat.
2. At various temperature levels: –18 to –10°C (0–50°F), chilled brine and glycol solutions; –45 to –10°C (–50 to –40°F), ammonia, Freon, and butane; –100 to –45°C (–150 to –50°F), ethane or propane.
3. Compression refrigeration with 38°C (100°F) condenser requires kW/tonne (hp/ton) at various temperature levels; 0.93 (1.24) at –7°C (20°F), 1.31 (1.75) at –18°C (0°F); 2.3 (3.1) at –40°C (–40°F); 3.9 (5.2) at –62°C (–80°F).

4. Below  $-62^{\circ}\text{C}$  ( $-80^{\circ}\text{F}$ ), cascades of two or three refrigerants are used.
5. In single-stage compression, the compression ratio is limited to 4.
6. In multistage compression, economy is improved with interstage flashing and recycling, the so-called “economizer operation.”
7. Absorption refrigeration: ammonia to  $-34^{\circ}\text{C}$  ( $-30^{\circ}\text{F}$ ) and lithium bromide to  $7^{\circ}\text{C}$  ( $45^{\circ}\text{F}$ ) is economical when waste steam is available at 0.9 barg (12 psig).

## SIZE SEPARATION OF PARTICLES

1. Grizzlies that are constructed of parallel bars at appropriate spacings are used to remove products larger than 50 mm in diameter.
2. Revolving cylindrical screens rotate at 15–20 rpm and below the critical velocity; they are suitable for wet or dry screening in the range of 10–60 mm.
3. Flat screens are vibrated, shaken, or impacted with bouncing balls. Inclined screens vibrate at 600–7000 strokes/min and are used for down to 38  $\mu\text{m}$ , although capacity drops off sharply below 200  $\mu\text{m}$ . Reciprocating screens operate in the range of 30–1000 strokes/min and handle sizes to 0.25 mm at the higher speeds.
4. Rotary sifters operate at 500–600 rpm and are suited to a range of 12 mm–50  $\mu\text{m}$ .
5. Air classification is preferred for fine sizes because screens of 150 mesh and finer are fragile and slow.
6. Wet classifiers mostly are used to make two product size ranges, oversize and undersize, with a break commonly in the range between 28 and 200 mesh. A rake classifier operates at about 9 strokes/min when making separation at 200 mesh and 32 strokes/min at 28 mesh. Solids content is not critical, and that of the overflow may be 2–20% or more.
7. Hydrocyclones handle up to 600 ft<sup>3</sup>/min and can remove particles in the range of 300–5  $\mu\text{m}$  from dilute suspensions. In one case, a 20-in. diameter unit had a capacity of 1000 gpm with a pressure drop of 5 psi and a cutoff between 50 and 150  $\mu\text{m}$ .

## UTILITIES, COMMON SPECIFICATIONS

1. Steam: 1–2 bar (15–30 psig), 121–135° C (250–275° F); 10 barg (150 psig), 186° C (366° F); 27.6 barg (400 psig), 231° C (448° F); 41.3 barg (600 psig), 252° C (488° F) or with 55–85° C (100–150° F) superheat.
2. Cooling water: For design of cooling tower use, supply at 27–32° C (80–90° F); from cooling tower, return at 45–52° C (115–125° F); return seawater at 43° C (110° F); return tempered water or steam condensate above 52° C (125° F).
3. Cooling air supply at 29–35° C (85–95° F); temperature approach to process, 22° C (40° F).
4. Compressed air at 3.1 (45), 10.3 (150), 20.6 (300), or 30.9 barg (450 psig) levels.
5. Instrument air at 3.1 barg (45 psig),  $-18^{\circ}\text{C}$  ( $0^{\circ}\text{F}$ ) dew point.
6. Fuels: gas of 37,200 kJ/m<sup>3</sup> (1000 Btu/SCF) at 0.35–0.69 barg (5–10 psig), or up to 1.73 barg (25 psig) for some types of burners; liquid at 39.8 GJ/m<sup>3</sup> (6 million British Thermal unit per barrel).
7. Heat-transfer fluids: petroleum oils below 315° C (600° F), Dowtherms below 400° C (750° F), fused salts below 600° C (1100° F), and direct fire or electricity above 232° C (450° F).
8. Electricity: 0.75–74.7 kW (1–100 hp), 220–550 V; 149–1864 kW (200–2500 hp), 2300–4000 V.

## VESSELS (DRUMS)

1. Drums are relatively small vessels to provide surge capacity or separation of entrained phases.
2. Liquid drums are usually horizontal.
3. Gas/liquid phase separators are usually vertical.
4. Optimum length/diameter = 3, but the range 2.5–5.0 is common.
5. Holdup time is 5 min half-full for reflux drums and gas/liquid separators, 5–10 min for a product feeding another tower.
6. In drums feeding a furnace, 30 min half-full drum is allowed.
7. Knockout drums placed ahead of compressors should hold no less than 10 times the liquid volume passing through per minute.
8. Liquid/liquid separators are designed for a settling velocity of 0.85–1.27 mm/s (2–3 in./min).
9. Gas velocity in gas/liquid separators,  $v = k\sqrt{\rho_L/\rho_V - 1}$  m/s (ft/s), with  $k = 0.11$  (0.35) for systems with a mesh deentrainer and  $k = 0.0305$  (0.1) without a mesh deentrainer.
10. Entrainment removal of 99% is attained with 102–305 mm (4–12 in.) mesh pad thickness; 152.5 mm (6 in.) thickness is popular.
11. For vertical pads, the value of the coefficient in step 9 is reduced by a factor of 2/3.
12. Good performance can be expected at velocities of 30–100% of those calculated with the given  $k$ ; 75% is popular.
13. Disengaging spaces of 152–457 mm (6–18 in.) ahead of the pad and 305 mm (12 in.) above the pad are suitable.
14. Cyclone separators can be designed for 95% collection of 5- $\mu\text{m}$  particles, but usually only droplets greater than 50  $\mu\text{m}$  need be removed.

## VESSEL (PRESSURE)

1. Design temperature between  $-30$  and  $345^{\circ}\text{C}$  ( $-20^{\circ}\text{F}$  and  $650^{\circ}\text{F}$  if  $50^{\circ}\text{F}$ ) above maximum operating temperature; higher safety margins are used outside the given temperature range.
2. The design pressure is 10% or 0.69–1.7 bar (10–25 psi) over the maximum operating pressure, whichever is greater. The maximum operating pressure, in turn, is taken as 1.7 bar (25 psi) above the normal operation.
3. Design pressures of vessels operating at 0–0.69 barg (0–10 psig) and 95–540° C (200–1000° F) are 2.76 barg (40 psig).
4. For vacuum operation, design pressures are 1 barg (15 psig) and full vacuum.
5. Minimum wall thickness for rigidity: 6.4 mm (0.25 in.) for 1.07 m (42 in.) diameter and under, 8.1 mm (0.32 in.) for 1.07–1.52 m (42–60 in.) diameter, and 9.7 mm (0.38 in.) for over 1.52 m (60 in.) diameter.
6. Corrosion allowance 8.9 mm (0.35 in.) for known corrosive conditions, 3.8 mm (0.15 in.) for noncorrosive streams, and 1.5 mm (0.06 in.) for steam drums and air receivers.
7. Allowable working stresses are one-fourth the ultimate strength of the material.
8. Maximum allowable stress depends sharply on temperature

Temperature (° F) (° C)	-20–650 -30–345	750 400	850 455	1,000 540
Low-alloy steel, SA 203 (psi) (bar)	18,759 1,290	15,650 1,070	9,550 686	2,500 273
Type 302 stainless (psi) (bar)	18,750 1,290	18,750 1,290	15,950 1,100	6,250 431

**VESSELS (STORAGE TANKS)**

1. For less than  $3.8 \text{ m}^3$  (1000 gal.), use vertical tanks on legs.
2. For  $3.8\text{--}38 \text{ m}^3$  (1000–10,000 gal.), use horizontal tanks on concrete supports.
3. Beyond  $38 \text{ m}^3$  (10,000 gal.) use vertical tanks on concrete foundations.
4. Liquids subject to breathing losses may be stored in tanks with floating or expansion roofs for conservation.
5. Freeboard is 15% below  $1.9 \text{ m}^3$  (500 gal.) and 10% above  $1.9 \text{ m}^3$  (500 gal.) capacity.

6. A 30-day capacity often is specified for raw materials and products but depends on connecting transportation equipment schedules.
7. Capacities of storage tanks are at least 1.5 times the size of connecting transportation equipment; for instance,  $28.4\text{-m}^3$  (7500 gal.) tanker trucks,  $130\text{-m}^3$  (34,500 gal.) rail cars, and virtually unlimited barge and tanker capacities.

*Source:* The above mentioned rules of thumb have been adapted from Walas, S.M., *Chemical Process Equipment: Selection and Design*, copyright 1988 with permission from Elsevier, all rights reserved.

**Physical Property Heuristics**

	Units	Liquids	Liquids	Gases	Gases	Gases
Heat Capacity	$\text{kJ/kg}^\circ\text{C}$	Water 4.2	Organic Material 1.0–2.5	Steam 2.0	Air 1.0	Organic Material 2.0–4.0
Density	$\text{kg/m}^3$	1000	700–1500		1.29 at STP	
Latent Heat	$\text{kJ/kg}$	1200–2100	200–1000			
Thermal Conductivity	$\text{W/m}^\circ\text{C}$	0.55–0.70	0.10–0.20	0.025–0.07	0.025–0.05	0.02–0.06
Viscosity	$\text{kg/ms}$	$0^\circ\text{C } 1.8 \times 10^{-3}$ $50^\circ\text{C } 5.7 \times 10^{-4}$ $100^\circ\text{C } 2.8 \times 10^{-4}$ $200^\circ\text{C } 1.4 \times 10^{-4}$	Wide Range	$10\text{--}30 \times 10^{-6}$	$20\text{--}50 \times 10^{-6}$	$10\text{--}30 \times 10^{-6}$
Prandtl No.		1–15	10–1000	1.0	0.7	0.7–0.8

(Source: Turton, R. et al., *Analysis, Synthesis, and Design of Chemical Processes*, Prentice Hall International Series, 2001.)

**Typical Physical Property Variations with Temperature and Pressure**

	Liquids	Liquids	Gases	Gases
Property Density	$\rho_l \propto (T_c - T)^{0.3}$	Pressure Negligible	$\rho_g = \frac{\text{MW } P / ZRT}{T^{1.5}}$	Pressure $\rho_g = \frac{\text{MW } P / ZRT}{T^{1.5}}$
Viscosity	$\mu_l = Ae^{B/T}$	Negligible	$\mu_g \propto \frac{1}{(T + 1.47T_b)}$	Significant only for $> 10 \text{ bar}$
Vapor Pressure	$P^* = ae^{b/(T+c)}$	–	–	–

*Note:*  $T$  is temperature (K),  $T_c$  is the critical Temperature (K),  $T_b$  is the normal boiling point (K), MW is molecular weight,  $P$  is pressure,  $Z$  is compressibility,  $R$  is the gas constant, and  $P^*$  is the vapor pressure.

(Source: Turton, R. et al., *Analysis, Synthesis, and Design of Chemical Processes*, Prentice Hall International Series, 2001.)

**Capacities of Process Units in Common Usage<sup>a</sup>**

Process unit	Capacity Unit	Maximum Value	Minimum Value	Comment
Horizontal Vessel	Pressure (bar)	400	Vacuum	L/D typically 2–5
	Temper. ( $^\circ\text{C}$ )	400 <sup>b</sup>	–200	
	Height (m)	10	2	
	Diameter (m)	2	0.3	
	L/D	5	2	
Vertical Vessel	Pressure (bar)	400	400	L/D typically 2–5
	Temperature ( $^\circ\text{C}$ )	400 <sup>b</sup>	–200	
	Height (m)	10	2	
	Diameter (m)	2	0.3	
	L/D	5	2	
Towers	Pressure (bar)	400	Vacuum	Normal Limits Diameter $L/D$
	Temperature ( $^\circ\text{C}$ )	400 <sup>b</sup>	–200	
	Height (m)	50	2	
	Diameter (m)	4	0.3	
	L/D	30	2	

3.0–40<sup>c</sup>  
2.5–30<sup>c</sup>  
1.6–23<sup>c</sup>  
1.8–13<sup>c</sup>

<b>Pumps</b>				
Reciprocating	Power <sup>d</sup> (kW)	250	<0.1	
	Pressure (bar)	1,000		
Rotary and Positive Displacement	Power <sup>d</sup> (kW)	150	<0.1	
	Pressure (bar)	300		
Centrifugal	Power <sup>d</sup> (kW)	250	<0.1	
	Pressure (bar)	300		
<b>Compressors</b>				
Axial, Centrifugal + Recipr.	Power <sup>d</sup> (kW)	8,000	50	
Rotary	Power <sup>d</sup> (kW)	1,000	50	
<b>Drives for Compressors</b>				
Electric	Power <sup>e</sup> (kW)	15,000	<1	
Steam Turbine	Power <sup>e</sup> (kW)	15,000	100	
Gas Turbine	Power <sup>e</sup> (kW)	15,000	10	
Internal Combustion Eng.	Power <sup>e</sup> (kW)	15,000	10	
Process Heaters	Duty (MJ/h)	500,000	10,000	Duties different for reactive heaters/furnaces.
Heat Exchangers	Area (m <sup>2</sup> )	1,000	10	For Area < 10 m <sup>2</sup> use double-pipe exchanger.
	Tube Dia. (m)	0.0254	0.019	
	Length (m)	6.5	2.5	
	Pressure (bar)	150	Vacuum	For 150<P<400 bar need special design.
	Temperature (° C)	400 <sup>b</sup>	-200	

<sup>a</sup> Most of the limits for equipment sizes shown here correspond to the limits used in the costing program (CAPCOST.BAS).

<sup>b</sup> Maximum temperature and pressure are related to the materials of construction and may differ from values shown here.

<sup>c</sup> For 20<L/D<30 special design may be required. Diameter up to 7 m possible but over 4 m must be fabricated on site.

<sup>d</sup> Power values refer to fluid/pumping power.

<sup>e</sup> Power values refer to shaft power.

(Source: Turton, R. et al., *Analysis, Synthesis, and Design of Chemical Processes*, Prentice Hall International Series, 2001.)

### Effect of Typical Materials of Construction on Product Color, Corrosion, Abrasion, and Catalytic Effects

<b>Metals</b>		
<b>Material</b>	<b>Advantages</b>	<b>Disadvantages</b>
Carbon Steel	Low cost, readily available, resists abrasion, standard fabrication, resists alkali	Poor resistance to acids and strong alkali, often causes discoloration and contamination
Stainless Steel	Resists most acids, reduces discoloration, available with a variety of alloys, abrasion less than mild steel	Not resistant to chlorides, more expensive, fabrication more difficult, alloy materials may have catalytic effects
Monel-Nickel	Little discoloration, contamination, resistant to chlorides	Not resistant to oxidizing environments, expensive
Hastelloy	Improved over Monel-Nickel	More expensive than Monel-Nickel
Other Exotic Metals	Improves specific properties	Very high cost
<b>Non-Metals</b>		
Glass	Useful in laboratory and batch system, low diffusion at walls	Fragile, not resistant to high alkali, poor heat transfer, poor abrasion resistance
Plastics	Good at low temperature, large variety to select from with various characteristics, easy to fabricate, seldom discolors, minor catalytic effects possible	Poor at high temperature, low strength, not resistant to high alkali conditions, low heat transfer, low cost
Ceramics	Withstands high temperatures, variety of formulations available, modest cost	Poor abrasion properties, high diffusion at walls (in particular hydrogen), low heat transfer, may encourage catalytic reactions

(Source: Turton, R. et al., *Analysis, Synthesis, and Design of Chemical Processes*, Prentice Hall International Series, 2001.)

# PROCESS PLANNING, SCHEDULING, AND FLOWSHEET DESIGN

**P**rocess engineering design is the application of chemical, mechanical, petroleum, gas and other engineering talents to the process-related development, planning, designs, and decisions required for economical and effective completion of a process project [1]. Although process design engineers are organizationally located in research, technical service, economic evaluation, as well as other specific departments, the usual arrangement is to have them available to the engineering groups concerned with developing the engineering details as well as to evaluate bids for the various equipment items. Process design is usually a much more specific group responsibility in engineering contractor organizations than in a chemical or petrochemical production company, and the degree of distinction varies with the size of the organization.

The average process engineer has the following responsibilities:

1. prepares studies of process cycles and systems for various product production or improvements or changes in existing production units; prepares material and heat balances;
2. prepares economic studies associated with process performance;
3. designs and/or specifies equipment items required to define the process flowsheet or flow system; specifies corrosion-resistant materials of construction;

4. evaluates competitive bids for equipment;
5. evaluates operating data for existing or test equipment;
6. guides flowsheet draftsmen in detailed flowsheet preparation.

The process engineer also develops tests and interprets data and information from the research pilot plant. He/she aids in scaling up the research type flow cycle to one of commercial feasibility.

The process engineer must understand the interrelationship between the various research, engineering, purchasing, expediting, construction, and operational functions of a project. He/she must appreciate that each function may and often does affect or influence the process design decisions. For example, it is futile to waste time designing or calculating in detail when the basic components of the design cannot be economically fabricated, or if capable of being fabricated, cannot possibly be delivered by the construction schedule for the project. Some specific phases of a project that require process understanding include plant layout, materials of construction for corrosion as well as strength, start-up operations, troubleshooting, maintenance, performance testing, and so forth.

## 1.1 ORGANIZATIONAL STRUCTURE

The process design function may be placed in any one of several workable locations in an organization. These locations will be influenced by the primary function of the overall company, that is, chemical production, engineering, engineering sales, design, and manufacture of packaged or specific equipment manufacture, and so on. For best efficiency, regardless of the business nature of the company, the process design being a specialty type operation, works best when specifically identified and given the necessary freedom of contact within and without the company to maintain a high level of practical, yet thorough direction.

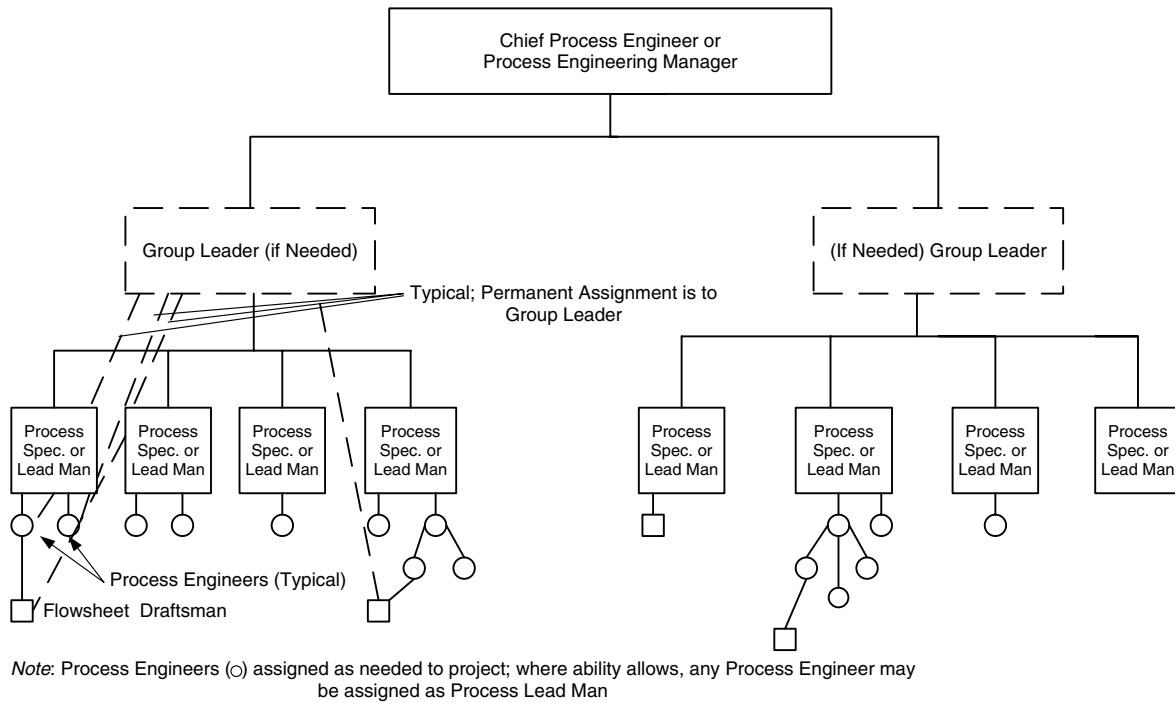
A typical working arrangement is shown in Figure 1-1 [1]. In a consulting or engineering contractor organization, process design and/or process engineering is usually a separate group responsible for developing the process with the customer, or presenting the customer with a turnkey proposed process.

In an operating or producing chemical or petrochemical company, the process engineering and design may be situated in a research, technical service, or engineering department. In most cases it is associated with an engineering department if new projects and processes are being planned for the company. If located elsewhere, the designs and planning must be closely coordinated with the engineering activity.

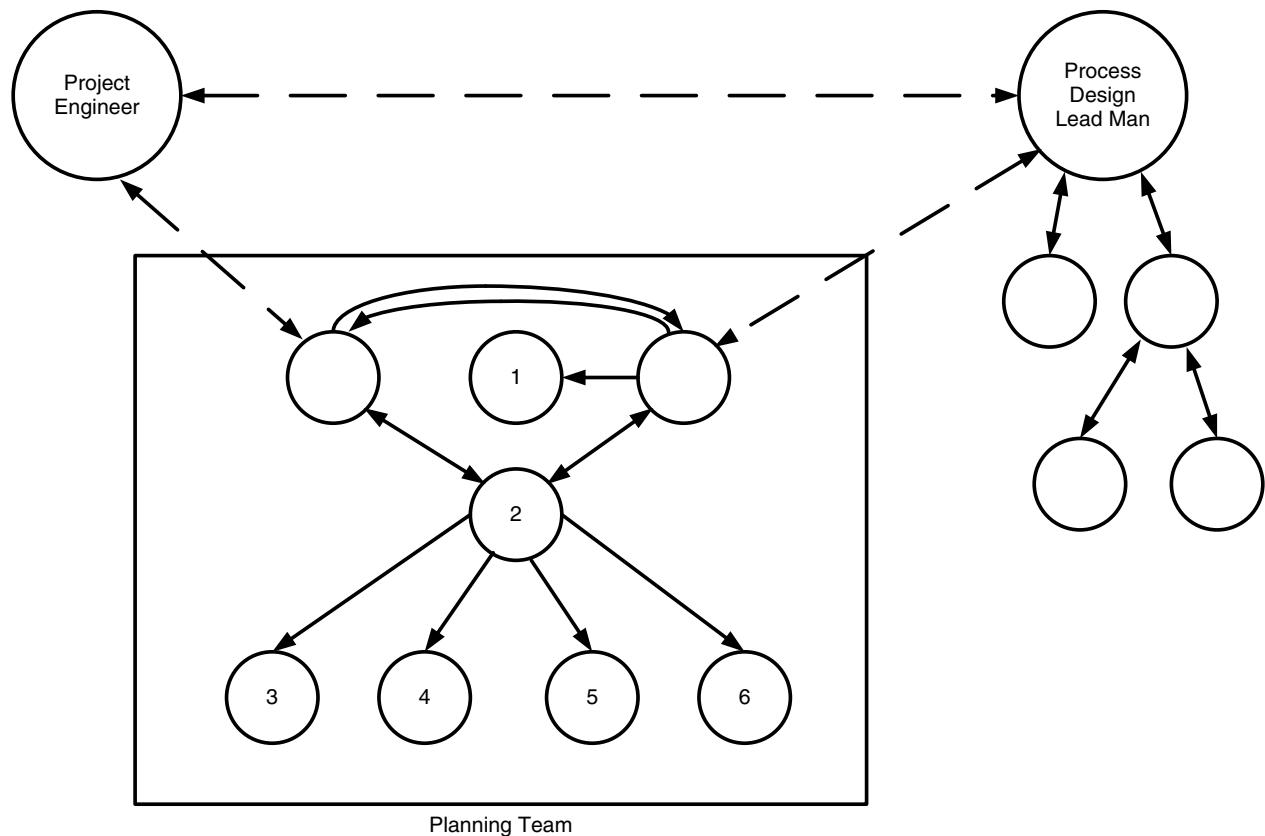
Most current thinking establishes a project team headed by a project engineer or manager to oversee the accomplishment of a given plant development for a process company. If the projects or jobs are small, then the scope of activity is limited and may often be consolidated in a single individual for project and process responsibility. For projects larger than \$500,000, the project and process responsibility usually are best kept separate in order to expedite the specific accomplishment of the process design phase. When the process design engineer is required to interpret calculations and specification development and to follow some electrical, structural, or even expediting delivery question or problem, the design work cannot be completed at best efficiency and often the quality of process design suffers, assuming there is a fixed target date for completion of the various phases as well as the overall project.

Figure 1-2 diagrammatically suggests a team arrangement for accomplishing the planning of a process project. The arrows indicate directions of flow of communications and also the tie-in relationship of the process design function in the accomplishment of an assignment. The planning team in the box works to place the proper perspective on all phases of the engineering functions by developing a working atmosphere of understanding for accomplishing the engineering design. This is physically represented by mechanical vessels, piping, structures, electrical, instrumentation, civil, and any other specialized functions. In many projects, the Lead

## 2 PROCESS PLANNING, SCHEDULING, AND FLOWSHEET DESIGN



**Figure 1-1** A process engineering section supervision chart. (By permission from Ludwig [1].)



**Figure 1-2** Typical organization of “engineering planning team”. (By permission from Ludwig [2].)

Process Engineer and the Project Lead Engineer are the only individuals who see the details of the overall scope of the project.

## 1.2 PROCESS DESIGN SCOPE

The project engineer appoints a chief process engineer who puts together a process design team to be responsible for all the chemical engineering aspects of the plant. The term “process design” is used here to include what is sometimes referred to as process engineering. Yet in some process engineering operations, all process design functions may not be carried out in detail. As discussed, process design is intended to include the following:

1. process material and heat balances;
2. process cycle development, correlation of pilot or research data, and correlation of physical property data;
3. auxiliary services material and heat balances;
4. flowsheet development and detailed completion;
5. chemical engineering performance design for specific items of equipment required for a flowsheet, and mechanical interpretation of this to a practical and reasonable specification. Here, the process requirements are converted into hardware details to accomplish the process end results at each step in the product production process;
6. instrumentation as related to process performance, presentation, and interpretation of requirements to instrument specialists;
7. process interpretation for proper mechanical, structural, civil, electrical, instrument, and so on, handling of the respective individual phases of the project;
8. preparation of specifications in proper form and/or detail for use by the project team as well as for the purchasing function;
9. evaluation of bids and recommendation of qualified vendor.

Most of the functions are fairly self-explanatory; therefore, emphasis will be placed only on those requiring detailed explanation. Another process design scope issue for the process design team is to identify and scope any key layout issues early in the design effort. Such issues include concerns like

1. locations where gravity feed is used;
2. product quality concerns where product residence times must be minimized;
3. equipment requiring easy access for maintenance.

## 1.3 ROLE OF THE PROCESS DESIGN ENGINEER

Although the working role of the process design engineer may include all of the technical requirements listed above, it is very important to recognize what this entails in some detail. The process design engineer, in addition to being capable of participating in evaluation of research and pilot plant data and the conversion of this data into a proposed commercial process scheme, must also do the following:

1. Prepare heat and mass balance studies for a proposed process, both “by hand” and by use of computer programs. The use of a spreadsheet package (e.g., Excel) is now prevalent in accomplishing these calculations.
2. Prepare rough cost economics, including preliminary sizing and important details of equipment, factor to an order of magnitude capital cost estimate [3] (see also [2]), prepare a production cost estimate and work with economic evaluation representatives to establish a payout and the financial economics of the proposed process.
3. Participate in layout planning for the proposed plant [4, 5].

4. Prepare final detailed heat and material balances.
5. Prepare detailed sizing of all process equipment and possibly some utility systems. It is important that the process engineer *visualizes* the flow and processing of the fluids through the system and *inside* the various items of equipment in order to adequately recognize what will take place during the process.
6. Prepare/supervise preparation of draft of process flowsheets for review by others.
7. Prepare/supervise preparation of piping or mechanical flow diagram or piping and instrumentation diagram (P&ID), with necessary preliminary sizing of all pipe lines, distillation equipment, pumps, compressors, and so on, and representation of all instrumentation for detailing by instrument engineers.
8. Prepare mechanical and process specifications for all equipment, tanks, pumps, compressors, separators, drying systems, and refrigeration systems. This must include the selection of materials of construction and safety systems and the coordination of specifications with instrumentation and electrical requirements.
9. Determine size and specifications for all safety relief valves and/or rupture disks for process safety relief (including runaway reactions) and relief in case of external fire.
10. Prepare valve code specifications for incorporation on item 7 above, or select from existing company standards for the fluids and their operating conditions (see Figures 1-39 and 1-40).
11. Select from company insulation standards (or prepare, if necessary) the insulation codes to be applied to each hot or cold pipe or equipment. Note that insulation must be applied in some cases only to prevent operating personnel from contacting the base equipment. Table 1-1 shows typical insulation thickness from which code numbers can be established.
12. Establish field construction hydraulic test pressures for each process equipment. Sometimes the equipment is blanked or blocked off, and no test pressure is applied in the field, because all pressure equipment must be tested in the fabricators’ or manufacturers’ shop as per American Society of Mechanical Engineers (ASME) Code.
13. Prepare drafts of line schedule and/or summary sheets (Figures 1.31a–1-31d), and equipment summary schedules (Figures 1-34–1-37), plus summary schedules for safety relief valves and rupture disks, compressors and other major equipment. *Some of the process data sheets and equipment schedules (over 30) are readily available for downloading from the companion website.*
14. Prepare detailed process and mechanical specifications for developing proposals for purchase by the purchasing department.
15. Participate and possibly lead the process hazard reviews (i.e., hazard and operability studies (HAZOP) Chapter 9).

The process design engineer actually interprets the process into appropriate hardware (equipment) to accomplish the process requirements. Therefore, the engineer must be interested in and conversant with the layout of the plant; the relationship of equipment for maintenance; the safety relationships of equipment in the plant; the possibilities for fire and/or explosion; the possibilities for external fire on the equipment areas of the plant; the existence of hazardous conditions, including toxic materials and pollution, that could arise; and, in general, the overall picture.

The engineer’s ability to recognize the interrelationships of the various engineering disciplines with the process requirements is essential to thorough design. For example, the recognition of metallurgy and certain metallurgical testing requirements as they relate to the corrosion in the process environment is absolutely

**TABLE 1-1 Typical Thickness Chart – Insulation for Services 70°F through 1200°F. Piping Vessels, and Equipment – 36" Diameter and Smaller**

Pipe Size	Insulation Thickness				
	1"	1½"	2"	2½"	3"
≤2½"	700°F	1000°F	1200°F		
≤3"	700	900	1100	1200°F	
≤4"	700	900	1100	1200	
≤6"	600	800	1000	1200	
≤8"	–	800	1000	1200	
≤10"	–	800	1000	1200	
≤12"	–	800	1000	1200	
≤14"	–	800	1000	1100	1200°F
≤16"	–	800	900	1100	1200
≤18"	–	800	900	1100	1200
≤20"	–	800	900	1100	1200
≤24"	–	800	900	1100	1200
≤30"	–	800	900	1100	1200
≤36"	–	800	900	1000	1200

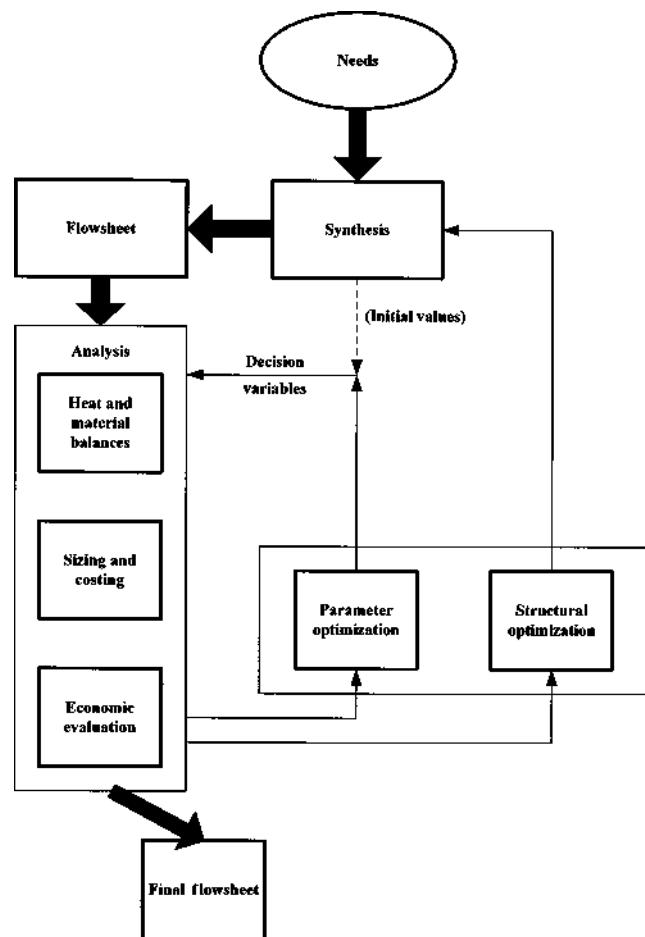
**Notes:**

1. Temperatures in chart are maximum operating temperatures in degrees Fahrenheit for given thickness.
2. All hot insulated piping shall be coded, including piping insulated for personnel protection. Thickness is a function of insulation composition.

necessary to obtain a reliable process design and equipment specification. An example of the importance of this is hydrogen brittleness (see latest charts [6]). Another important area is water service [7]. The engineer selecting the materials of construction should recognize the importance of plastics and plastic composites in the design of industrial equipment and appreciate that plastics often serve as better corrosion-resistant materials than do metals.

#### 1.4 COMPUTER-AIDED FLOWSHEETING

The term “flowsheeting” with computer application performs steady state heat and mass balancing, sizing, and costing calculations for a chemical process. There are many approaches to computer-aided flowsheeting, but the principal ones are sequential modular flowsheeting simulation and equation-based flowsheeting simulation. Figure 1-3 shows the partitioning of the design step into three basic steps: synthesis, analysis, and optimization. The flowsheet structure is chosen from the synthesis where the particular equipment is being used and its interconnections are selected. The synthesis step allows one to set initial values for the variables. The second step is analysis, which is broken into three parts: solving the heat and material balances, sizing and costing the equipment, and economic evaluation. The final step is parameter and structural optimization. During analysis of a given flowsheet, some pertinent parameters such as temperature and pressure can adversely influence the resulting equipment sizes and invariably the flowsheet evaluation. Consequently, a decision is made either to alter the equipment and its interconnection because an improvement is required or to revise the present version of the flowsheet due to high cost. Structural optimization refers to changing the equipment type and its interconnection while parameter optimization deals with altering the temperature or pressure levels within a fixed flowsheet. The final flowsheet with its decision variable values and the resulting flowsheet illustrate the final design. Flowsheeting is the examination of a flowsheet of the process together with the complete system characteristic. It does not include dimensions and structural design of the plants nor instrumentation and planning of the piping network system. The steps involved in flowsheeting are discussed in the following sections.



**Figure 1-3** Partitioning the process design task into interrelated subtasks.  
(Source: Westerberg et al. [8].)

## DEVELOPING A SIMULATION MODEL

A simple flowsheet with a recycle stream consists of four functional units (two mixers, a reactor, and a separator) with a recycle stream. The equations for this flowsheet as depicted in Figure 1-4 [8] are as follows:

### 1. Model equations

$$\left. \begin{array}{l} f_1(x_{11}, x_{12}, y_{11}) = 0 \\ f_2(x_{21}, y_{21}, u_2) = 0 \\ f_3(x_{31}, y_{31}, y_{32}, u_3) = 0 \\ f_4(x_{41}, x_{42}, y_{41}) = 0 \end{array} \right\} \quad (1-1)$$

where

$x_{ij}$  = input  $j$  to unit  $i$

$y_{ij}$  = output  $j$  from unit  $i$

$u_i$  = unit parameters for unit  $i$ .

### 2. Connection equations

$$\left. \begin{array}{l} x_{12} - y_{31} = 0 \\ x_{21} - y_{11} = 0 \\ x_{31} - y_{21} = 0 \\ x_{42} - y_{32} = 0 \end{array} \right\} \quad (1-2)$$

The model equations are written functionally and they represent the heat and material balances, the physical property and other correlations, and so on, thereby enabling one to relate the unit inputs and outputs. A number of equipment parameters  $u_i$  exist for each unit, and must be specified to complete its description. A functional unit such as the reactor volume and its operating temperature and pressure are present among these parameters. In contrast, the mixer as considered here has no other parameters and is therefore not considered in functions  $f_1$  and  $f_2$ .

A list of interconnecting streams can be presented to the computer for the simple flowsheet specification, where the input and output streams can be named for each unit. Here, the following lists represent the flowsheet for Figure 1-4.

Streams	Link on Flowsheet
S1	Feed $x_{11}$
S2	$y_{31} - x_{12}$
S3	$y_{11} - x_{21}$
S4	$y_{21} - x_{31}$
S5	$y_{32} - x_{42}$
S6	Feed $x_{41}$
S7	Output $y_{41}$

Unit	Model Designators	Feed Streams	Output Streams
1	Mixer	S1, S2	S3
2	Extent of conversion reactor	S3	S4
3	Adiabatic flash separator	S4	S2, S5
4	Mixer	S5, S6	S7

The model designator informs the computer which equations the engineer wishes to use to model each unit. This is because the flowsheeting system could have a number of different separator types or reactor types, each with a different designator and equation set in its library of unit types.

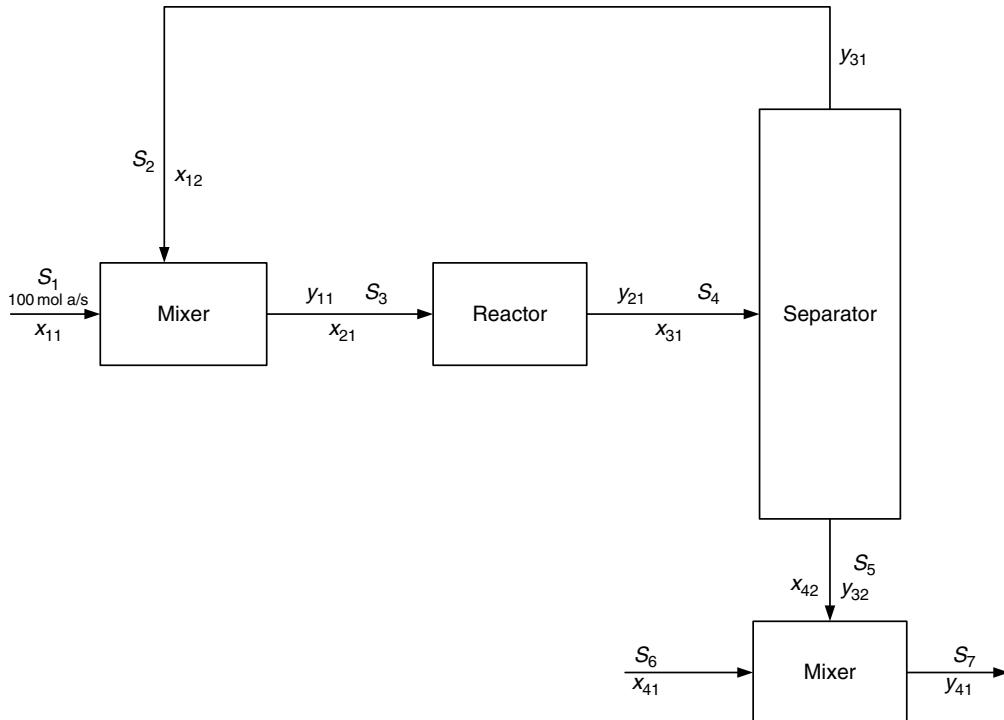


Figure 1-4 Simplified flowsheet. (Source: Westerberg et al. [8].)

## 6 PROCESS PLANNING, SCHEDULING, AND FLOWSHEET DESIGN

### 3. Elements of a modular simulation

The process flowsheet is constructed in terms of blocks or modules (i.e., process units or operations) and streams connecting them. Names that might be given to these blocks are as follows.

MIX	Mix several inlet streams adiabatically to form one product stream
SPLIT	Split a single inlet stream into two or more product streams with the same composition and temperature
COMPRESS	Raise the pressure of a gas by a specified amount
PUMP	Raise the pressure of a liquid by a specified amount
FLASH	Convert a liquid stream at one pressure to liquid and vapor streams in equilibrium at a lower pressure
DISTIL EXTRACT CRYSTAL ABSORB REACT	Simulate the separation process of distillation, extraction, crystallization, and absorption

Table 1-2 [9] provides a partial list of the blocks and modules for the four major commercial simulators. In most simulators, new subroutines (blocks or modules) may be programmed by a

**TABLE 1-2 Unit Subroutines**

ASPEN PLUS		
Crushers	CRUSHER	Gyratory-jaw crusher, cage mill breaker, and single or multiple roll crushers
Solid-solid separators	SCREEN	Solid-solid separation using screens
Gas-solid separators	FABFL CYCLONE VSCRUB EXP	Fabric filters Cyclones Venturi scrubbers Dry electrostatic precipitators
Liquid-solid separators	HYCYC CFUGE FILTER	Hydrocyclones Centrifuge filters Continuous rotary vacuum filters
Crystallizers	CRYSTALLIZER	Mixed suspension, mixed product removal crystallizer
Washers	SWASH CCD SSPLIT	Single-stage solids washer Multistage washer or countercurrent decanter Substream splitter
Mixers and splitters	MIXER FSPLIT	Stream mixer Stream splitter
Separators	SEP SEP2	Component separator – multiple outlets Component separator – two outlets
Flash drums	FLASH2 FLASH3	Two-outlet flash drums Three-outlet flash drums
Approximate distillation	DSTWU DISTL SCFRAC	Winn-Underwood-Gilliland design Edmister simulation Edmister simulation-complex columns
Multistage separation (Equilibrium-based simulation)	RADFRAC MULTIFRAC PETROFRAC ABSBR EXTRACT	Two and three phases, with or without reaction Ditto- with interlinked column sections Ditto- for petroleum refining Absorbers and Strippers Liquid-liquid extractors
(Mass-transfer simulation)	RATEFRAC	Two-phases-mass-transfer model for staged or packed columns
Heat exchange	HEATER HEATX MHEATX	Heater or cooler Two-stream heat exchanger Multistream heat exchanger

user and incorporated into the library. These in turn can call on the extensive libraries of subroutines and data banks provided by the process simulators for estimation of the thermophysical and transport properties, equipment sizes, costs, and so forth.

### SIMULATE A CHEMICAL REACTOR

An additional block type is the convergence block that performs such calculations using the Wegstein algorithm. Here, the output stream from this block contains the assumed set of tear stream variables (i.e., within the cycle). Correspondingly, the input stream contains the values calculated by working around the cycle. If the assumed and calculated tear stream variables agree within a specified tolerance, the solution is found, otherwise the new values or some combination of the new and old values are used to start another computation around the cycle. This process is iterative until convergence is achieved.

### 1.5 THE SEQUENTIAL MODULAR SIMULATION

The material balance of a process defines the flow of material within the process in terms of mass, mols, and usually

(continued)

**TABLE 1-2—(continued)**

<b>ASPN PLUS</b>		
Reactors	RSTOIC RYIELD RGIBBS REQUAL RCSTR RPLUG	Extent of reaction specified Reaction yields specified Multiphase, chemical equilibrium Two-phase, chemical equilibrium Continuous-stirred tank reactor Plug-flow tubular reactor
Pumps, compressors, and turbines	PUMP COMPR MCOMPR VALVE	Pump or hydraulic turbine Compressor or turbine Multistage compressor or turbine Control valves and pressure reducers
Pipeline	PIPE PIPELINE	Pressure drop in a pipe Pressure drop in a pipeline
Steam manipulators	MULT DUPL	Stream multiplier Stream duplicator
<b>HYSYS</b>		
Mixers and splitters	Mixer Tee	Stream mixer Stream splitter
Separators	Component Splitter	Component separator – two outlets
Flash drums	Separator 3-Phase separator Tank	Multiple feeds, one vapor and one liquid product Multiple feeds, one vapor and two liquid products Multiple feeds, one liquid product
Approximate distillation	Shortcut Column	Fenske-Underwood design
Multistage separation (Equilibrium-based simulation)	Column	Generic multiphase separation, including absorber, stripper, rectifier, distillation, liquid-liquid extraction. Additional strippers and pump-arounds can be added. All models support two or three phases and reactions. Physical property models are available for petroleum-refining applications.
Heat exchange	Cooler/Heater Heat Exchanger Lng	Cooler or Heater Two-stream heat exchanger Multistream heat exchanger
Reactors	Conversion Reactor Equilibrium Reactor Gibbs Reactor CSTR PFR	Extent of reaction specified Equilibrium reaction Multiphase chemical equilibrium (stoichiometry not required) Continuous-stirred tank reactor Plug-flow tubular reactor
Pumps, compressors, and turbines	Pump Compressor Expander Valve	Pump or hydraulic turbine Compressor Turbine Adiabatic valve
Pipeline	Pipe Segment	Single/multiphase piping with heat transfer
<b>CHEMCAD</b>		
Mixers and splitters	MIXE DIVI	Stream mixer Stream splitter
Separators	CSEP CSEP	Component separator – multiple outlets Component separator – two outlets
Flash drums	FLAS LLVF VALV	Two-outlet flash drums Three-outlet flash drums Valve
Approximate distillation	SHOR	Winn-Underwood-Gilliland design
Multistage separation (Equilibrium-based simulation)	SCDS, TOWR TPLS EXTR	Two and three phases, with or without reaction Ditto- with interlinked column sections Liquid-liquid extractors
Heat exchange	HTXR HTXR LNGH FIRE	Heater or Cooler Two-stream heat exchanger Multistream heat exchanger Fired heater
Reactors	REAC EREA GIBS KREA KREA	Extent of reaction specified Two-phase chemical equilibrium Multiphase, chemical equilibrium Continuous-stirred tank reactor Plug-flow tubular reactor

(continued)

**TABLE 1-2—(continued)**

Pumps, compressors, and turbines	PUMP COMP, EXPN	Pump or hydraulic turbine Compressor or Turbine
Pipeline	PIPE	Pressure drop in a pipe
Stream manipulators	SREF SREF	Stream multiplier Stream duplicator
		<i>PRO/II</i>
Mixers and splitters	MIXER SPLITTER	Combines two or more streams Splits a single feed or mixture of feeds into two or more streams
Flash drums	FLASH	Calculates the thermodynamic state of any stream when two variables are given by performing phase equilibrium calculations
Distillation column	COLUMN	Splits feed stream(s) into its components based on temperature and pressure By default, a distillation column includes a condenser and reboiler
Heat exchanger	HX	Heats or cools a single process stream, exchanges heat between two process streams or between a process and utility stream
	HXRIG	Rates a TEMA shell-and-tube heat exchanger rigorously calculating heat transfer and pressure drop
	LNGHX	Exchanges heat between any number of hot and cold streams; identifies zone temperature crossovers and pinch points
Reactors	REACTOR EQUILIBRIUM GIBBS CSTR PLUG	Models simultaneous reactions defined by fraction converted Models one reaction defined as an approach to equilibrium temperature or as a fractional approach to chemical equilibrium Simulates a single-phase reactor at minimum Gibbs free energy Simulates a continuously fed, perfectly mixed reactor; adiabatic, isothermal, or constant volume Simulates a tubular reactor exhibiting plug-flow behavior (no axial mixing or heat transfer)
Pumps, compressors, and turbines	PUMP COMPRESSOR EXPANDER VALVE PIPE	Increases P of a stream Compresses the feed stream according to specifications Expands stream to the specified conditions and determines the work produced Simulates the pressure drop Simulates the pressure drop in a pipe

(Source: Seider et al. [9].)

volume in unit time. Before commencing the material balance, the engineer must have some knowledge of the process itself. The sequence of flow from one equipment item (functional unit) to another must be known, and also the reason for the flow and the sequence. The nature of the feed and the product must be known, together with the occurrence of intermediate products and byproducts.

For the flowsheeting program, the design decision that all unit models will be written so that, given the input stream values and the unit parameter values, the output stream values of a unit can be computed. For a mixer, by material balance the flow out of each chemical component is readily obtained, and by heat balance the outlet stream enthalpy can be fixed, since the inlet stream enthalpies must be known. For the reactor, the outputs  $y_{21}$  can be calculated, if the inputs  $x_{21}$ , the functions  $f_2$ , as well as values for the unit parameters  $u_2$  are used, and this applies for the separator

unit as well. Therefore, the unit model equations can be expressed as follows:

$$\left. \begin{array}{l} f_1(x_{11}, x_{12}, y_{11}) = 0 & y_{11} = g_{11}(x_{11}, x_{12}) \\ f_2(x_{21}, y_{21}, u_2) = 0 & y_{21} = g_{21}(x_{21}, u_2) \\ f_3(x_{31}, y_{31}, y_{32}, u_3) = 0 \Rightarrow y_{31} = g_{31}(x_{31}, u_3) & y_{32} = g_{32}(x_{31}, u_3) \\ f_4(x_{41}, x_{42}, y_{41}) = 0 \Rightarrow y_{41} = g_{41}(x_{41}, x_{42}) & \end{array} \right\} \quad (1-3)$$

where each set of rearranged equations, for example,  $y_{11} = g_{11}(x_{11}, x_{12})$ , can be represented by a computer subroutine. Then internal calculation would be assigned as follows:

- |                     |                                                             |
|---------------------|-------------------------------------------------------------|
| Given as input      | $x_{11}, x_{12}$                                            |
| By material balance | Calculate the exiting flow rate of each chemical component. |
| By heat balance     | Calculate the output enthalpy of the exit stream.           |

These exiting flows and enthalpy are the values represented by the stream values  $y_{11}$ .

For designing a sequential modular system, the user must specify all input streams ( $x_{11}$  and  $x_{41}$ ) for the flowsheet, and all recycles will be calculated by iteration of a guessed value. The algorithm for the process flowsheet as illustrated by Westerberg et al. [8] is as follows:

1. Given input stream values for  $x_{11}$ .
2. Guess recycle stream values for  $x_{12}$ .
3. Use model equations  $y_{11} = g_{11}(x_{11}, x_{12})$  to calculate  $y_{11}$ .
4. Equate the stream values for  $x_{21}$  and  $y_{11}$  (using the appropriate connection equations).
5. Use model equations  $y_{21} = g_{21}(x, u_2)$  to calculate  $y_{21}$ .
6. Equate the stream values for  $x_{31}$  and  $y_{21}$ .
7. Use model equations  $y_{31} = g_{31}(x_{31}, u_3)$  and  $y_{32} = g_{32}(x_{31}, u_3)$  to calculate the output stream values for  $y_{31}$  and  $y_{32}$ .
8. Examine the difference in streams values for  $y_{31}$  and  $x_{12}$ , which should be equal, if the problem is solved.
  - (i) If the values are not approximately equal, use the values of  $y_{31}$  and  $x_{12}$  to guess new values for  $x_{12}$  and repeat from step 3.
  - (ii) If the values are approximately equal, the analysis for the recycle has converged and then continue to step 9.
9. Equate stream values for  $x_{12}$  and  $y_{32}$ .
10. Use model equations  $y_{41} = g_{41}(x_{41}, x_{42})$  to calculate  $y_{41}$ .
11. Exit.

The most commonly used architectures for the sequential modular approach are Aspen Plus, Pro II, CHEMCAD, and DESIGN II for Windows. Some packages along with the website address at which they can be accessed are as follows.

ASPEN PLUS, SPEED UP, DYNAPLUS, SPLIT, ADVENT, ADSIM, HYSYS ([www.aspentechn.com](http://www.aspentechn.com))  
 PRO/II, PROVISION, PROTASS, HEXTRAN ([www.simscl.com](http://www.simscl.com))  
 CHEMCAD ([www.chemstations.net](http://www.chemstations.net))  
 DESIGN II for Windows ([www.winsim.com](http://www.winsim.com))

## 1.6 THE EQUATION MODULAR APPROACH

The equation modular approach requires that all modules are written as they are for the sequential modular approach. Each unit model is written in order to calculate the output stream values of a unit, given its input stream values and equipment parameter values. Additionally, a module for each unit type is written, which enables each output value to be related to a linear combination of all input values. The algorithm for this modular type approach is as follows:

- 1 (a) Given the unit inputs,  $x_{i1}, x_{i2}, \dots$   
 Given the equipment parameters  $u_i$   
 Given any initial guesses for iterated variable
- (b) Use the unit model to calculate the unit outputs,  $y_{i1}, y_{i2}, \dots$
- 2 (a) Given the unit inputs  $x_{i1}, x_{i2}, \dots$   
 Given the unit outputs  $y_{i1}, y_{i2}, \dots$
- (b) Find linear relationships, which adequately model the units for small perturbations of the inputs, that is, find the coefficients  $a_{ijk}$  such that

$$y_{ij} \simeq \sum_{k=1}^{k_i} a_{ijk} x_{ik} \quad (1-4)$$

where  $k_i$  is the number of input streams to unit  $i$ . The coefficients  $a_{ijk}$  can vary with the accuracy required, and desired for these linear relationships if they are to behave as realistic, linear models. The remaining equations for the system are the connection equations of the form

$$x_{pq} = y_{st} \quad (1-5)$$

which are also linear in all variables  $x$  and  $y$ . After calculating the  $a_{ijk}$  from step 2(b) above, two sets of linear equations that approximately describe the flowsheet are of the forms as follows.

$$\text{Approximate models: } y_{ij} \simeq \sum_{k=1}^{k_i} a_{ijk} x_{ik}$$

and

$$\text{Connection equations: } x_{pq} = y_{st}$$

The models and connection equations are collected together for the entire flowsheet. These can be solved in principle; however, they contain more variables than there are equations. The exact number of extra variables is readily determined because these equations are identical to the original set, namely

$$\text{Exact model } y_{ij} = g_{ij}(x_{i1}, x_{i2}, \dots, x_{ik}, u_i) \quad (1-6)$$

$$\text{Connection equations: } x_{pq} = y_{st}$$

The model equations involve the same  $x$  and  $y$  variables and only the form has been simplified to obtain a linear relationship. Once the unit parameter variables  $u_i$  are fixed in value, they become the constants in the exact model. They also influence the  $a_{ijk}$  coefficients that are found in the step to fit the linear model to the exact. However, they are not explicitly considered in the linear model. Therefore, in choosing to fix the  $u_i$  values for the exact mode, and devising a calculation sequence in solving the exact model, the same sequence should work for the approximate linear model equations. The preferred architectures of the equation modular approach are SPEEDUP, RTO-OPT, gPROMS, which can be accessed at [www.psenterprise.com](http://www.psenterprise.com).

## 1.7 DEGREES-OF-FREEDOM MODELING

An important aspect of combined material balance and energy balance problems in process modeling is how to ensure that the process equations or sets of modules are determinate, that is, ensuring that the model equations (differential and algebraic) provide a unique relation among all inputs and outputs. Therefore, it is important to determine the number of unknown variables, and consequently must have their values specified in a set of independent equations to which values must be assigned so that the equations can be solved. The number of degrees of freedom is the number of variables in a set of independent equations to which values must be assigned so that the equations can be solved. A degree-of-freedom analysis can be incorporated in the development of a block or model that simulates a process unit. Each block solves a set of  $N_{\text{Equations}}$  involving  $N_{\text{Variables}}$ , where  $N_{\text{Equations}}$  is less than  $N_{\text{Variables}}$ . The number of manipulated variables cannot exceed the number of degrees of freedom, and this is determined using a process model by

$$N_D = N_{\text{Variables}} - N_{\text{Equations}} \quad (1-7)$$

where  $N_D$  is the number of degrees of freedom,  $N_{\text{Variables}}$  is the number of variables and  $N_{\text{Equations}}$  is the number of independent

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equations that describe the process.  $N_{\text{Variables}} - N_{\text{Equations}}$  variables must be specified as long as the  $N_{\text{Equations}}$  are still independent.

Hence a degree-of-freedom analysis classifies modeling problems into three categories:

1.  $N_D = 0$ : exactly determined (exactly specified) process. If  $N_D = 0$ , then the number of equations is equal to the number of process variables, and the set of equations has a unique solution.
2.  $N_D > 0$ : under-determined (under-specified) process. If  $N_D > 0$ , then  $N_{\text{Variables}} > N_{\text{Equations}}$ , and as such, there are more process variables than equations. Consequently, the  $N_{\text{Equations}}$  have an infinite number of solutions since  $N_D$  process variables can be specified arbitrarily. The process model is classified as being undetermined or unspecified.
3.  $N_D < 0$ : over determined (over specified) process. For  $N_D < 0$ , there are fewer process variables than equations and consequently the set of equations has no solution. The process model is classified as being over-determined or over-specified. The only satisfactory category is  $N_D = 0$ . If  $N_D > 0$ , then sufficient inputs have not been identified. Alternatively, if  $N_D < 0$ , then additional independent model equations must be developed.

Both extensive and intensive variables are included in the analysis in contrast with the degrees of freedom obtained from application of the phase rule, which treats only intensive variables. Typical variables are temperature, pressure, either mass (mole) flow rate of each component in a stream or the concentration of each component in a stream or the concentration of each component plus the total flow rate, specific enthalpies, heat flow rate, work (in the energy balance), and recycle ratio.

The number of manipulated variables is generally less than the number of degrees of freedom, since several variables may be defined externally; that is,  $N_D = N_{\text{Manipulated}} + N_{\text{Externally Defined}}$ . Therefore, the number of independent manipulated variables can be expressed in terms of the number of externally defined variables as suggested by Seider et al. [9]

$$N_{\text{Manipulated}} = N_{\text{Variables}} - N_{\text{Externally Defined}} - N_{\text{Equations}} \quad (1-8)$$

The number of independent manipulated variables equals the number of controlled variables that can be regulated. However, whenever a manipulated variable is paired with a regulated output, its degree of freedom is transferred to the output's set point, which becomes the new independent variable.

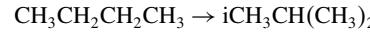
A structured approach to modeling comprises the following: First, state which quantities in the model are known constants or parameters that can be fixed on the basis of equipment dimensions, constant physical properties, and so forth. Second, identify the  $N_{\text{Equations}}$  output variables, those that are obtained through solution of the model differential equations, the algebraic equations, and integration using specified boundary conditions. Third, identify the variables that are specified functions of time, the inputs in the model, as these will be determined by the process environment. For example, the process feed flow rate may be the output flow rate of an upstream process units, or they can be specified by the control system designer to act as manipulated variables in a control strategy. It should be noted that time  $t$  is not one of the  $N_{\text{Variables}}$  process variables because it is neither a process input nor an output.

### 1.8 ISOBUTANE CHEMICALS ( $iC_4H_{10}$ )

Isobutane is mainly used as an alkylating agent to produce different compounds (alkylates) with a high octane number. Isobutane is in high demand as an isobutene precursor for producing oxygenates

such as methyl and ethyl tertiary butyl ethers (MTBE and ETBE). In producing isobutene ( $iC_4H_8$ ), a substantial amount of *n*-butane is isomerized to isobutane ( $iC_4H_{10}$ ), which is further dehydrogenated to isobutene. Figure 1-5 shows the Butamer process, a fixed bed reactor containing a high selective catalyst that promotes the conversion of *n*-butane to isobutane equilibrium mixture. Isobutane is then separated in a deisobutanizer tower. The *n*-butane is recycled with makeup hydrogen.

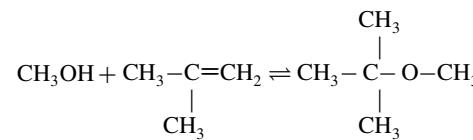
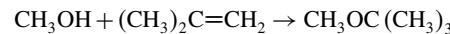
The isomerization reaction occurs at a relatively low temperature.



The Catofin process is currently used to dehydrogenate isobutane to isobutene. Alternatively, isobutane could be thermally cracked to yield predominantly isobutene and propane. Other by-products are fuel gas and  $C_5^+$  liquid. The steam cracking process is made of three sections: a cracking furnace, a vapor recovery section, and a product fractionation section. The dehydrogenation of isobutane to isobutene is represented by

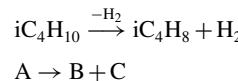


and further reaction of isobutene with methanol produces MTBE, as represented by



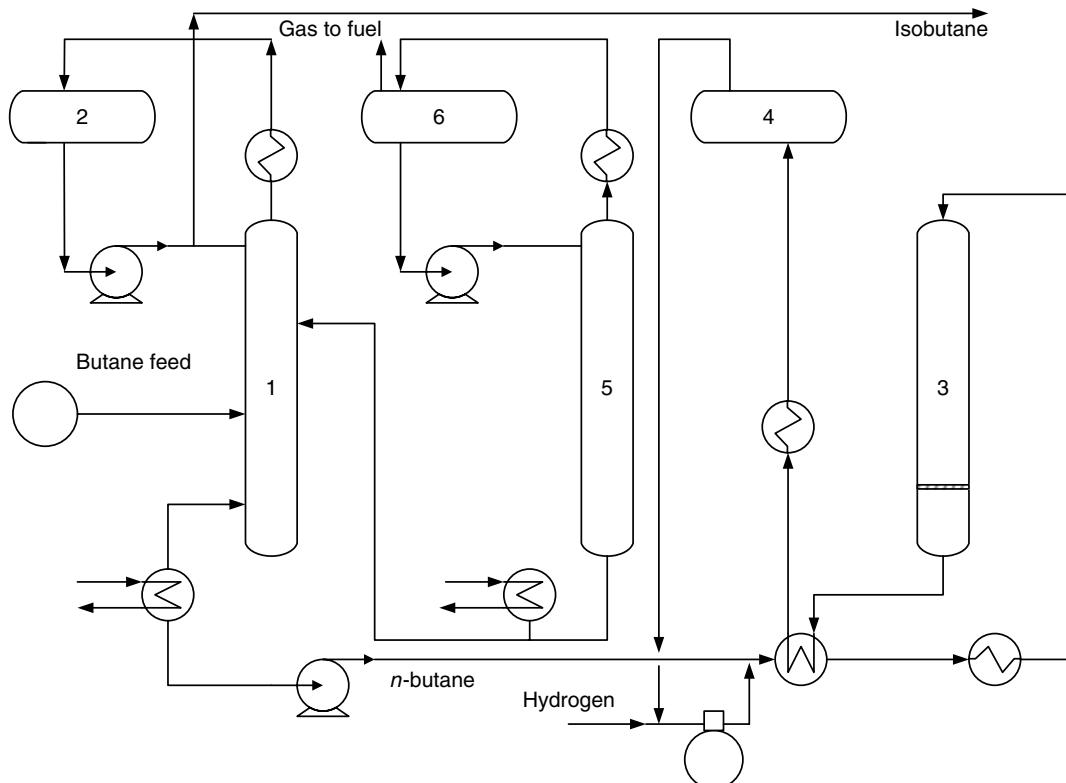
### A CASE STUDY (FELDER AND ROUSSEAU [10])

The gas phase dehydrogenation of isobutane to isobutene

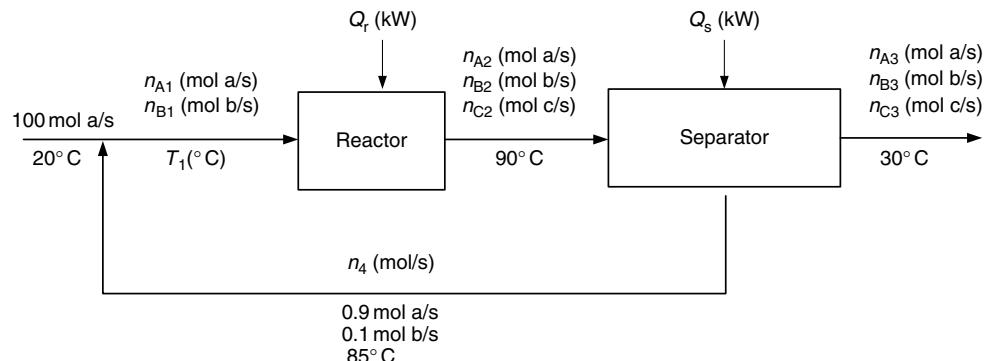


is carried out in a continuous reactor. A stream of pure isobutane (the fresh feed to the process) is mixed adiabatically with a recycle stream containing 90 mole % isobutane and the balance isobutene, and the combine stream goes to a catalytic reactor. The effluent from this process goes through a multistage separation process: one product stream containing all of the hydrogen and 10% of the isobutane leaving the reactor as well as some isobutene is sent to another part of the plant for additional processing, and the other product stream is recycled to the reactor. The single pass isobutane conversion is 35%.

Figure 1-6 shows the process flowchart for a fresh feed of 100 mol isobutane, where A represents isobutane, B denotes isobutene, and C denotes hydrogen. Figure 1-7a is a functional block diagram of the flowsheet, which is being simulated. It contains all the functional modules and their interconnecting streams which are used to model this process. Three modules represent the process: the reactor, the separation process, and the mixing point, which in the process may be a tee in the feed line or a stirred tank. The local degrees of freedom associated with each module is determined, and then the net degree of freedom for the process.



**Figure 1-5** The UOP Butamer process for isomerization of *n*-butane to isobutane: (1, 2) deisobutanizers, (3) reactor, (4) separator (for separation and recycling H<sub>2</sub>), (5, 6) stabilizers.



**Figure 1-6** A simplified process flowsheet for dehydrogenation of isobutane.

#### Mixing Point

4 unknowns ( $\dot{n}_{A1}$ ,  $\dot{n}_{B1}$ ,  $\dot{n}_4$ ,  $T_1$ )

–3 balances (2 material balances, 1 energy balance)

1 local degree of freedom

#### Reactor (Count material balances on molecular species)

7 unknowns ( $\dot{n}_{A1}$ ,  $\dot{n}_{B1}$ ,  $\dot{n}_{A2}$ ,  $\dot{n}_{B2}$ ,  $\dot{n}_{C2}$ ,  $T_1$ ,  $\dot{Q}_r$ )

–4 balances (3 molecular species balances, 1 energy balance)

–1 additional relation (35% single-pass conversion)

+1 chemical reaction

3 local degrees of freedom

#### Separator

8 unknowns ( $\dot{n}_{A2}$ ,  $\dot{n}_{B2}$ ,  $\dot{n}_{C2}$ ,  $\dot{n}_{A3}$ ,  $\dot{n}_{B3}$ ,  $\dot{n}_{C3}$ ,  $\dot{n}_4$ ,  $\dot{Q}_s$ )

–4 balances (3 material balances, 1 energy balance)

–1 additional relation (isobutane split)

3 local degrees of freedom

#### Overall Process

7 local degrees of freedom (1 + 3 + 3)

–7 ties ( $\dot{n}_{A1}$ ,  $\dot{n}_{B1}$ ,  $\dot{n}_{A2}$ ,  $\dot{n}_{B2}$ ,  $\dot{n}_{C2}$ ,  $\dot{n}_4$  and  $T_1$  were each counted twice)

0 net degrees of freedom

Therefore, the problem can be solved for all given variables.

In observing the flowchart, the solution strategy in solving the units involves a trial and error. Taking a unit-to-unit procedure, the mixing unit has one local degree of freedom and four unknowns;

## 12 PROCESS PLANNING, SCHEDULING, AND FLOWSHEET DESIGN

$\dot{n}_{A1}$  and  $\dot{n}_{B1}$  require  $\dot{n}_4$ . This variable requires the separation process balance equation, which in turn depends on the values of  $\dot{n}_{A2}$ ,  $\dot{n}_{B2}$ , and  $\dot{n}_{C2}$ . Calculating these variables requires solving the reactor equations, which invariably depend on knowing  $\dot{n}_{A1}$  and  $\dot{n}_{A2}$ . These variables in turn depend on solving the mixer equations, and thus back to the starting point. The solution in solving this problem is tearing the cycle, and the most appropriate point would be between the separator and the mixing point, as it involves only one variable  $\dot{n}_4$ . The Excel spreadsheet (SEQUENTIAL-SIMULATION.xls) is developed, showing an initially assumed value of the recycle flow rate ( $\dot{n}_{4a}$ ) of 100 mol/s (E13) and an assumed value of the mixing point outlet temperature of 50°C (D8) in Figure 1-7a. The value of  $\dot{n}_{4a}$  will be varied until the calculated recycle flow rate ( $\dot{n}_{4c}$ ) in cell J13 equals the assumed value. This is carried out by finding the value of  $\dot{n}_{4a}$  that drives the value of  $\dot{n}_{4a} - \dot{n}_{4c}$  in cell I17 to 0. Once the flow rates are correct, the mixing point temperature is varied to determine the value that drives the enthalpy of reaction  $\Delta\dot{H}_r = \sum \dot{n}_{out} \dot{H}_{out} - \sum \dot{n}_{in} \dot{H}_{in}$  (in D4) to zero for the adiabatic mixer. The calculations were achieved by using the Solver from the Tools menu in the Excel spreadsheet. When the Solver Parameters dialog box appears, the Set Target Cell (\$I\$17) is set equal to zero by Changing Cell (\$E\$13) (Figure 1-7b). When all of the required information has been entered correctly, select Solve. This will initiate the actual solution procedure giving a new dialog box labeled Solver Results, stating whether or not Solver has been able to solve the problem (i.e., whether convergence has been obtained). If a solution has been obtained, a value will appear in the cell originally containing the initial guess (\$E\$13). The cell containing the formula will show

a value that is zero, or is very close to zero. An alternative to Solver is Goal Seek. However both Solver and Goal Seek use different mathematical solution procedures. Hence, the values obtained with Solver may not be identical to those obtained using Goal Seek, though they should be very close.

The specific heat capacity equation for any component can be expressed as follows:

$$C_p = A + B(T) + C(T)^2 + D(T)^3, \text{ J/mol K}$$

where  $A$ ,  $B$ ,  $C$ , and  $D$  are the regression Constants and  $T$  is the adiabatic mixing temperature.

For isobutane ( $i\text{C}_4\text{H}_{10}$ ), the regression constants are

$$A = 0.08946, B = 30.13 \times 10^{-5}, C = -18.91 \times 10^{-8},$$

$$D = 49.87 \times 10^{-12}$$

For isobutene ( $i\text{C}_4\text{H}_8$ ), the regression constants are

$$A = 0.08288, B = 25.64 \times 10^{-5}, C = -17.27 \times 10^{-8},$$

$$D = 50.5 \times 10^{-12}$$

For hydrogen ( $\text{H}_2$ ), the regression constants are

$$A = 0.02884, B = 0.00765 \times 10^{-5}, C = 0.3288 \times 10^{-8},$$

$$D = -0.8698 \times 10^{-12}$$

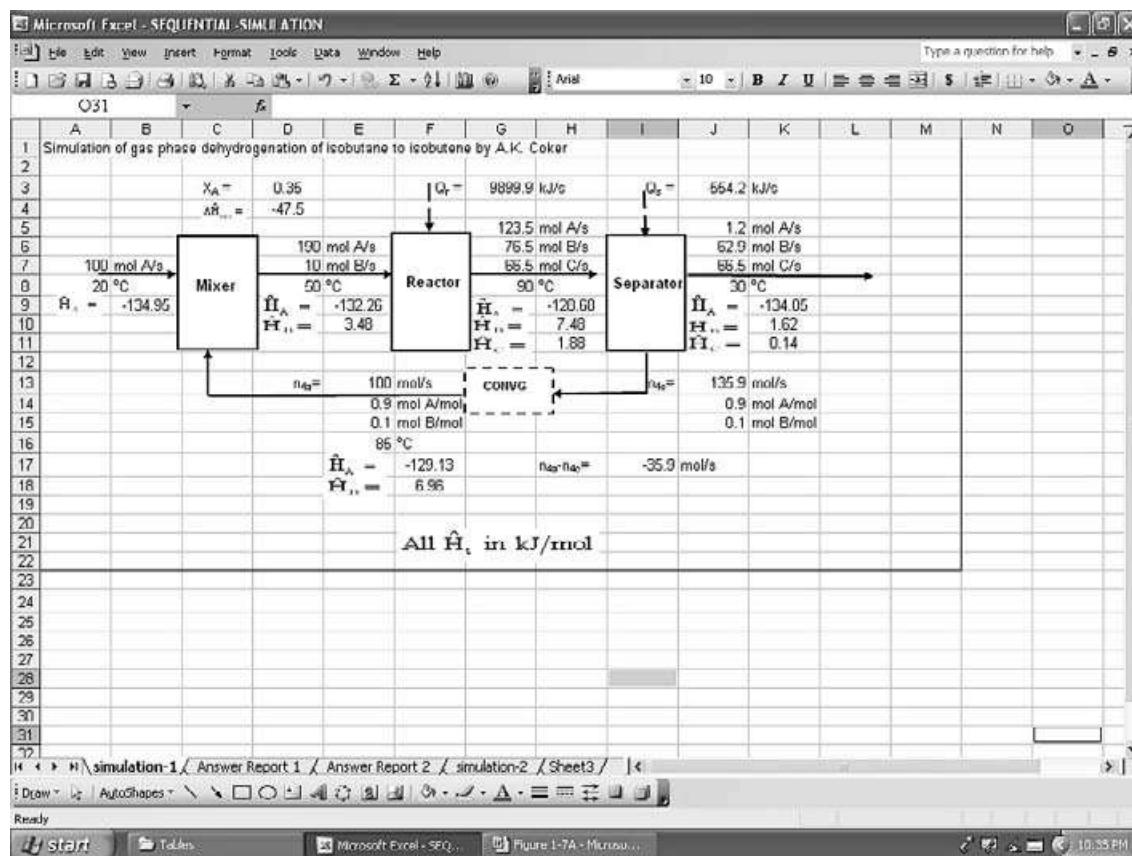
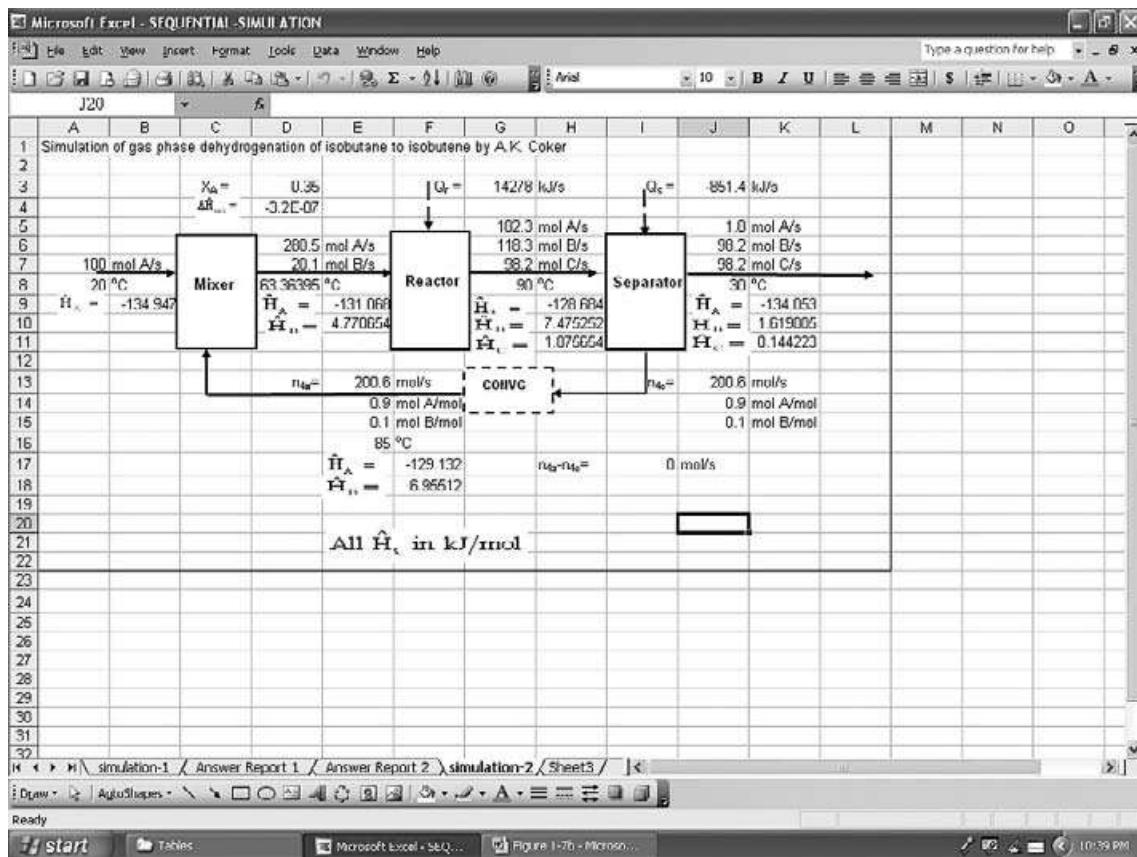


Figure 1-7a Snapshot of Simulation of gas phase dehydrogenation of isobutane before convergence.



**Figure 1-7b** Snapshot of Simulation of gas phase dehydrogenation of isobutane after convergence.

The enthalpy of formation of  $iC_4H_{10}$ ,  $iC_4H_8$ , and  $H_2$  are

$$\Delta H_f^\circ iC_4H_{10} = -134.5 \text{ kJ/kmol}$$

$$\Delta H_f^\circ iC_4H_8 = 1.17 \text{ kJ/kmol}$$

$$\Delta H_f^\circ H_2 = 0.0 \text{ kJ/kmol}$$

The enthalpy of reaction is given by

$$\Delta H_r = \Delta H_f^\circ + \int_{25^\circ\text{C}}^T C_p \, dT \quad (1-9)$$

and integrating between the temperature limits gives

$$\begin{aligned} \Delta H_r = \Delta H_f^\circ &+ A(T - 25) + \frac{B}{2}(T^2 - 25^2) \\ &+ \frac{C}{3}(T^3 - 25^3) + \frac{D}{4}(T^4 - 25^4) \end{aligned} \quad (1-10)$$

Table 1-3 illustrates the spreadsheet formulas used to calculate the material and heat balances of the units. The resulting spreadsheet as indicated by Figure 1-7b gives the correct flowchart for the process. The recycle rate is 200.56 mol/s, and the adiabatic mixing temperature is 63.56° C. The flowchart can be recalculated with any input variable values.

**TABLE 1-3** Excel Spreadsheet Formulas for the Dehydrogenation of Isobutene

- D3 = 0.35 (given fractional conversion of A)
- A7 = 100 (basis of calculation)
- A8 = 20 (given fresh feed temperature)
- G8 = 90 (given reactor effluent temperature)
- J8 = 30 (given product stream temperature)
- E13 = 100 (initial guess for recycle stream flow rate)
- E14 = 0.90, J14 = 0.90 (given mole fraction of A in recycle stream)
- E15 = 0.10, J15 = 0.1 (calculated mole fraction of B in recycle stream)
- E16 = 85 (given temperature of recycle stream)
- D6 = A7 + (E13 \* E14) (A balance on mixing point)

(continued)

**TABLE 1-3—(continued)**


---

D7 = E13*E15 (B balance on mixing point)
D8 = 50 (initial guess for combined stream temperature. Later vary to make $\Delta\hat{H}_{\text{mixer}} = 0$ )
G5 = D6*(1 - D3) (unreacted A)
G6 = D7 + [D6*D3] (B balance on reactor: output = input + generation)
G7 = D6*D3 (C balance on reactor: output = generation)
J5 = 0.01*G5 (1% of A reactor effluent)
J13 = (G5 - J5)/J14 (from A balance on separator)
J6 = G6 - (J13*J15) (B balance on separator)
J7 = G7 (C balance on separator)
B9 = $-134.5 + 0.08946^*(A8 - 25) + 30.13e - 15^*(A8^2 - 25^2)/2 - 18.91e - 8^*$ $(A8^3 - 25^3)/3 + 49.87e - 12^*(A8^4 - 25^4)/4 \left[ = (\Delta\hat{H}_f^\circ)_A + \int_{25^\circ C}^T C_{pA} dT \right]$ specific enthalpy of A at 20° C relative to elements at 25° C
E9 = $-134.5 + 0.08946^*(D8 - 25) + 30.13e - 15^*(D8^2 - 25^2)/2 - 18.91e - 8^*$ $(D8^3 - 25^3)/3 + 49.87e - 12^*(D8^4 - 25^4)/4 \left[ = (\Delta\hat{H}_f^\circ)_A + \int_{25^\circ C}^T C_{pA} dT \right]$
E10 = $1.17 + 0.08288^*(D8 - 25) + 25.64e - 5^*(D8^2 - 25^2)/2 - 17.27e - 8^*$ $(D8^3 - 25^3)/3 + 50.5e - 12^*(D8^4 - 25^4)/4 \left[ = (\Delta\hat{H}_f^\circ)_B + \int_{25^\circ C}^T C_{pB} dT \right]$ specific enthalpies of A and B at 50° C relative to elements at 25° C
H9 = $-134.5 + 0.08946^*(G8 - 25) + 30.13e - 15^*(G8^2 - 25^2)/2 - 18.91e - 8^*$ $(G8^3 - 25^3)/3 + 49.87e - 12^*(G8^4 - 25^4)/4 \left[ = (\Delta\hat{H}_f^\circ)_A + \int_{25^\circ C}^T C_{pA} dT \right]$
H10 = $1.17 + 0.08288^*(G8 - 25) + 25.64e - 5^*(G8^2 - 25^2)/2 - 17.27e - 8^*$ $(G8^3 - 25^3)/3 + 50.5e - 12^*(G8^4 - 25^4)/4 \left[ = (\Delta\hat{H}_f^\circ)_B + \int_{25^\circ C}^T C_{pB} dT \right]$
H11 = $0.0 + 0.02884^*(G8 - 25) + 0.00765^*e - 5^*(G8^2 - 25^2)/2 + 0.3288e - 8^*$ $(G8^3 - 25^3)/3 - 0.8698e - 12^*(G8^4 - 25^4)/4 \left[ = (\Delta\hat{H}_f^\circ)_C + \int_{25^\circ C}^T C_{pC} dT \right]$ specific enthalpies of A, B and C at 90° C relative to elements at 25° C.
K9 = $-134.5 + 0.08946^*(J8 - 25) + 30.13e - 15^*(J8^2 - 25^2)/2 - 18.91e - 8^*$ $(J8^3 - 25^3)/3 + 49.87e - 12^*(J8^4 - 25^4)/4 \left[ = (\Delta\hat{H}_f^\circ)_A + \int_{25^\circ C}^T C_{pA} dT \right]$
K10 = $1.17 + 0.08288^*(J8 - 25) + 25.64e - 5^*(J8^2 - 25^2)/2 - 17.27e - 8^*$ $(J8^3 - 25^3)/3 + 50.5e - 12^*(J8^4 - 25^4)/4 \left[ = (\Delta\hat{H}_f^\circ)_B + \int_{25^\circ C}^T C_{pB} dT \right]$
K11 = $0.0 + 0.02884^*(J8 - 25) + 0.00765^*e - 5^*(J8^2 - 25^2)/2 + 0.3288e - 8^*$ $(J8^3 - 25^3)/3 - 0.8698e - 12^*(J8^4 - 25^4)/4 \left[ = (\Delta\hat{H}_f^\circ)_C + \int_{25^\circ C}^T C_{pC} dT \right]$ specific enthalpies of A, B and C at 30° C relative to elements at 25° C.
F17 = $-134.5 + 0.08946^*(E16 - 25) + 30.13e - 15^*(E16^2 - 25^2)/2 - 18.91e - 8^*$ $(E16^3 - 25^3)/3 + 49.87e - 12^*(E16^4 - 25^4)/4 \left[ = (\Delta\hat{H}_f^\circ)_A + \int_{25^\circ C}^T C_{pA} dT \right]$
F18 = $1.17 + 0.08288^*(E16 - 25) + 25.64e - 5^*(E16^2 - 25^2)/2 - 17.27e - 8^*$ $(E16^3 - 25^3)/3 + 50.5e - 12^*(E16^4 - 25^4)/4 \left[ = (\Delta\hat{H}_f^\circ)_B + \int_{25^\circ C}^T C_{pB} dT \right]$ specific enthalpies of A and B at 85° C relative to elements at 25° C
D4 = $(D6*E9) + (D7*E10) - (A7*B9) - (E13*E14*F17) - (E13*E15*F18)(\Delta\hat{H}_{\text{mixing point}})$
G3 = $(G5*H9) + (G6*H10) + (G7*H11) - (D6*E9) - (D7*E10)(\Delta\hat{H}_{\text{reactor}})$
J3 = $(J5*K9) + (J6*K10) + (J7*K11 + J13*J14*F17) + (J13*J15*F18) - (G5*H9 - G6*H10)$ $- (G7*H11)(\Delta\hat{H}_{\text{separator}})$
I17 = $E13 - J13$ (assumed recycle rate – recalculated recycle rate)

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## 1.9 FLOWSHEETS – TYPES

The flowsheet is the “road-map” of a process, and serves to identify and focus the scope of the process for all interested and associated functions of the project. As a project progresses, the various engineering disciplines read their portions of responsibility from the flowsheet, although they may not understand the process or other details relative to some of the other phases of engineering. Here is where the process and/or project engineer serves to tie together these necessary segments of work. This often involves explanations sufficiently clear to enable these other groups to obtain a good picture of the objective and the problems associated with attaining it.

The flowsheet also describes the process to management as well as those concerned with preparing economic studies for process evaluation.

A good process flowsheet pictorially and graphically identifies the chemical process steps in proper sequence. It is done in such a manner and with sufficient detail to present to others a proper mechanical interpretation of the chemical requirements.

There are several types of flowsheets:

### 1. Block Diagram (Figure 1-8)

This is usually used to set forth a preliminary or basic processing concept without details. The blocks do not describe how a given step will be achieved, but rather what is to be done. These are often used in survey studies to management, research summaries, process proposals for “packaged” steps, and to “talk-out” a processing idea. For management presentations, Figures 1-9, 1-10, 1-11a and b are pictorial and help illustrate the basic flow cycle.

### 2. Process Flowsheet or Flow Diagram (Figures 1-12a and b)

This is used to present the heat balance and material balance of a process. This may be in broad block form with specific key points delineated, or in more detailed form identifying essentially every flow, temperature, and pressure for each basic piece of process equipment or processing step. This may and usually does include auxiliary services to the process, such as steam,

water, air, fuel gas, refrigeration, circulating oil, and so on. A typical process flow diagram for the production of MTBE can be viewed in <http://books.elsevier.com/companions/9780750677660>. This type of sheet is not necessarily distributed to the same groups as in the case of piping flowsheet described next, because it may contain detailed confidential process data.

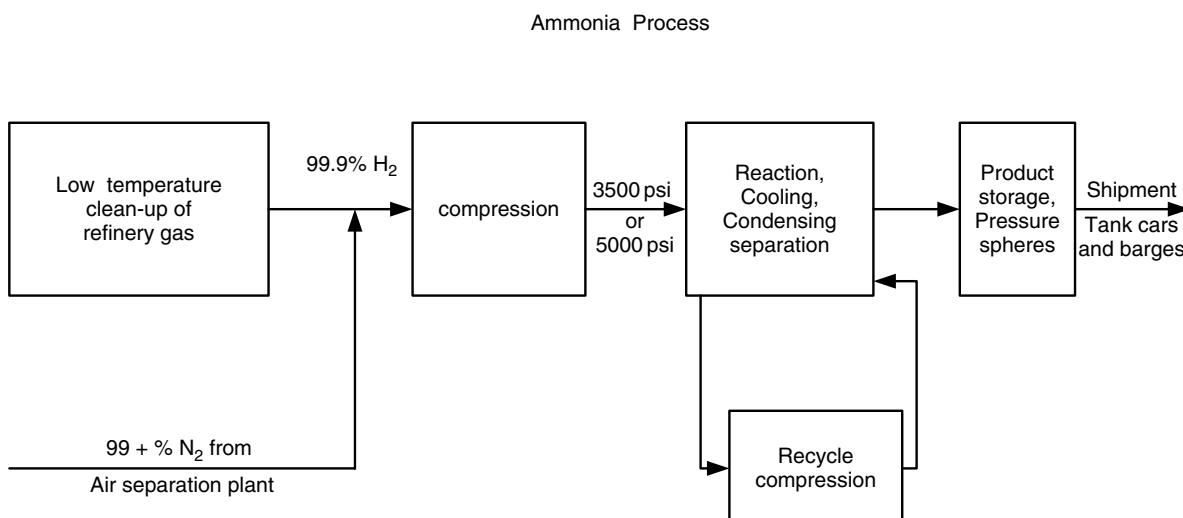
### 3. Piping Flowsheet or Mechanical Flow Diagram (Figures 1-13–1-15) or Piping and Instrumentation Diagram (P&ID)

This is used to present “mechanical-type” details to piping and mechanical vessel designers, electrical engineers, instrument engineers, and other engineers not directly in need of process details. Piping and Instrument Diagrams are graphical summary of the actual hardware elements in a chemical process plant and their inter-relationships of connections to form an operable, safe, and reliable plant. The P&IDs include vessels (columns and tanks), pipe sizes, schedule (thickness), materials of construction, all valves (sizes and types), pumps, heat exchangers, reactors, furnaces, compressors, expanders, relief and drain valves, traps, filters, conveyors, hoppers, purchased subsystems, sensors, insulation requirements (thickness and type), controllers (flow, pressure, temperature, level), spares, and other manufactured items, all in a logical configuration. The P&IDs do not include piping lengths and bends. In some engineering systems, detailed specifications cannot be completed until this flowsheet is basically complete.

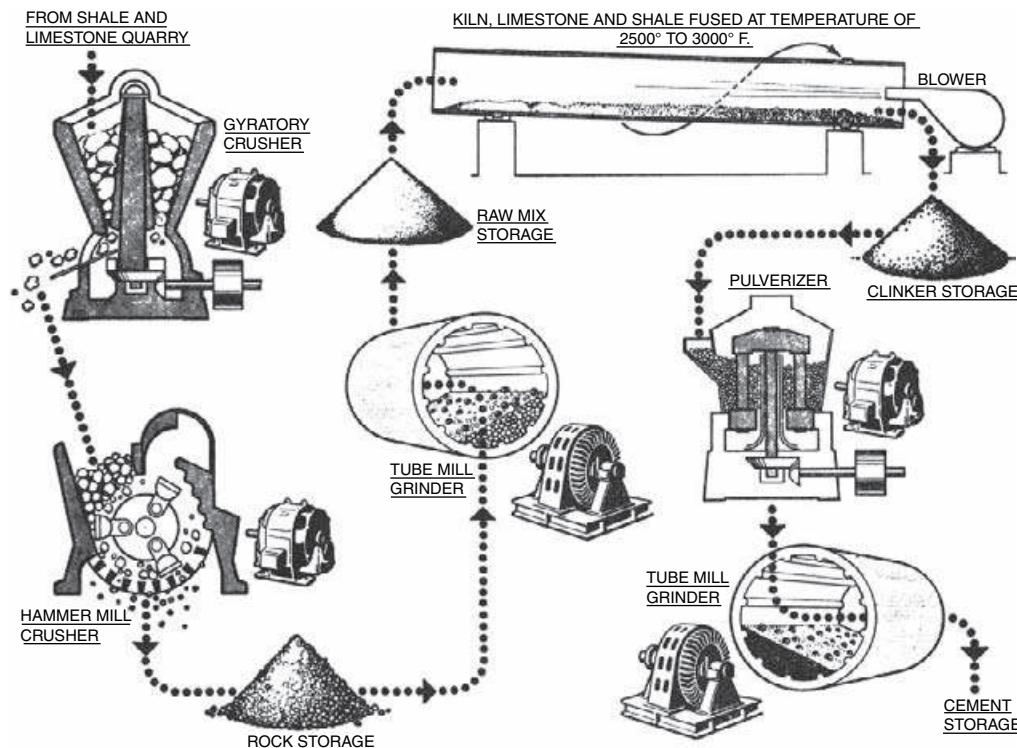
### 4. Combined Process and Piping Flowsheet or Diagram (Figures 1-16 and 1-17)

This is used to serve the combined purpose of both the process and the piping flowsheets. This necessarily results in a drawing with considerably more detail than either of types 2 and 3 just discussed. However, the advantage is in concentrating the complete data and information for a project at one point. It does require close attention in proper reading and often opens data to larger groups of persons who might misinterpret or misuse it.

Some companies do not allow the use of this sheet in their work primarily because of the confidential nature of some of the process data. Where it is used, it presents a concise summary of the complete process and key mechanical data for assembly. This



**Figure 1-8** Block flow diagram.



**Figure 1-9** Pictorial flow diagram establishes key processing steps: Cement manufacture. (By permission from E-M Synchronizer, Electric Machinery Mfg. Co.)

type of sheet requires more time for complete preparation, but like all engineering developments preliminary issues are made as information is available. Often the sheet is not complete until the piping and other detailed drawings are finished. This then is an excellent record of the process as well as a worksheet for training operators of the plant.

#### 5. Utility Flowsheets or Diagrams (Figures 1-18 and 1-19)

Utility line diagram (ULD) includes hardware details of the steam, water piping, and control systems. Used to summarize and detail the interrelationship of utilities such as air, water (various types), steam (various types), heat transfer mediums such as Dowtherm, process vents and purges, safety relief blow-down, and so on to the basic process. The amount of detail is often too great to combine on other sheets, so separate sheets are prepared. These are quite valuable and time saving during the engineering of the project. They also identify the exact flow direction and sequence of tie-in relationships for the operating and maintenance personnel. The distribution of a utility such as steam is by a common distribution pipe, with each unit requiring the utility drawing its supply from that pipe. When a unit has taken its requirement, the utility distribution pipe can then be reduced in size as it continues to the next unit requiring the utility. The order of servicing units is affected by layout, and utility line sizes are affected by the order.

#### 6. Special Flowsheets or Diagrams

From the basic process containing flowsheet, other engineering specialties develop their own details. For example, the instrument engineer often takes the requirements of the process and prepares

a completely detailed flowsheet which defines every action of the instruments, control valves, switches, alarm horns, signal lights, and so on. This is his/her detailed working tool.

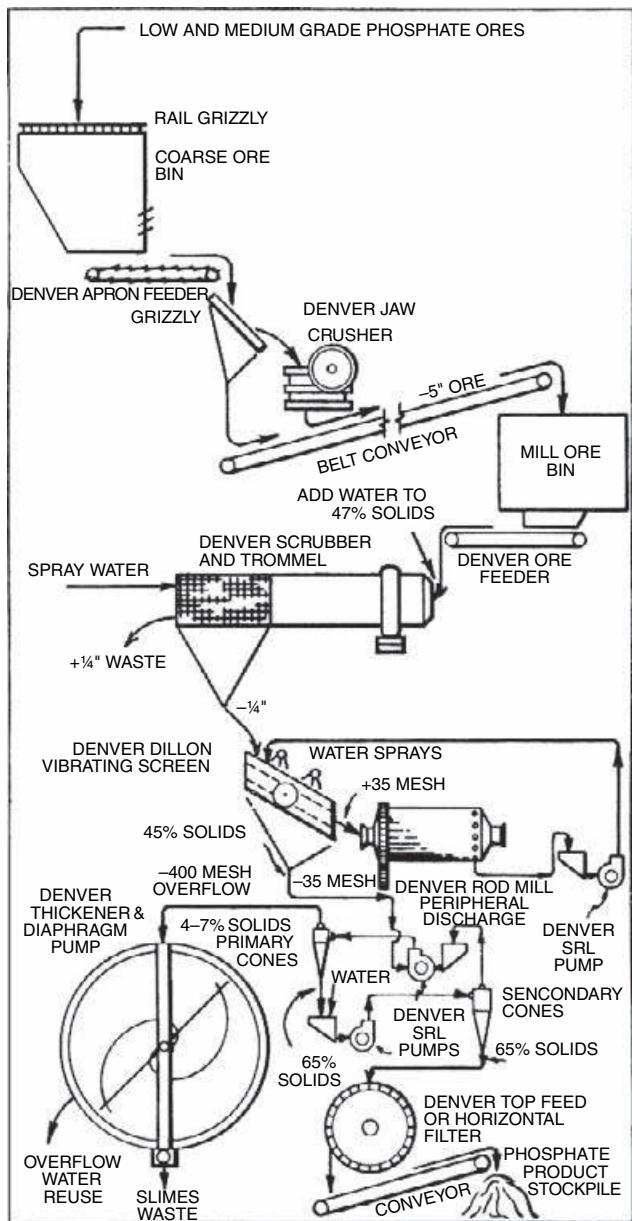
The electrical engineer likewise takes basic process and plant layout requirements and translates them into details for the entire electrical performance of the plant. This will include the electrical requirements of the instrumentation in many cases, but if not, they must be coordinated. O'Donnell [11] has described the engineering aspects of these special flowsheets.

#### 7. Special or Supplemental Aids – Plot Plans (Figure 1-20)

Plot plans are necessary for the proper development of a final and finished process, piping, or utility flowsheet. After broad or overall layout decisions are made, the detailed layout of each processing area is not only helpful but necessary in determining the first realistic estimate of the routing, lengths, and sequence of piping. This is important in such specifications as pipe sizing, and pump head and compressor discharge pressures. The nature of the fluids – whether hazardous, toxic, and so on – as well as the direction or location or availability for entrance to the area, definitely influences decisions regarding the equipment layout on the ground, in the structures, and in relation to buildings. Prevailing wind direction and any other unusual conditions should also be considered.

### 1.10 FLOWSHEET PRESENTATION

Experienced flowsheet layout personnel all emphasize the importance of breaking processes into systems and logical parts of



**Figure 1-10** Pictorial sections flow diagram for principle operations: Phosphate recovery. (By permission from Deco Trefoil, 1958 Denver Equipment Co.)

systems such as reaction, compression, separating, finishing, refrigeration, storage, and so on for detailed drafting. This point cannot be overemphasized, since considerably more space is needed for final completion of all details than is usually visualized at first. The initial layout of the key equipment should be spread farther than looks good to the eye. In fact, it probably looks wasteful of drawing space.

Later as process and sometimes service lines, valves, controls, and miscellaneous small accessories are added this "extra" space will be needed to maintain an easily readable sheet. As this develops, attention should be given to the relative weights and styles of lines to aid in the readability of the sheets.

## 1.11 GENERAL ARRANGEMENTS GUIDE

Each phase of the process is best represented on individual flowsheets. Electric power, fuel gas, drainage, and the many other auxiliary system requirements are also best defined by separate individual flowsheets. These should be complete including all headers, branch takeoffs, tie-ins to existing or known points, and so on. Only in this way can all the decisions as well as specifications be delineated for the various parts contributing to the entire project. The master process or mechanical flowsheet must contain specific references to the other sheets for continuation of the details and complete coordination.

Flowsheet size may vary depending upon the references of the individuals using them. The most popular system uses one size sheet about 24 × 36 in. for all flowsheets. The use of miscellaneous large and small sizes to represent the entire project is often awkward when collected, and increases the possibilities of sheets becoming misplaced. Some groups use sheets from a roll and these are sized to length by systems, becoming 24 × 60 in. and 24 × 72 in. or longer. These are fine for initial study but become tedious to handle on the usual desk. These sheets can be reduced to 11 × 36 in. or 11 × 48 in. both of which are more convenient to work with. These strip-type sheets allow large portions of the process to be grouped together, and are adaptable for folding into reports, and so on.

Since the flowsheet is the primary reference for all engineers working on a project, it must contain all of the decisions, data, flow connections, vents, drains, and so on, which can reasonably be included without becoming confusing and difficult to read.

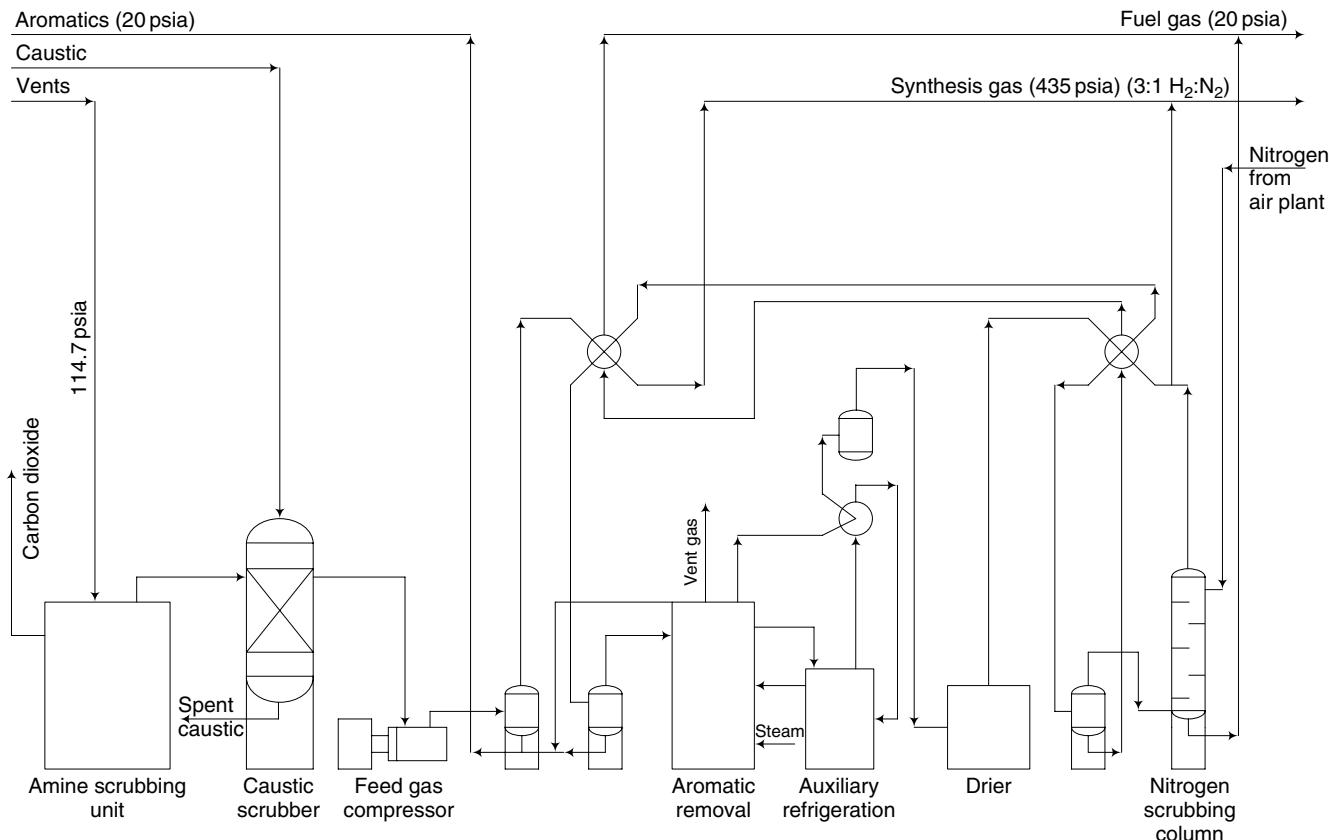
It is important that the various items of equipment and valves be spaced, pictorially represented, and sized as to be easy to read, recognized and followed. On the surface this may sound easy, while in reality it takes an experienced flowsheet detailer to arrange the various items in an eye-pleasing and efficient manner. Suggestive outline figures plus shading often yield the best-looking flowsheet, however, the extra time for detail costs time and money. Some compromise is often indicated. Reference to the various flowsheets illustrated here indicates that the equipment can be arranged by (1) working from a baseline and keeping all heights relative and (2) by placing the various items in a straight-through flow pattern without relative heights. The first scheme is usually preferred for working flowsheets. Whenever possible, all auxiliary as well as spare equipment is shown. This facilitates the full and proper interpretation of all the details.

Figure 1-21 [12] can be used as a guide in establishing relative sizes of equipment as represented on a flowsheet. This chart is based on approximate relative proportions pictured by the mind's eye [12]. For example, the 10-foot diameter × 33-foot high tank would scale to 1.5 in. high. By using the height-developed scale factor, the diameter would be  $(1.5''/33')(10') = 0.45''$  or say 0.5" on the flowsheet.

For some purposes the addition of equipment specification and performance data on the flowsheets adjacent to the item is of value. In many cases though, this additional information makes the sheets difficult to read. The use of equipment summary tables similar to flow and pipe data tables can avoid this objection and yet keep the information on the sheets. Some flowsheets include relief valve set pressures adjacent to the valves, volume capacities of storage tanks, and so on.

## 1.12 COMPUTER-AIDED FLOWSHEET DESIGN/DRAFTING

As the fundamental documents (apart from the process flow diagrams) of any sound chemical/process engineering designs, P&IDs are arguably, in terms of effort, the most demanding,



**Figure 1-11a** Typical flow scheme for separation and purification of vent streams.

and current technology allows the use of computer programs and databases to construct an accurate and detailed flowsheet. This may be a process type diagram or a piping and mechanical/instrument diagram, depending on the input, electrical structure, and so on.

Since the advent of computer-aided design (CAD), most notably represented by AutoCAD, several software packages have been more recently introduced for piping, structural, mechanical, and process drawings. These software packages conform to the standard techniques of chemical process industries that are common to most process designs; for example, the use of universally recognizable symbols for pumps, compressors, vessels and tanks, piping, and valves, and the acceptance of Instrument Society of America (ISA) standard symbols to describe instrumentation. These software packages (e.g., VISIO, CADPIPE, AutoPLANT, and PROCEDE) provide users via “pull-down” menus extensive symbol libraries and provide lists and graphics of the components that are required to produce P&IDs, and piping isometrics. Viola [13] has reviewed three of these software packages, and Table 1-4 shows the key features of these packages (see Figures 1-14, 1-16, 1-22a and b).

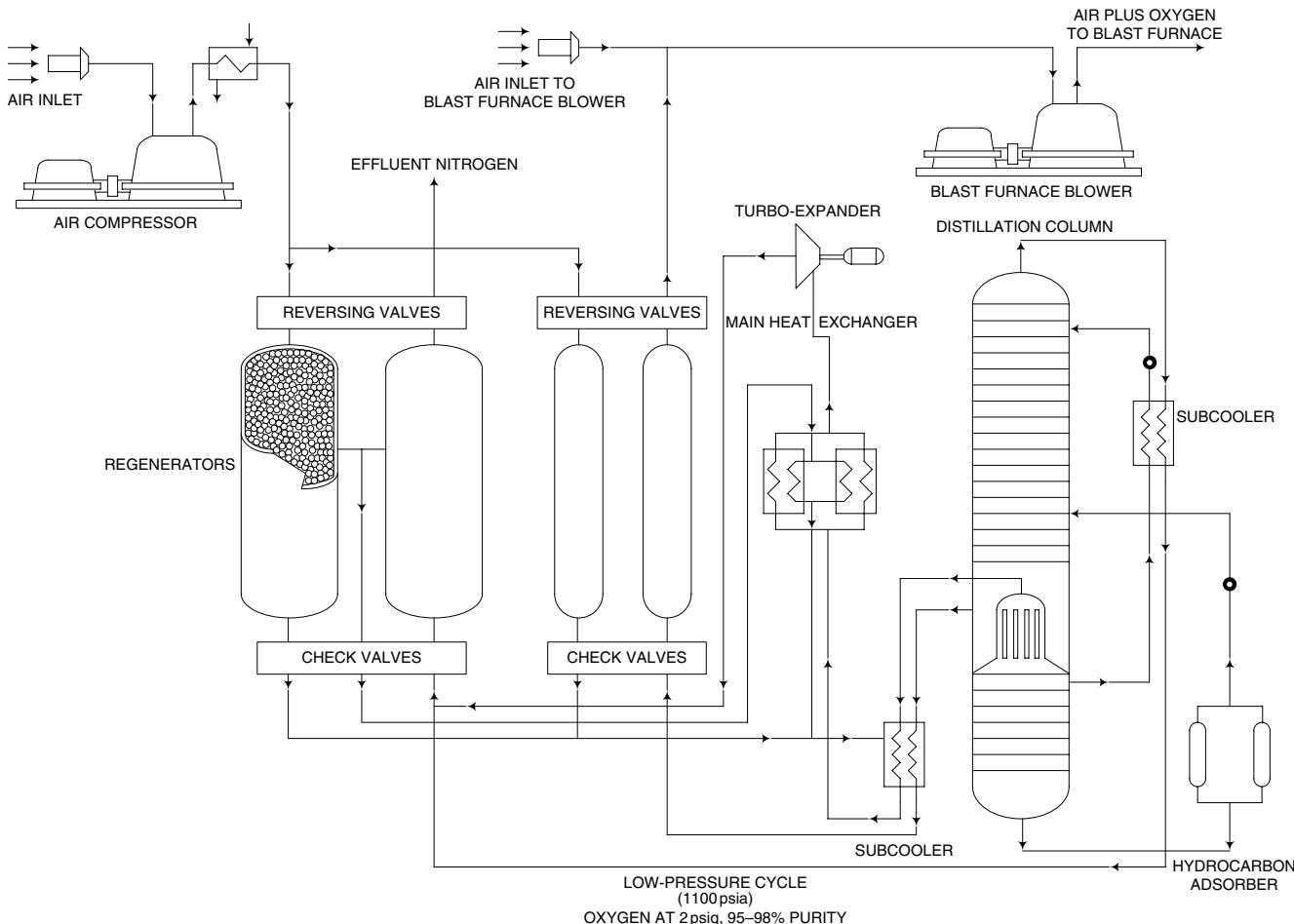
Recently, intelligent P&IDs are being incorporated in plant operations, whereas in the past P&IDs were hard-copy drawings that were CAD products. The latest generation of intelligent P&IDs uses software that incorporates commercial databases, thus allowing ease of integration to other applications such as calculation packages, inclusion of “smart objects”, data sheet information, ease of use, and additional features that surpass CAD deliverables of P&IDs. Novak [14] provides the advantages of incorporating intelligent P&IDs during engineering procurement construction (EPC) phases in terms of reduced capital expense by lowering project costs and improved engineering and design work flow and time to

the market. Intelligent P&IDs software enhances database access by separate design teams worldwide for concurrent engineering; its uses during procurement and construction phases include integrating with material management programs, easy creation of subcontractor packages, and tracking of as-built changes. Table 1-5 illustrates a number of uses of intelligent P&IDs by operations, resulting in significant return on investment (ROI).

### 1.13 OPERATOR TRAINING SIMULATOR SYSTEM

Operators in the process industries employ various process plant simulators to enhance plant performance and safety, and also enable them to learn how their plants will respond to emergency conditions. Additionally, these simulators provide operator training solutions, which can be offered through high fidelity, dynamic- and real-time simulations. Through full functions operator's training stations can be interfaced to the distributed control systems (DCSs), that is, emulated DCS operator stations. Alternatively, the simulator systems can provide full function direct interfaces to actual plant DCS as one solution for the operator/trainee solutions. Some simulators provide emulated operator systems for all major DCS vendors such as ABB/Bailey, Fisher Rosemount, Foxboro, Honeywell, and Yokogawa.

The process plant simulators enable operators to acquire first-hand knowledge of the plant startup, shutdown, emergency procedures, troubleshooting, and so forth. A real-time process plant simulator provides training in basic process equipment, instrumentation and control, distillation operation, pumps, compressors, heat exchangers, and so on. A typical simulator generally includes



**Figure 1-11b** Low pressure cycle used for production of oxygen in steady state conditions. (By permission from Air products and Chemicals Inc.)

several standard process models, and usually one or more custom-developed models that are designed to meet the customer's specifications and requirements. Many of the simulators have features such as the following:

- snapshots
- backtracks
- initial conditions
- multiple modes of operations
- faults and malfunctions
- field-operated devices
- multiple time scales, both slow time and fast time
- instructor variables
- process variable monitors.

Recently, the industry is employing PC-based simulator systems, which will eventually become a standard-based system. The Internet provides many process plant simulation vendors (over 80), and these can be accessed from such search engines as google.com or metacrawler.com with the key words *process plant simulators*. For process operators, the well-known simulators are listed with their websites as follows:

1. Trident Computer Resources Inc. ([www.tridentusa.com](http://www.tridentusa.com))
2. EnVision Systems Inc. ([www.envisionsys.com](http://www.envisionsys.com))
3. Simtronics Corporation ([www.simtronics.com](http://www.simtronics.com))

Figures 1-23, 1-24a, and b show a plan layout of the Autodynamic™ workstation, Autodynamic integrated keyboard, and a photograph of a typical simulation laboratory respectively. Figure 1-25a shows a snapshot of Windows assisted Simulation Advanced Trainer (WSAT) of the binary distillation and Figure 1-25b illustrates the P&ID of a binary distillation (methanol-water).

## 1.14 FLOWSHEET SYMBOLS

To reduce detailed written descriptions on flowsheets, it is usual practice to develop or adopt a set of symbols and codes which suit the purpose. Flowsheet symbol standardization has been developed by various professional and technical organizations for their particular fields. The American National Standard Institute (ANSI) has also adopted most of these symbols. The following symbol references are related and useful for many chemical and mechanical processes:

1. American National Standard Institute (ANSI) ([www.ansi.org](http://www.ansi.org))
2. American Institute of Chemical Engineers (AIChE) ([www.aiche.org](http://www.aiche.org))
  - (a) Letter Symbols for Chemical Engineering, ANSI Y10.12
3. American Society of Mechanical Engineers (ASME) ([www.asme.org](http://www.asme.org))

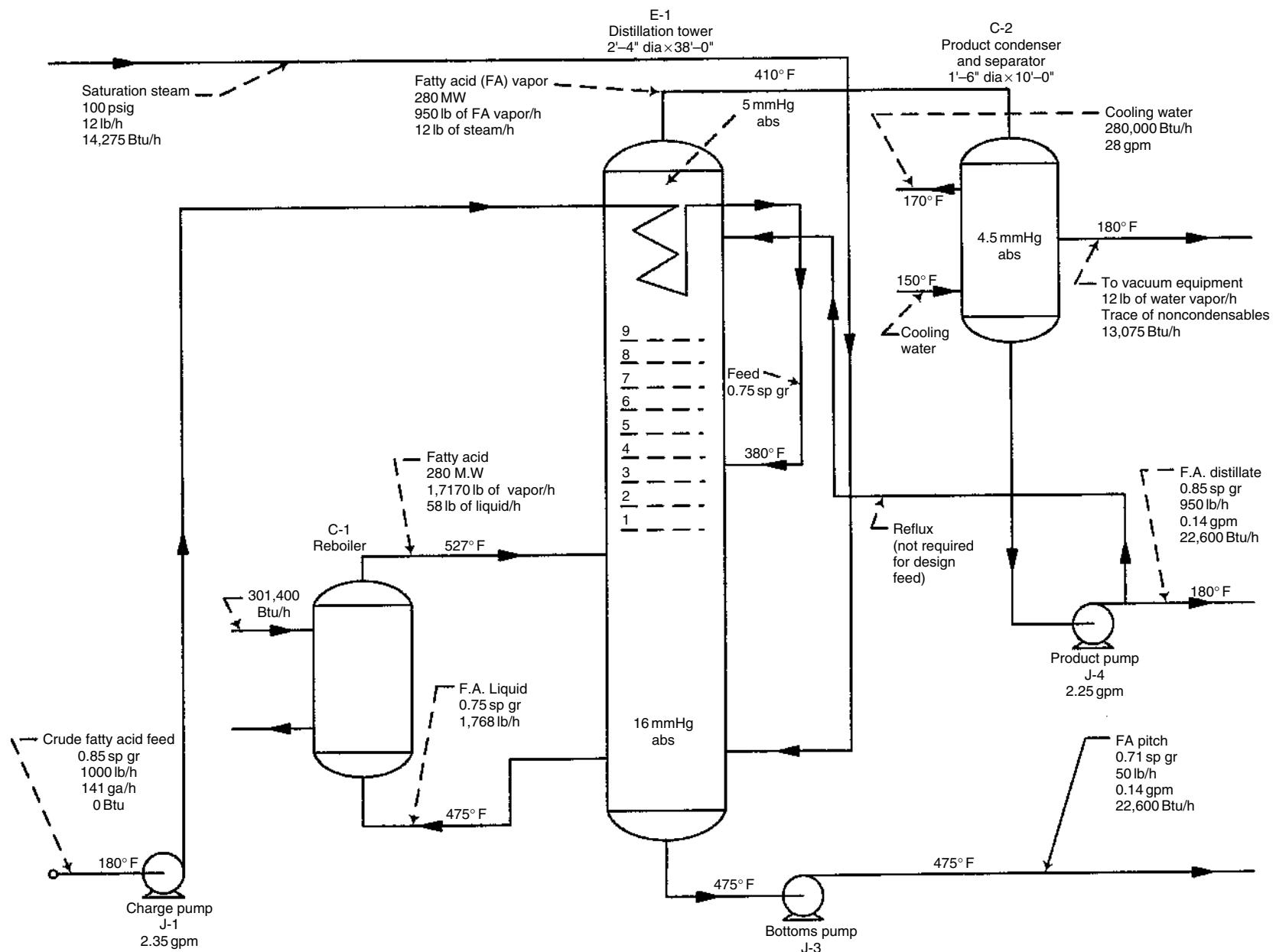
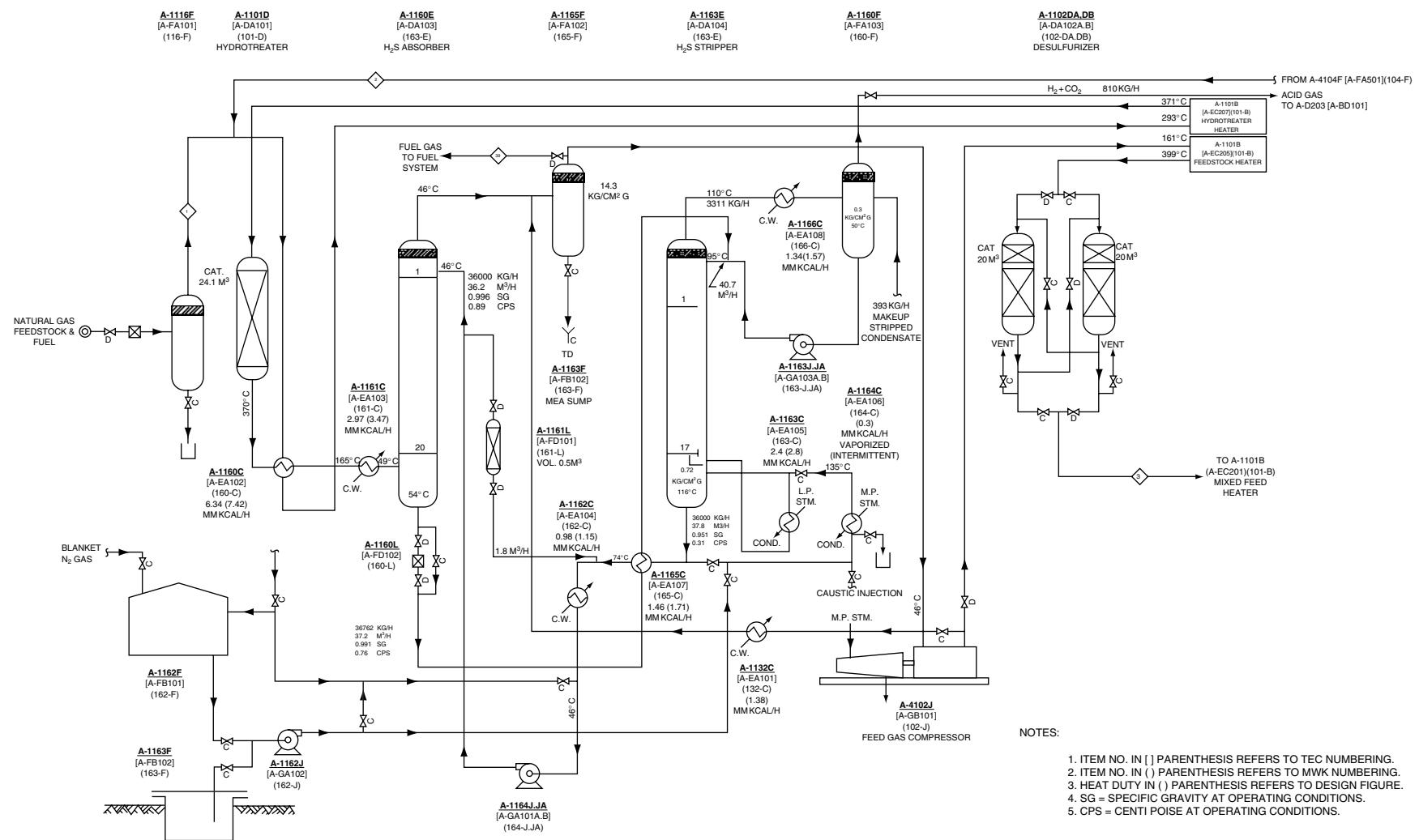


Figure 1-12a Heat and material balance established material and thermal requirements. (By permission, from J.O. O'Donnell [11].)



**Figure 1-12b** Process flow diagram (Feed and fuel desulfurization section).

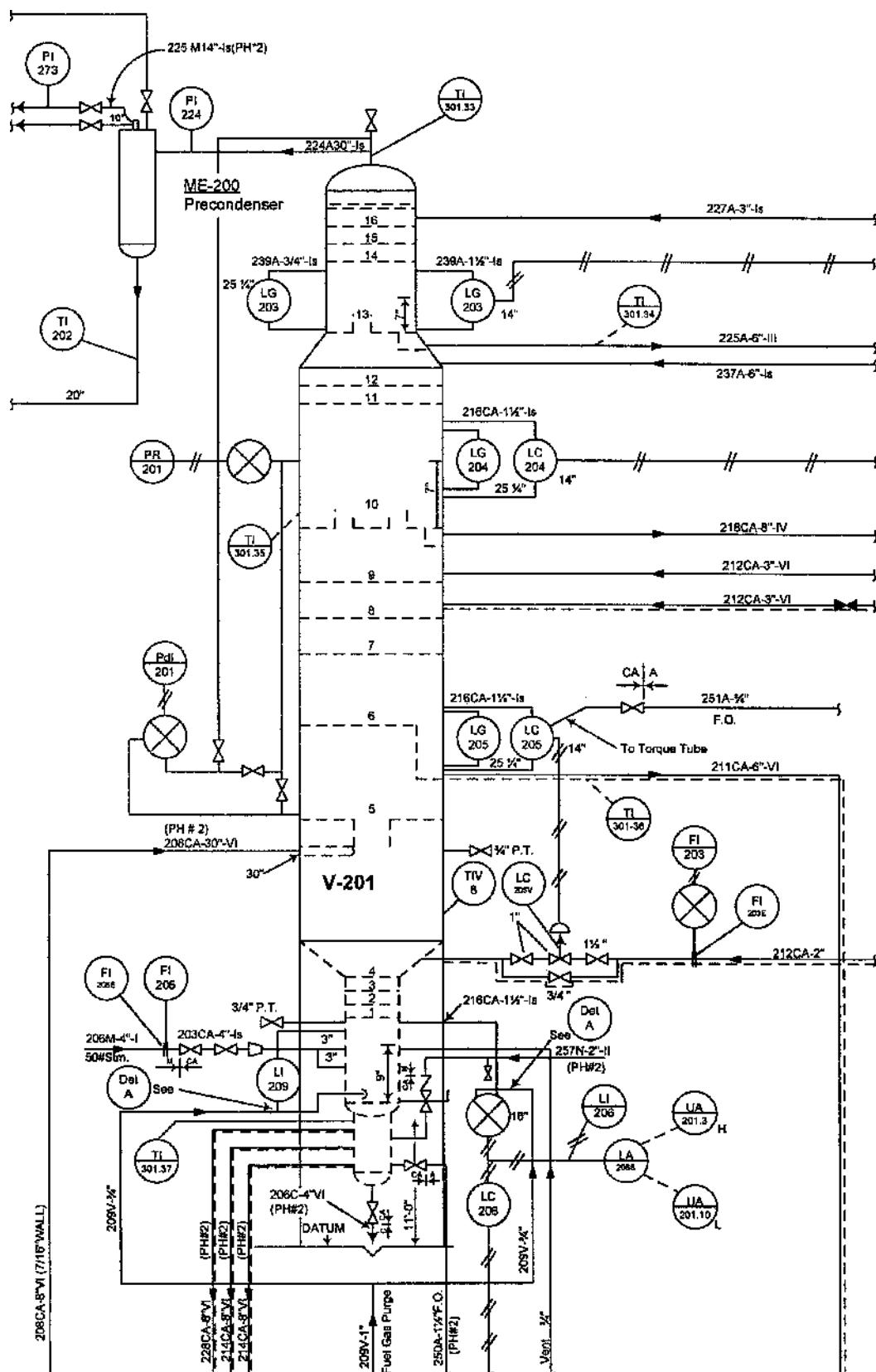
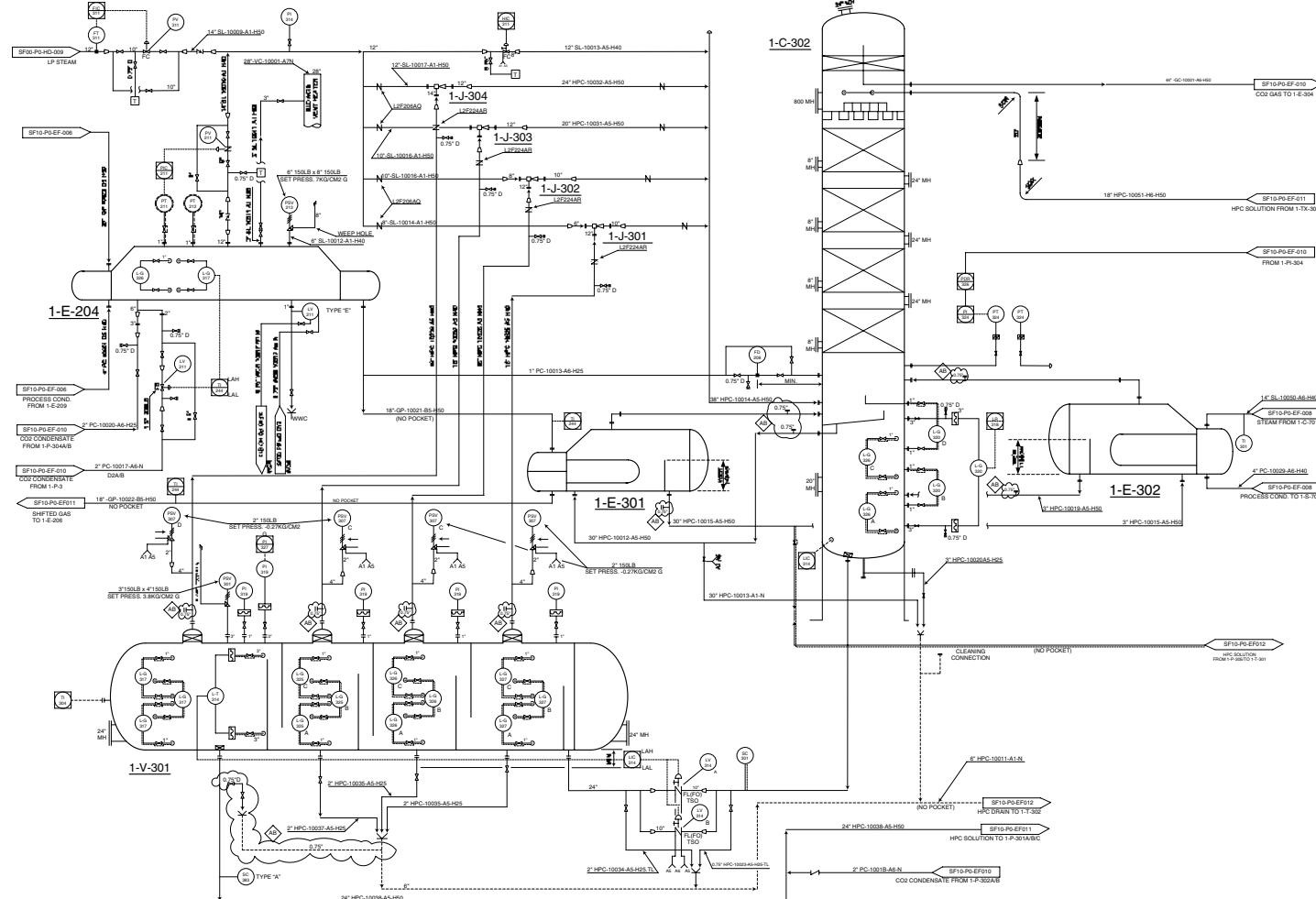


Figure 1-13 Mechanical detail flow diagram. (By permission from Fluor Corp. Ltd.)



ITEM NO.	1-E-204	
SERVICE	LP STEAM GENERATOR	
TYPE	BKM	
H-SURFACE	716 M <sup>2</sup>	
	SHELL	TUBE
DESIGN PRESS.	7 & F.V. KG/CM <sup>2</sup> G	34.5 KG/CM <sup>2</sup> G
DESIGN TEMP.	180 °C	200 °C
INSULATION	50 (HOT) MM	50 (HOT) MM

ITEM NO.	1-E-301	
SERVICE	HPC REBOILER	
TYPE	BKU	
H-SURFACE	472 M <sup>2</sup>	
	SHELL	TUBE
DESIGN PRESS.	3.8 & -0.3 KG/CM <sup>2</sup> G	34.5 KG/CM <sup>2</sup> G
DESIGN TEMP.	164°C	167°C
INSULATION	50 (HOT) MM	50 (HOT) MM

ITEM NO.	1-E-302	
SERVICE	HPC AUX. REBOILER	
TYPE	BKU	
H-SURFACE	553 M <sup>2</sup>	
	SHELL	TUBE
DESIGN PRESS.	3.8 & -0.3 KG/CM <sup>2</sup> G	5.6 & F.V. KG/CM <sup>2</sup> G
DESIGN TEMP.	164 °C	211 (HOT)
INSULATION	50 (HOT) MM	50 (HOT) MM

ITEM NO.	1-J-303
SERVICE	3 <sup>RD</sup> EJECTOR FOR FLASH DRUM
SUC. CAPACITY	(NORMAL-CASE 1) 6.017 KG/HR
DESIGN PRESS.	F. V. & 3.8 KG/CM <sup>2</sup> G
DESIGN TEMP.	260 °C
INSULATION	50 (HOT) MM

SERVICE	4 <sup>TH</sup> EJECTOR FOR FLASH DRUM
SUC. CAPACITY	(NORMAL-CASE) 16,410 KG/HR
DESIGN PRESS.	F.V. & 3.8 KG/CM <sup>2</sup> G
DESIGN TEMP.	260 °C
INSULATION	50 (HOT) MM
ITEM NO.	1-C-302
SERVICE	REGENERATOR
JD-N150-T1	1400-20-65 110V/1PH

DESIGN TEMP.	164 °C
INSULATION	50 (HOT) MM
ITEM NO.	1-V-301
SERVICE	FLASH DRUM
I.D. X TL - TL	4.200 X 10.250 MM
DESIGN PRESS.	3.8 - 0.3 KG/CM <sup>2</sup> G
DESIGN TEMP.	164 °C

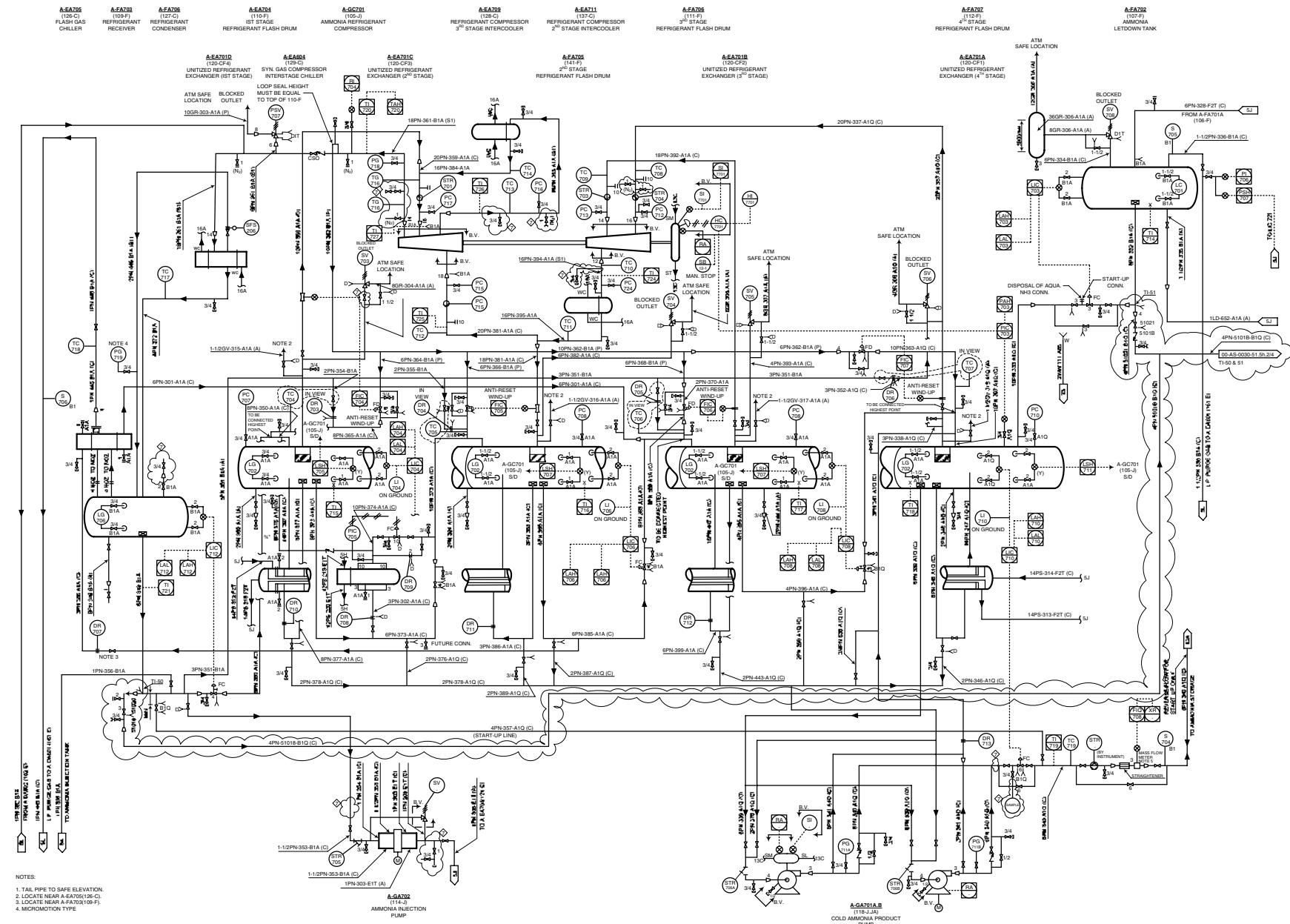
ITEM NO.	1-J-301
SERVICE	IST EJECTOR FOR FLASH DRUM
SUB. CAPACITY	(NORMAL CASE) 5.399 KG/H.R
DESIGN PRESS.	F.V. & 3.8 KG/CM <sup>2</sup> G
DESIGN TEMP.	260 °C
INSULATION	50 (HOT) MM

SUB. CAPACITY	(NORMAL - CASE1) 5,693 KG/H.R
DESIGN PRESS.	F.V. & 3.8 KG/CM <sup>2</sup> G
DESIGN TEMP.	260 °C
INSULATION	50 (HOT) MM

#### **GENERAL NOTE**

1. EACH INSTRUMENT TAG NUMBER SHALL HAVE THE UNIT PREFIX "1".

**Figure 1-14** Piping and instrumentation diagram for ammonia plant CO<sub>2</sub> removal (1/3). To view a larger version of this figure online, please visit <http://books.elsevier.com/companions/9780750677660>.



**Figure 1-15** Piping and instrumentation diagram: Ammonia synthesis and refrigeration unit (2). To view a larger version of this figure online, please visit <http://books.elsevier.com/companions/9780750677660>

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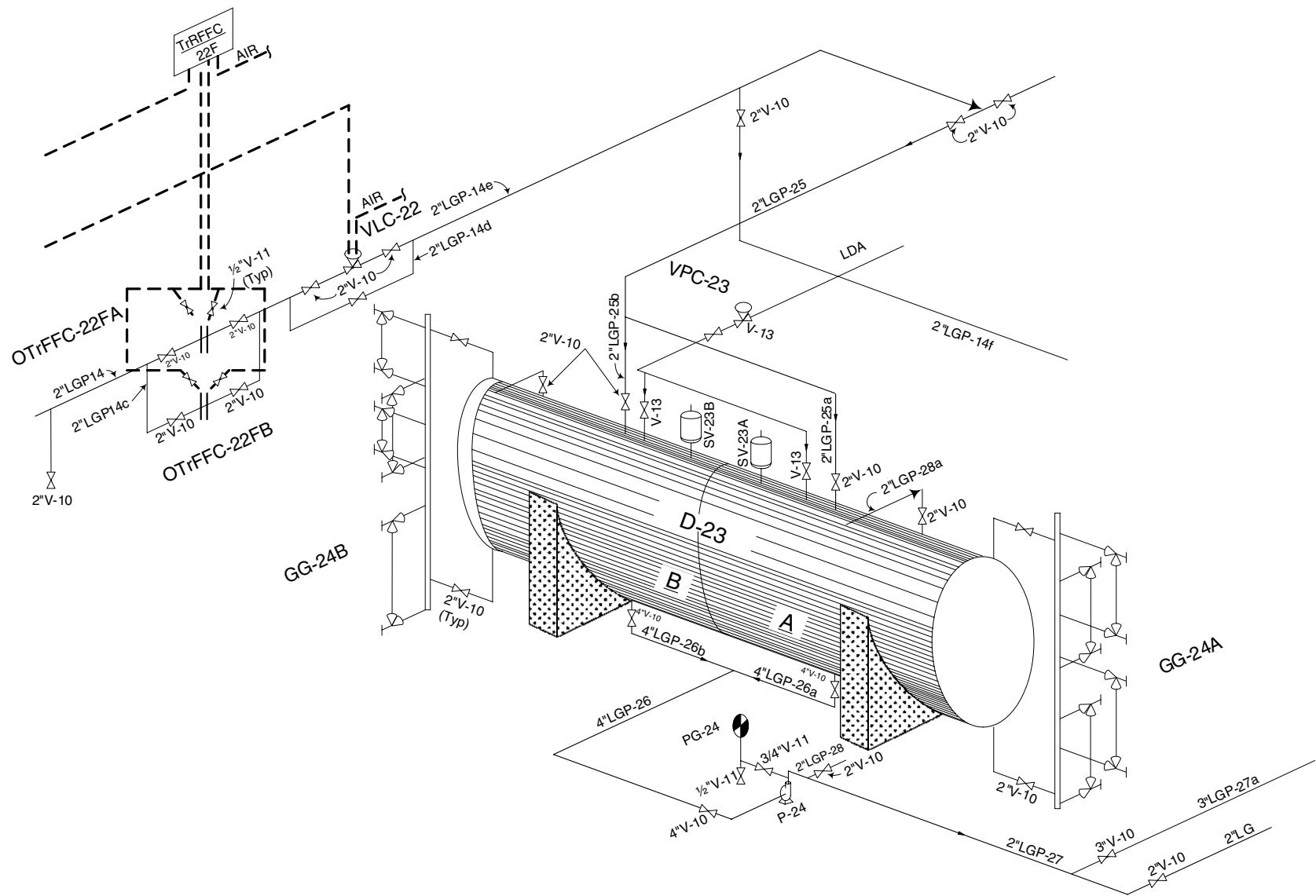
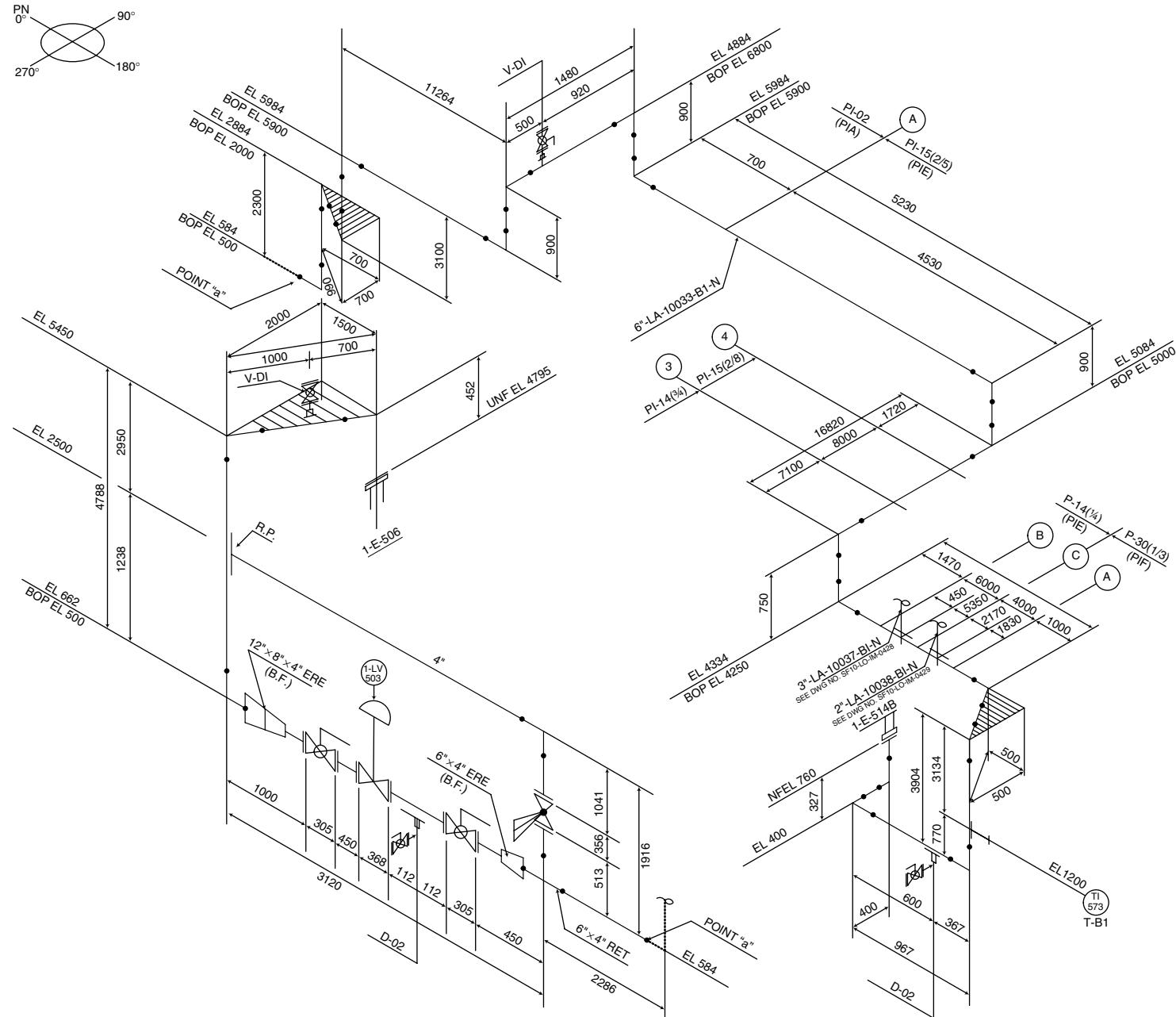


Figure 1-16 Piping details isometric diagram.



**Figure 1-17** Isometric drawing for liquid ammonia line (4LA-10033-B1-N).

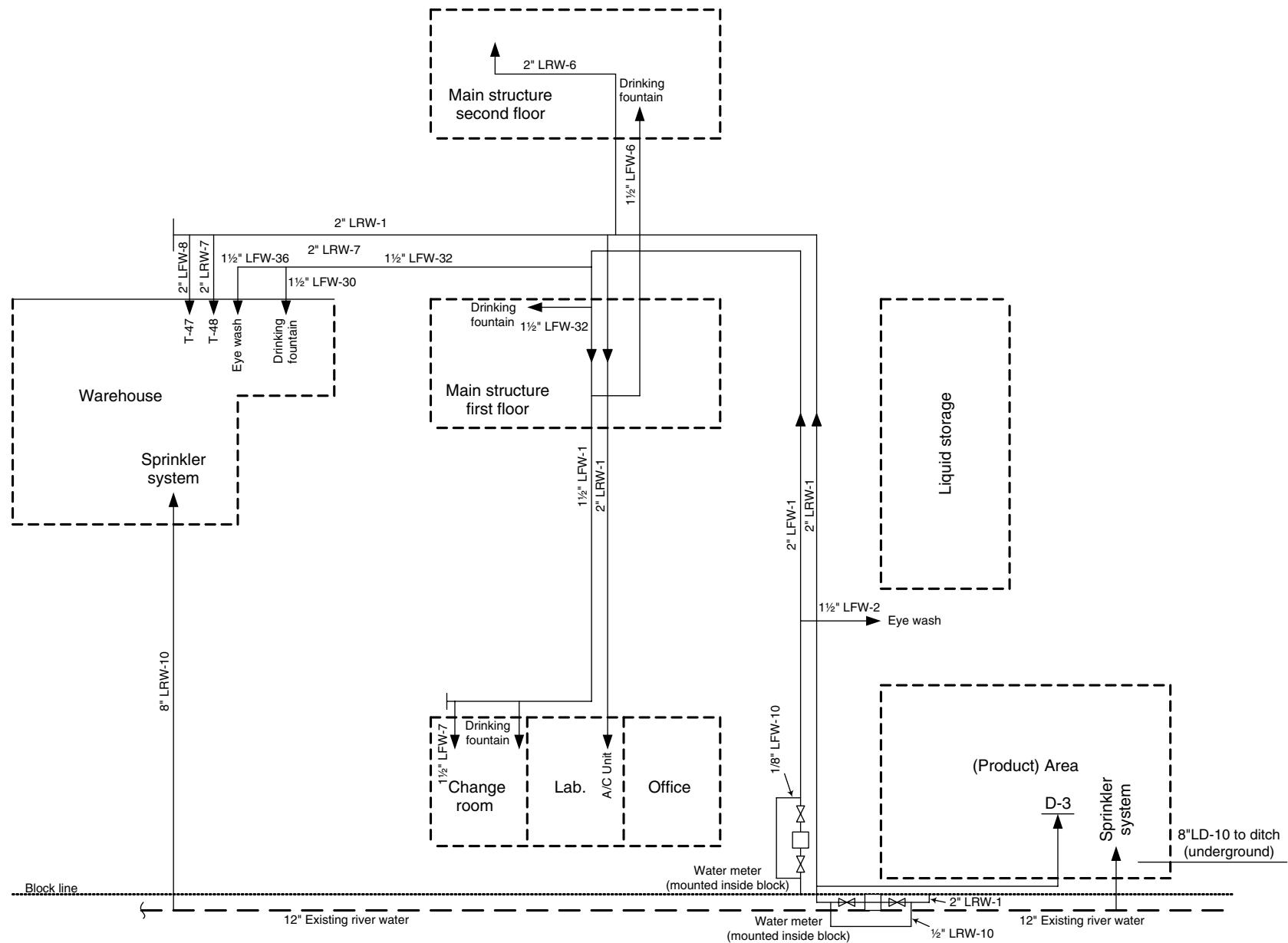
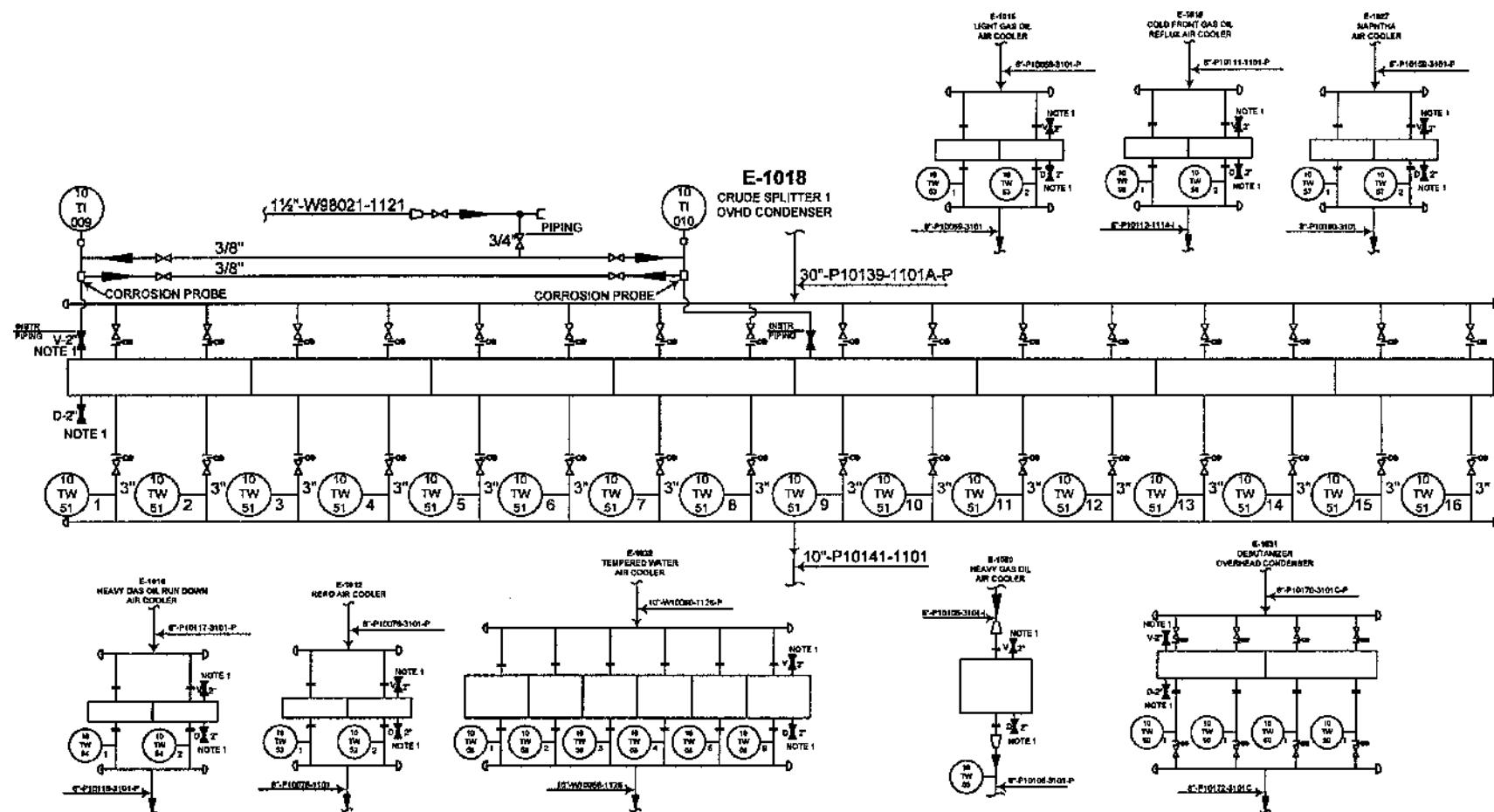


Figure 1-18 Standard type layout for service piping diagram.



ITEM NUMBER	NUMBER OF BUNDLES	NUMBER OF INLET NOZZLES PER BUNDLE	NUMBER OF OUTLET NOZZLES PER BUNDLE	SIZE OF INLET NOZZLES	SIZE OF OUTLET NOZZLES	TOTAL NUMBERS OF INLET NOZZLES	TOTAL NUMBERS OF OUTLET NOZZLES	PHASE INLET L-LIQUID V-VAPOR OR V-L	PHASE OUTLET L-LIQUID V-VAPOR OR V-L	TOTAL NUMBER OF 2" VALVED CONNECTIONS		AIR COOLER TYPE
										VENTS	DRAINS	
E-1012	2	1	1	8" 300' RF	8" 300' RF	2	2	L	L	8 (300' RF)	8 (300' RF)	2FPTA
E-1015	2	1	1	8" 300' RF	8" 300' RF	2	2	L	L	8 (300' RF)	8 (300' RF)	2FPTA
E-1016	2	1	1	8" 300' RF	8" 300' RF	2	2	L	L	8 (300' RF)	8 (300' RF)	2FPTA
E-1018	8	2	2	8" 300' RF	3" 300' RF	16	16	V	L+V	32 (300' RF)	32 (300' RF)	2FPTA
E-1019	2	1	1	8" 300' RF	8" 300' RF	2	2	L	L	8 (300' RF)	8 (300' RF)	2FPTA
E-1020	1	1	1	8" 300' RF	8" 300' RF	1	1	L	L	4 (300' RF)	4 (300' RF)	2FPTA
E-1027	2	1	1	8" 300' RF	8" 300' RF	2	2	L	L	8 (300' RF)	8 (300' RF)	2FPTA
E-1031	2	2	2	8" 300' RF	4" 300' RF	4	4	V	L+V	8 (300' RF)	8 (300' RF)	2FPTA
E-1032	6	1	1	8" 150' RF	6" 150' RF	6	6	L	L	24 (150' RF)	24 (150' RF)	2FPTA

Figure 1-19 Process engineering flow scheme (Crude distiller unit 1000 air cooler manifold).

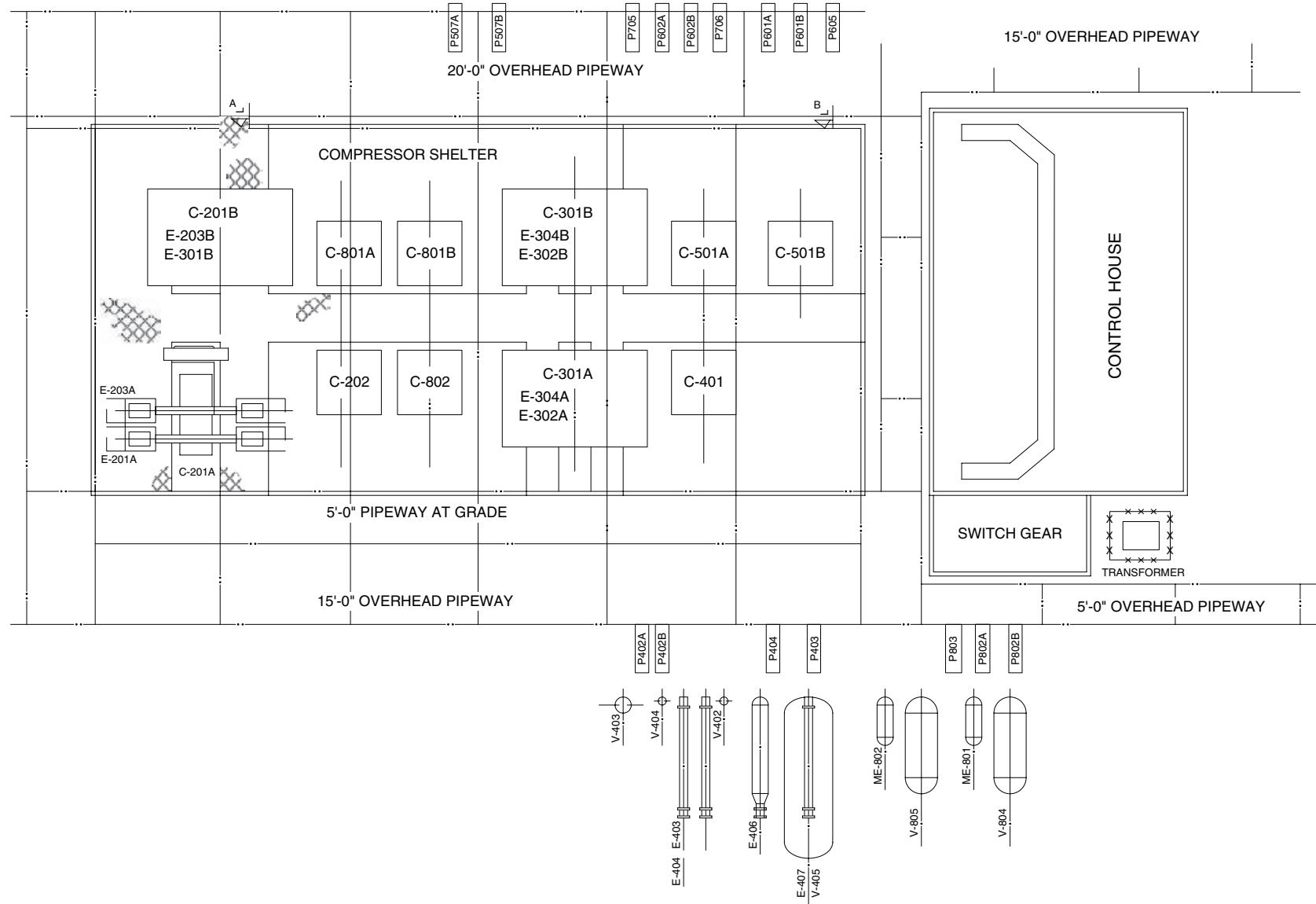
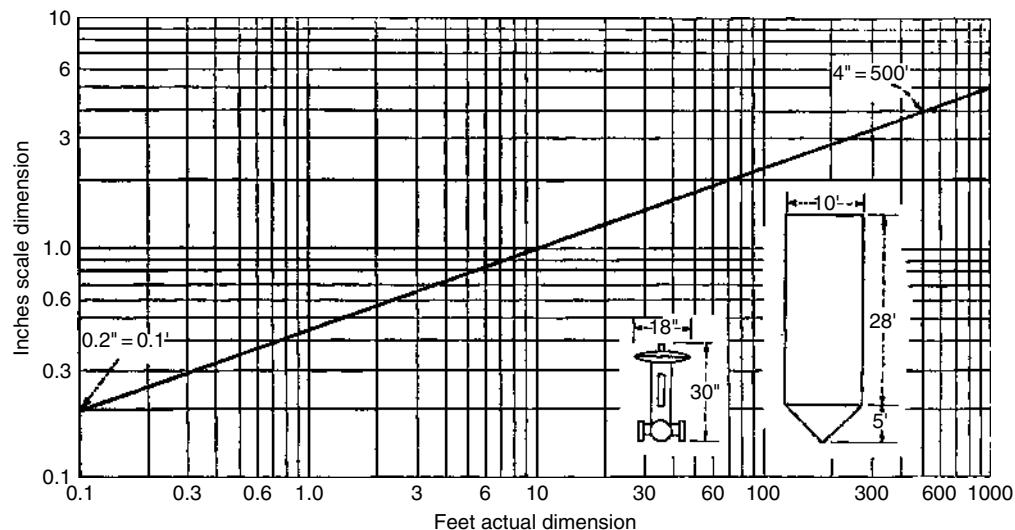


Figure 1-20 Typical process area plot plan and study elevations. (By permission, Fluor Corp. Ltd.)



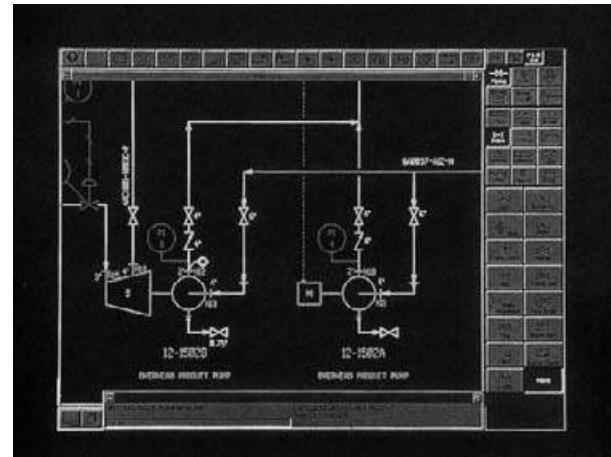
**Figure 1-21** Flowsheet scale reference diagram. (By permission from Berg [12].)

**TABLE 1-4 Three Programs Offering a Mix of Different Features**

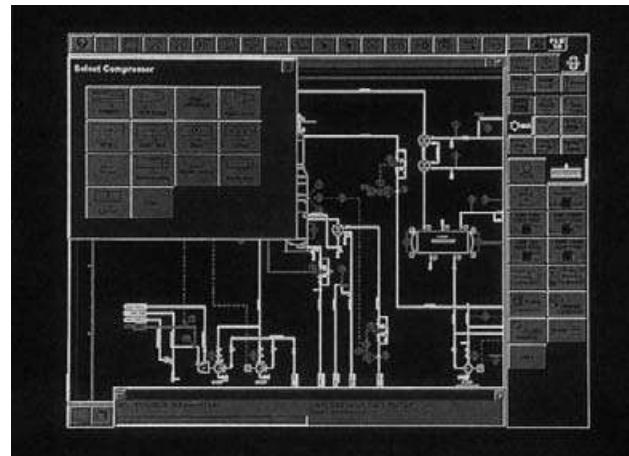
	PROCEDE	AutoPLANT	CADPIPE
AutoCAD required	No	Yes	Yes
Automatic line breaks for symbols	Yes	Yes	Yes
Automatic line clash	Yes	Yes	Yes
Rotate command required for symbols	No	No	Yes
Specification checking available	No	Yes	Yes
Dual automatic tagging	No	Yes	Yes
Data form with secondary tag	No	No	Yes
Automatic signal connect	No	Yes	No

(Source: Viola [13].)

- (a) Graphic Symbols for Plumbing, ANSI and ASA Y32.4
  - (b) Graphic Symbols for Railroads Maps and Profiles, ANSI or ASA Y32.7
  - (c) Graphic Symbols for Fluid Power Diagrams, ANSI or ASA Y32.10
  - (d) Graphic Symbols for Process Flow, ANSI or ASA Y32.11
  - (e) Graphic Symbols for Mechanical and Acoustical Elements as Used in Schematic Diagrams, ANSI or ASA Y32.18
  - (f) Graphic Symbols for Pipe Fittings, Valves, and Piping, ANSI or ASA Z32.2.3
  - (g) Graphic Symbols for Heating, Ventilating, and Air Conditioning, ANSI or ASA Z32.2.4
  - (h) Graphic Symbols for Heat-Power Apparatus, ANSI or ASA Z32.2.6
4. Instrument Society of America (ISA) ([www.isa.org](http://www.isa.org))  
 (a) Instrumentation Symbols and Identification, ISA-S5.1.
5. American Petroleum Institute (API) ([www.api.org](http://www.api.org))
6. British Standards Institute ([www.bsi-global.com](http://www.bsi-global.com))



**Figure 1-22a** Computer-generated P&ID flowsheet. (Courtesy of Intergraph Corp., Bul. DP016A0.)



**Figure 1-22b** Computer-generated instrumentation detail for P&ID flow-sheet. (Courtesy of Intergraph Corp., Bul. DP016A0.)

**TABLE 1-5 Uses of Intelligent P&IDs by Operations**

Provides high integrity of P&ID Data – no need to field verify for decision support	Integration with other operations support software such as reliability solutions. Plant management (PM) or turnaround planning
Provides fast access to accurate, current data to aid troubleshooting	Aids in planning and executing emergency or maintenance procedures
Allows change filtering and recording of updates to meet regulatory compliance	HAZOP analysis for safety checks and audit trails for OSHA, EPA, and other compliance initiatives
Shortened commissioning times	Risk-based inspection and critical system documentation
Data filtration helps evaluate operational costs related to design scenarios	Production adjustments/reanalysis of process for feedstock variances
Quicker startups	Allows multiple locations to view P&ID data

(Source: Novak [14] By permission from Hydroc. Proc.)

Other symbols are established for specialized purposes. The physical equipment symbols established in some of these standards are often not as descriptive as those in the chemical, petrochemical, and petroleum industry are accustomed to using. The bare symbolic outlines given in some of the standards do not adequately illustrate the detail needed to make them useful. Accordingly, many process engineers develop additional detail to include in flowsheets, such as Figures 1-26a–e and 1-27a–c, which enhance the detail in many of these standards. Various types of processing suggest unique, yet understandable symbols which do not fit the generalized forms.

Many symbols are pictorial, which is helpful in representing process as well as control and mechanical operations. In general, experience indicates that the better the representation including relative location of connections, key controls and even utility connections, and service systems, the more useful will be the flowsheets for detailed project engineering and plant design.

To aid in readability by plant management as well as engineering and operating personnel, it is important that a set of symbols be developed as somewhat of a standard for a particular plant or company. Of course, these can be improved and modified with time and as needed, but with the basic forms and letters established, the sheets can be quite valuable. Many companies consider their flowsheets quite confidential since they contain the majority of key processing information, even if in summary form.

## LINE SYMBOLS AND DESIGNATIONS

The two types of lines on a flowsheet are (1) those representing outlines and details of equipment, instruments, and so on, and (2) those representing pipe carrying process or utility liquids, solids, or vapors and electrical or instrument connections. The latter must be distinguished among themselves as suggested by Figure 1-28.

In order to represent the basic type of solution flowing in a line, designations or codes to assign to the lines can be developed for each process. Some typical codes are as follows.

RW – River Water  
 TW – Treated Water  
 SW – Sea Water  
 BW – Brackish Water  
 CW – Chilled Water

S	– Low Pressure Steam
S150	– 150 psi Steam
S400	– 400 psi Steam
V	– Vent or Vacuum
C	– Condensate (pressure may be indicated)
D	– Drain to sewer or pit
EX	– Exhaust
M	– Methane
A	– Air (or PA for Plant Air)
F	– Freon
G	– Glycol
SA	– Sulfuric Acid
B	– Brine
Cl	– Chlorine
P	– Process mixture (use for in-process lines not definitely designated by other symbols).

Sometimes it is convenient to prefix these symbols by L to indicate that the designation is for a line and not a vessel or instrument.

**Materials of Construction for Lines.** The process designer must also consider the corrosive nature of the fluids involved when selecting construction of materials for the various process and utility service lines. Some designers attach these materials designations to the line designation on the flowsheets, while others identify them on the Line Summary Table (Refer Figure 1-31d). Some typical pipe materials designations are as follows.

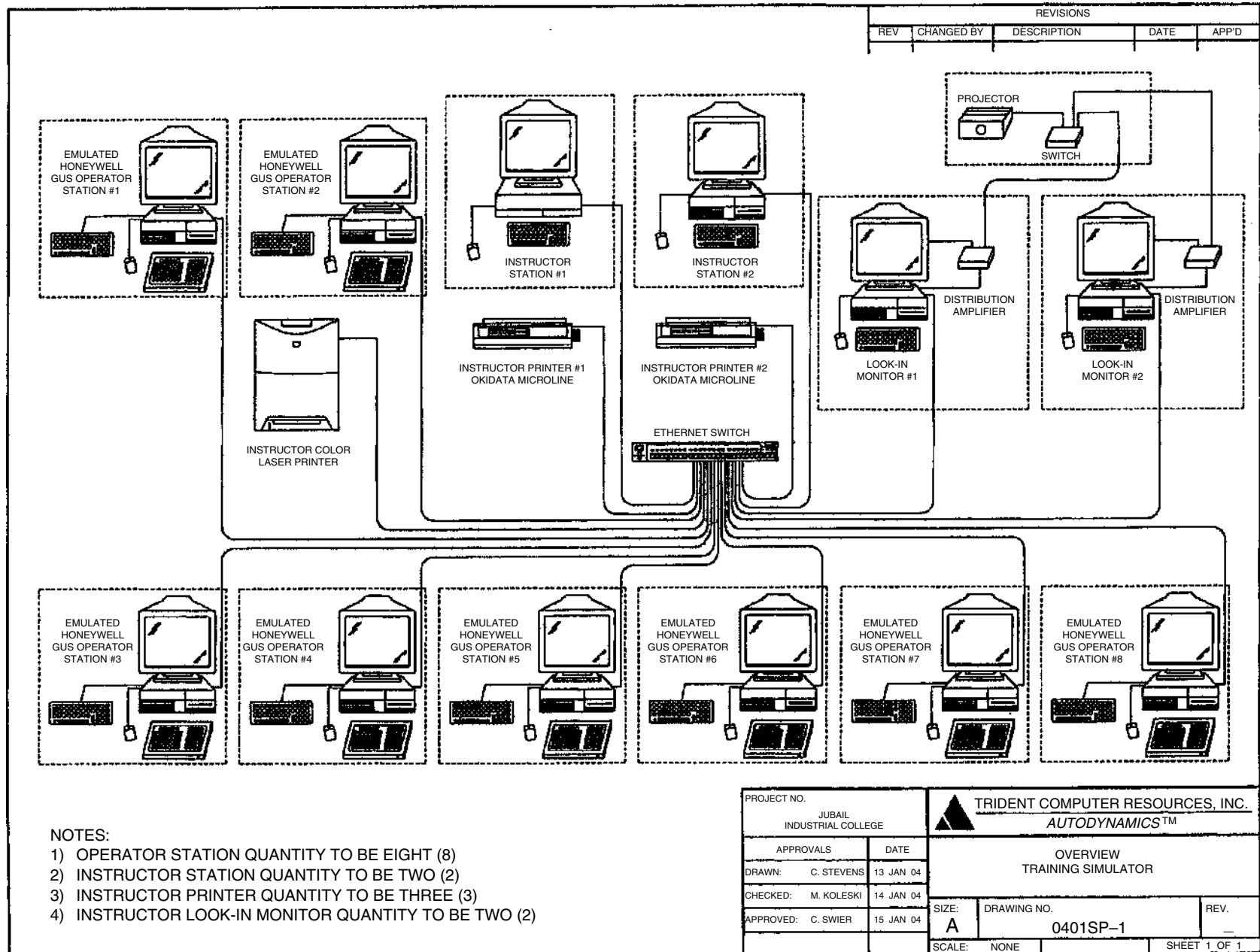
CS40	– Carbon steel, Schedule (Sch.) 40
CS80	– Carbon steel, Sch. 80
SS316/10	– Stainless steel, 316 m Sch. 10
GL/BE	– Glass bevel ends
N40	– Nickel, Sch. 40
TL/CS	– Teflon-lined carbon steel
PVC/CS	– Lined CS
Polyvinylchloride	– Solid polypropylene (designated weight Sch.)
PP	– Solid polypropylene (designated weight Sch.)

**Test Pressure for Lines.** The process designer also needs to designate the hydraulic test pressures for each line. This testing is performed after construction is essentially completed and often is conducted by testing sections of pipe systems, blanking off parts of the pipe or equipment, if necessary. Extreme care must be taken to avoid over pressuring any portion of pipe not suitable for a specific pressure, as well as extending test pressure through equipment not designed for that level. Vacuum systems must always be designed for “full vacuum,” regardless of the actual internal process absolute vacuum expected. This absolute zero designed basis will prevent the collapse of pipe and equipment should internal conditions vary. Some line design systems include the test pressure in the line code, but this often becomes too unwieldy for drafting purposes.

The usual complete line designation contains the following: (1) line size (nominal); (2) material code; (3) sequence number; and (4) materials of construction (see Figures 1-30 and 1.31a–d).

Examples:

2"-CL6-CS40  
 3"-CL6a-CS40  
 4"-RW1-CS40  
 16"-S150-CS40  
 3"-P-TL/CS.



**Figure 1-23** A plan layout of Autodynamic™ workstation. (Courtesy of Trident Computer Resources Inc.)



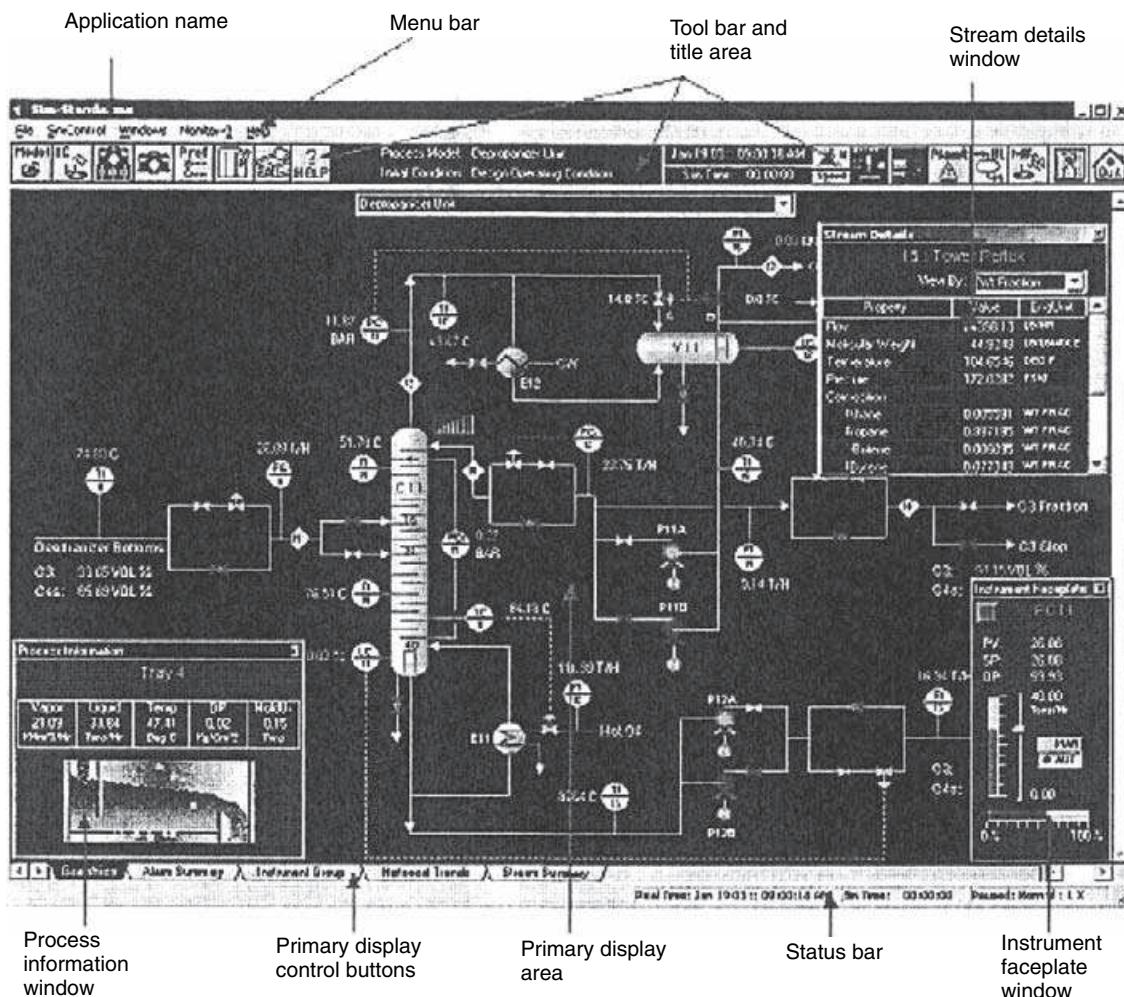
**Figure 1-24a** Autodynamic integrated keyboard. Courtesy of Trident Computer Resources Inc.



**Figure 1-24b** Typical Simulation Laboratory. Courtesy of Trident Computer Resources Inc.

Some engineers rearrange the sequence of the code although the information remains essentially the same. The line number sequence is conveniently arranged to start with 1 or 100 for each of the fluid designations (CL, P, etc.). Since the sequence numbers are for coordination purposes and will appear on piping drawings, Line Schedule (Figures 1.31a–d), the number has no significance in itself. It is convenient to start numbering with the first process flowsheet and carry on sequentially to each succeeding sheet. Sometimes, however, this is not possible when several detailers are

preparing different sheets, so each sheet can be given arbitrary beginning numbers such as 100, 300, 1000, and so on. Although the sequential number may be changed as the line connects from equipment to equipment, it is often convenient to use the system concept and apply alphabetical suffixes to the sequence number as shown in Figures 1-29 and 1-30. This contributes materially to the readability of the flowsheets. Each line on the flowsheet must represent an actual section or run of piping in the final plant and on the piping drawings.



**Figure 1-25a** Windows Assisted Simulation Advanced Trainer (WSAT) Snapshot of simulation graphical user interface of a binary system (Courtesy of Envision System Inc.)

Suggested guides for line identification for any one principal fluid composition are as follows:

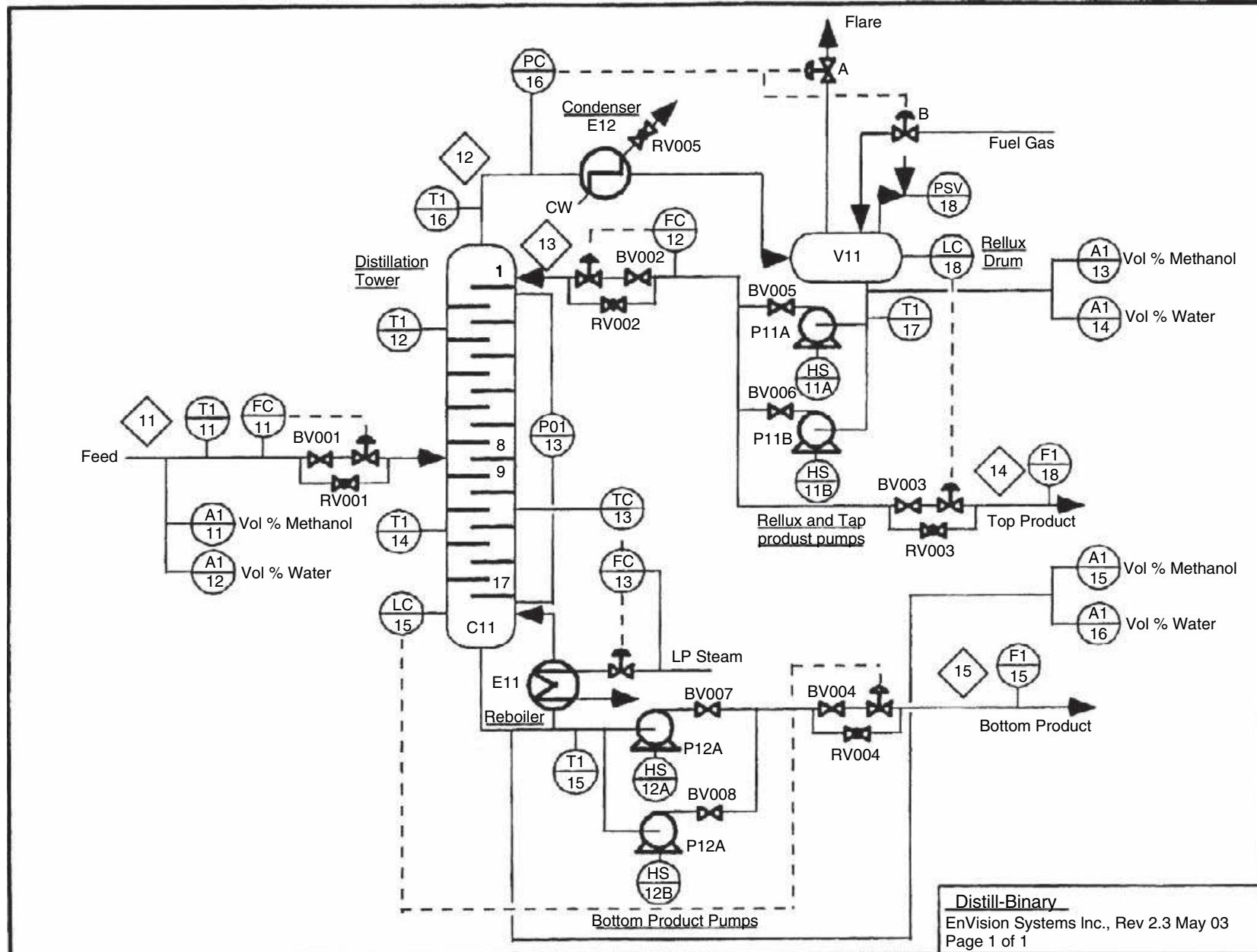
1. Main headers should keep one sequence number (Figure 1-30)
2. New sequence numbers should be assigned
  - (a) upon entering and leaving an item of equipment
  - (b) to take-off or branch lines from main headers
  - (c) to structural material composition of line changes.
3. Alphabetical suffixes should be used in the following situations as long as clarity of requirements is clear, otherwise add new sequence numbers
  - (a) for secondary branches from headers or header branches;
  - (b) for bypass lines around equipment, control valves, and so on. Keep same sequence number as the inlet or upstream line (Figure 1-30);
  - (c) for identical multiple systems, piping corresponding identical service items, and lines.

In order to coordinate the process flowsheet requirements with the mechanical piping specifications, Line Schedules are prepared as shown in Figures 1.31a-d. The complete pipe system specifications are summarized by codes on these schedules (refer Section 1.15).

Equipment code designations can be developed to suit the particular process, or as is customary a master coding can be established and followed for all projects. A suggested designation list (not all inclusive for all processes) for the usual process plant equipment is given in Table 1-6 and process functions in Table 1-7. The various items are usually numbered by type and in process flow order as set forth on the flowsheets. For example:

Item Code	Represents
C-1a	Three compressors of identical size operating in the same process service, connected in parallel
C-1b	Single compressor in different service (by fluid or compression ratio) from C-1's above
C-2	First separator in a process
S-1	Second separator in a process
S-2	Two identical separators connected in parallel, in the same process service.

Some equipment code systems number all items on first process flowsheet with 100 series as C-101, C-102, P-106 to represent compressors 101 and 102 in different services and pump 106 as the sixth pump on the sheet. The second sheet uses the 200 series, and



**Figure 1-25b** P & ID of a binary distillation system Methanol-Water (Courtesy of Envision Inc.)

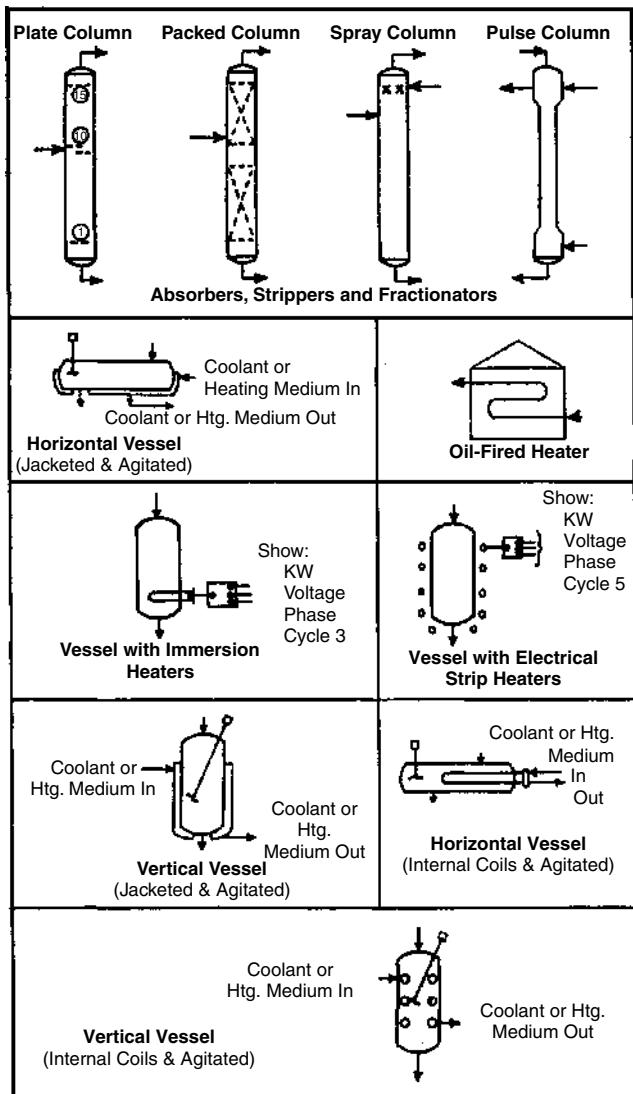


Figure 1-26a Process variables.

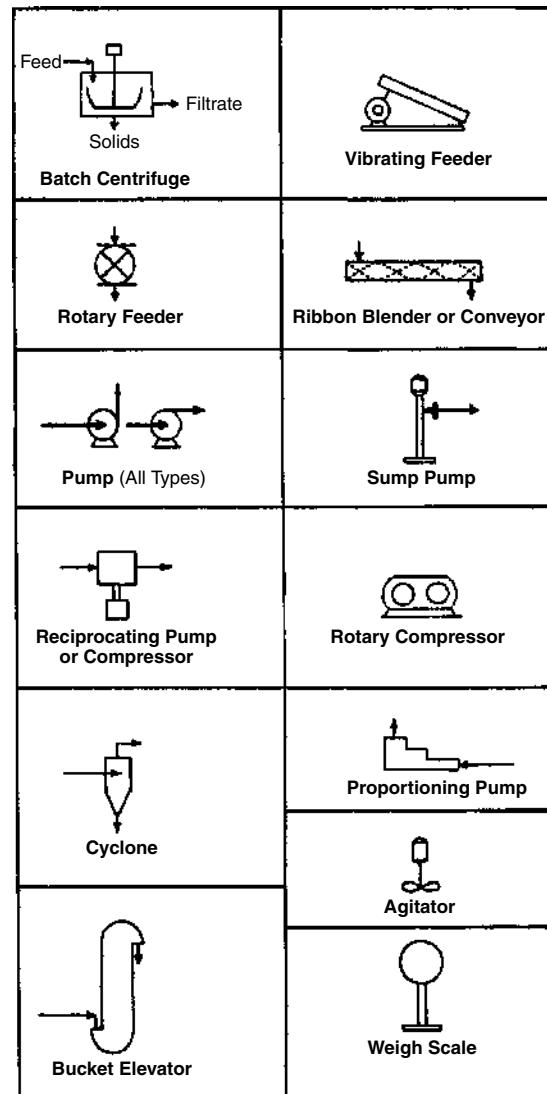


Figure 1-26b Pumps and solids.

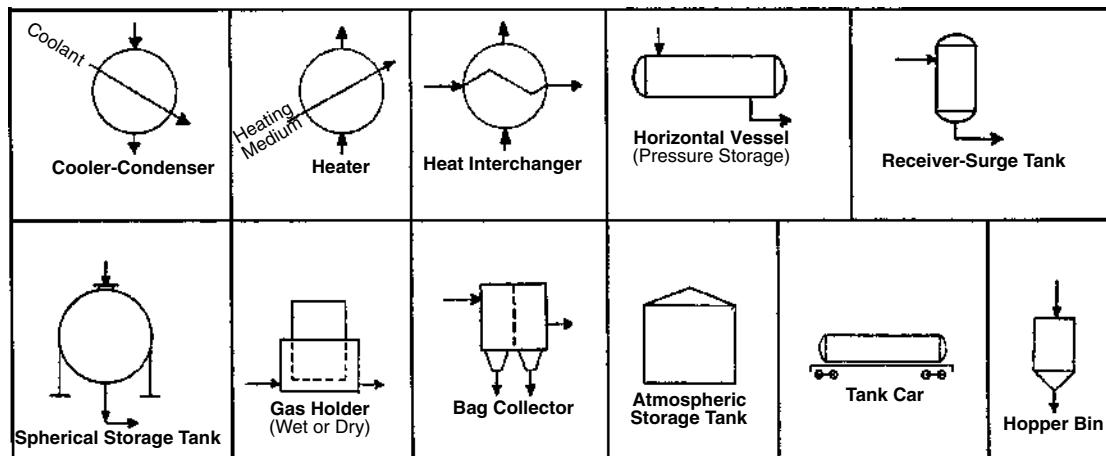


Figure 1-26c Storage and equipment.

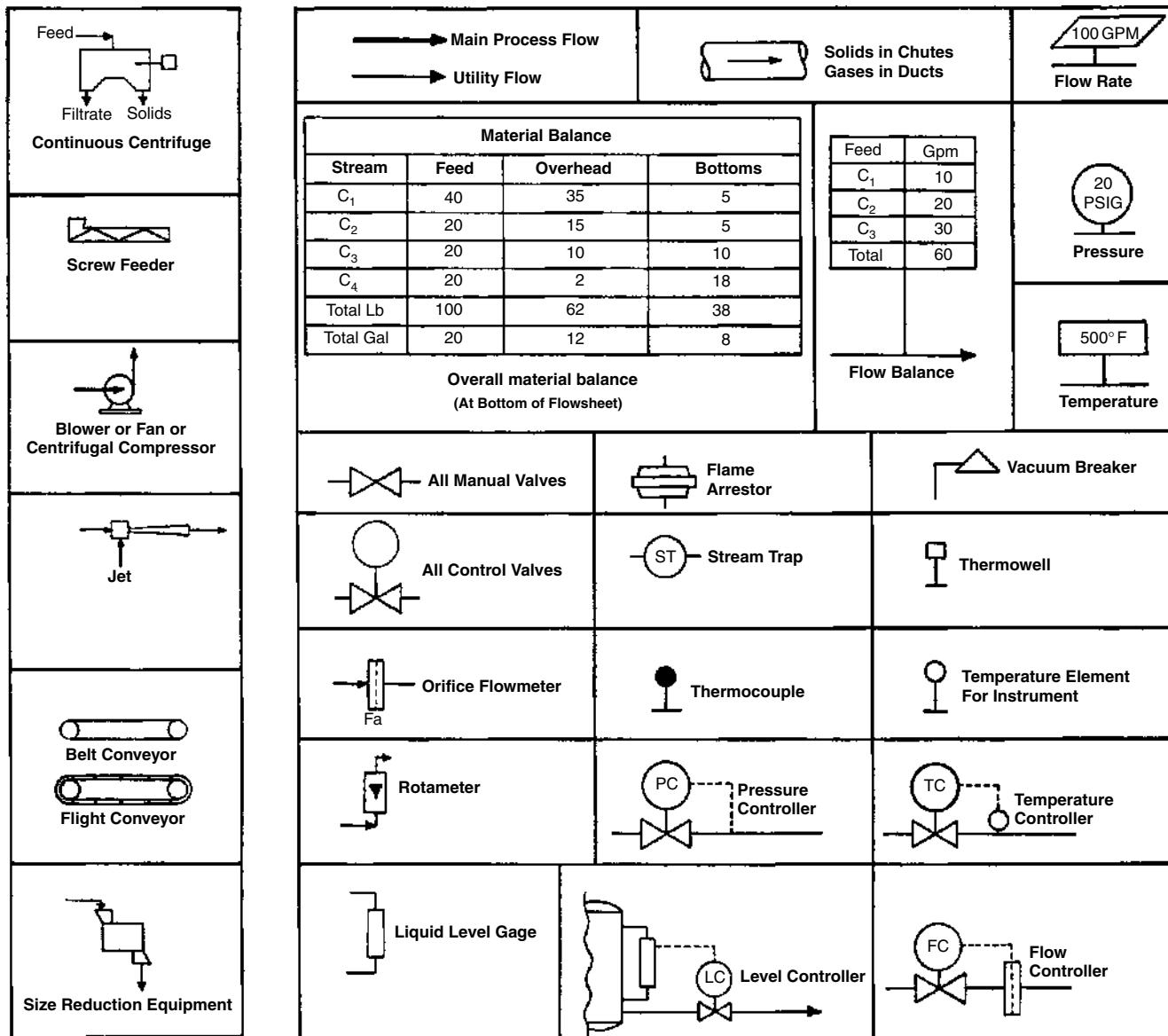


Figure 1-26d Flow and Instruments.

so on. This has some engineering convenience but is not always clear from the process view.

To keep process continuity clear, it is usually best to number all like items sequentially throughout the process, with no concern for which flowsheet they appear on. Also, another popular numbering arrangement is to identify a system such as reaction, drying, separation, purification, incineration, vent, and cooling tower waters and number all like process items within that system. For example, Reactor System, R: Reactor is RD-1

Reactor vent cooler is RE-1  
 Reactor vent condenser is RE-2  
 Reactor recycle pump is RP-1  
 Level control valve is RLC-1  
 Relief valve is RSV-1.

Then, establish the same concept for all other unit or block processing systems. This is often helpful for large projects, such

as refinery or grassroots chemical processes. Valve identification codes are usually used in preference to placing each valve specification on the flowsheet. This latter method is feasible for small systems, and is most workable when a given manufacturer (not necessarily the same manufacturer for all valves) can be selected and his valve catalog figure number used on the flowsheet. For large jobs, or where many projects are in progress at one time, it is common practice to establish valve specifications for the various process and utility services (Figures 1-32 and 1-33) by manufacturers' catalog figure numbers. These are coded as V-11, V-12, V-13, and so on, and such code numbers are used on the flowsheets whenever these valves are required (also see Figure 1-16). By completely defining the valve specification in a separate specification book the various valves – gate, globe, butterfly, plug, flanged end, screwed end, welding end – can be identified for all persons involved in a project, including piping engineers and field erection contractors.

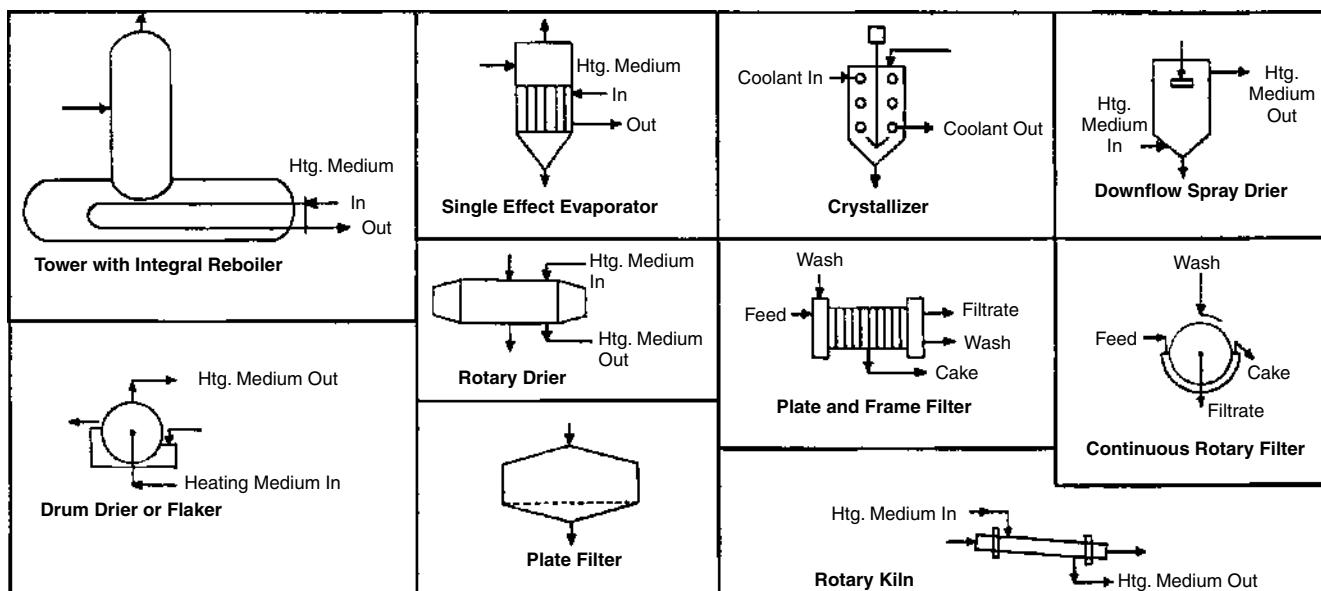


Figure 1-26e Filters, evaporators, and driers. (By permission from D. J. Orialo, Oil Gas J., 56, Nov 17, 1958, pp. 152–153.)

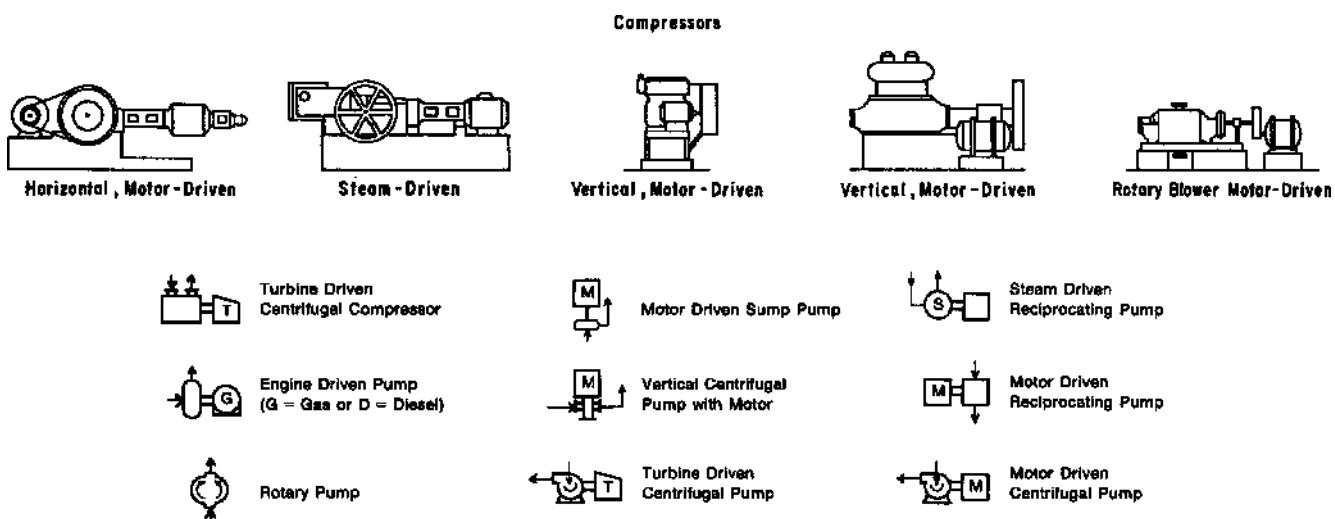


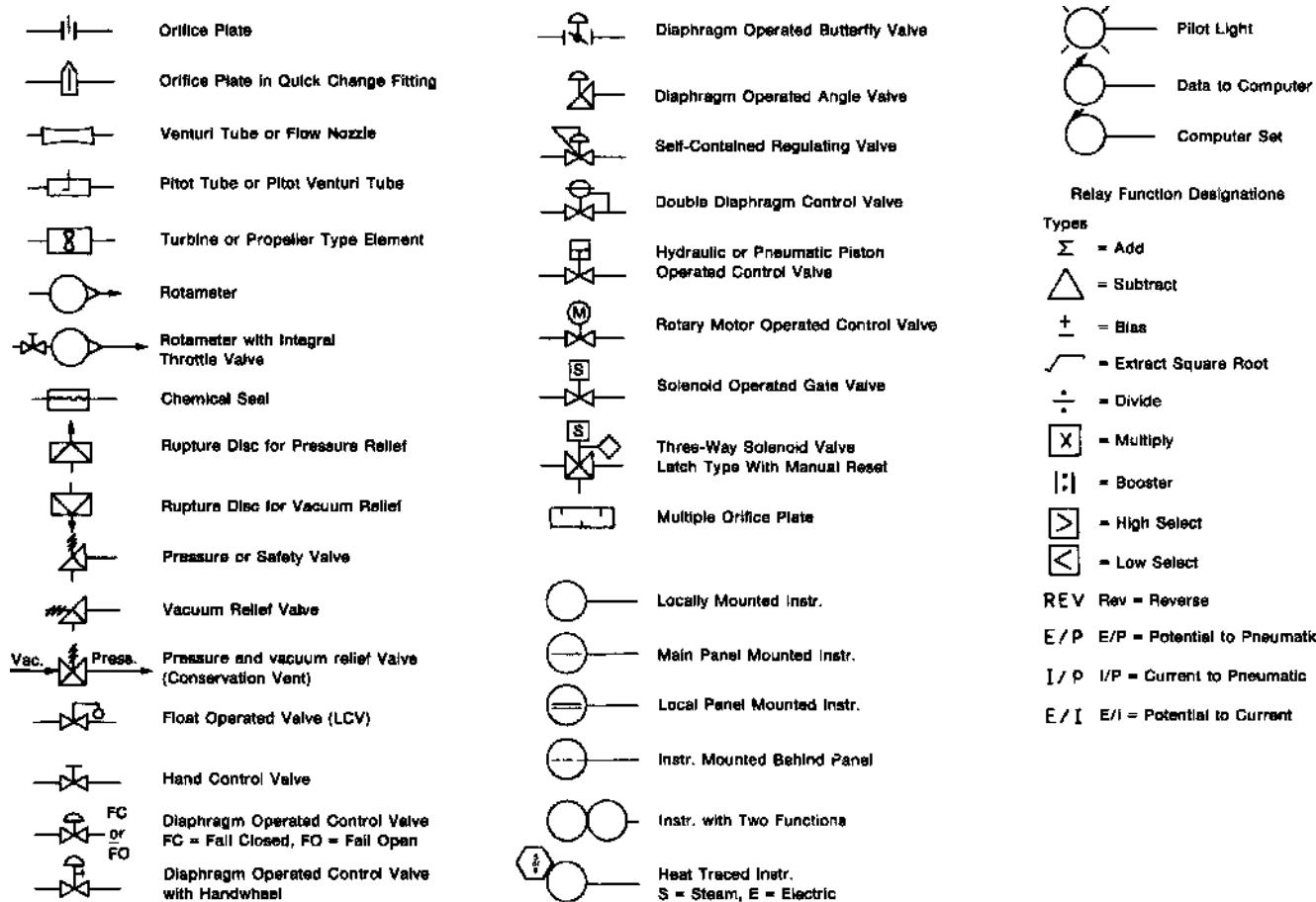
Figure 1-27a Special types of descriptive flowsheet symbols.

Figure 1-27c summarizes a system for representing components on the flowsheets. The instrument symbols and nomenclature shown in Table 1-8 and Figures 1-30a and b are representative of the types developed by the Instrument Society of America (ISA) and some companies.

Some other designation systems indicate the recording or indicating function in front of rather than behind the instrument function. For example:

RTC-1	Recording Temperature Controller No. 1
VRTC-1	Control Valve for Recording Temperature Controller No. 1
RFM-6	Recording Flow Meter No. 6

ORFM-6	Orifice flanges and plate for Recording Flow Meter No. 6
OTrRFC-1	Orifice flanges and plate used with Transmitter for Recording Flow Controller No. 1
TrRFC-1F	Flow Transmitter for Recording Flow Controller No. 1
IPC-8	Indicating Pressure Controller No. 8
IFC-6	Indicating Flow Controller No. 6
IFM-2	Indicating Flow Meter No. 2
RLC	Recording Level Controller
RLM	Recording Level Meter
ILC	Indicating Level Controller
LC	Level Controller
PC	Pressure Controller



**Figure 1-27b** Commonly used instruments flowsheets. (Adapted by permission from ISA Std. ANSI Y32.20 – 1975, ISA S5.1 – 1973, “Instrumentation Symbols and Identification,” latest edition, 1984.)

Control valves carry the same designation as the instrument to which they are connected. Thermocouples carry the same designation as the recorder or indicator to which they are connected. Sequential point numbers are indicated thus (see Table 1-8):

RTM-6-4 Thermocouple connected to Point No. 4 RTM  
Instrument No. 6 (also see Figure 1-16).

Additional symbols include the following:

PG-6	Pressure Gage No. 6 connected in the field on some item of equipment. If panel board mounted, it becomes 6B.
LTA-1	Low Temperature Alarm No. 1
HTA-1	High Temperature Alarm No. 1
LPA-2	Low Pressure Alarm No. 2
HPA-2	High Pressure Alarm No. 2
LLA-6	Low Level Alarm No. 6
HLA-8	High Level Alarm No. 8
PG	Push Button

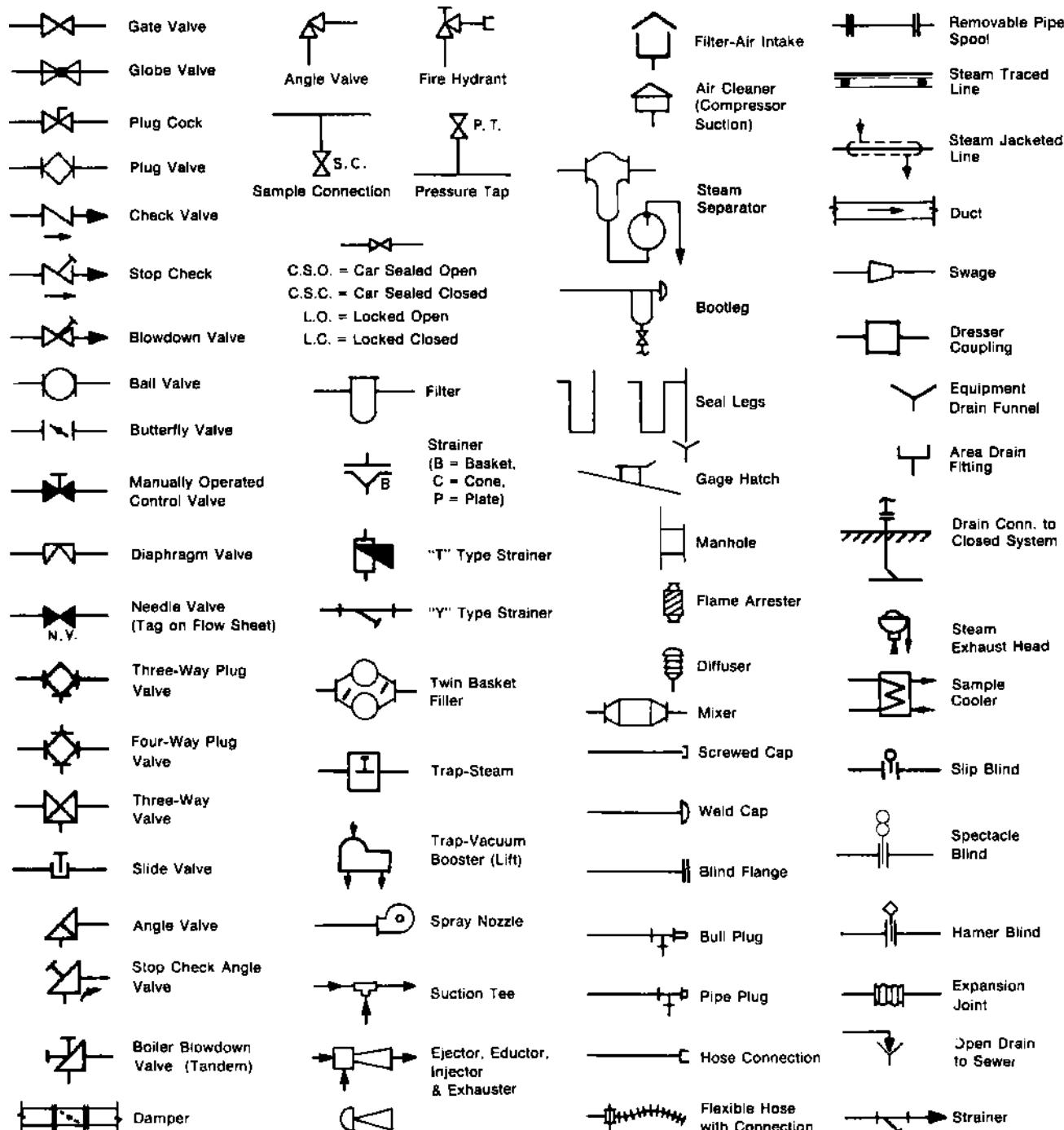
Process flowsheets do not normally show companion flanges for valves unless these serve as blinds or for orifice plates. This detail is sometimes shown on the piping flowsheet, but here again the use of detail, which does not contribute to the communication function of the sheets, is avoided. Such detail can be time-consuming when

considered over the entire set of sheets for a process. Figures 1-13 and 1-14 are typical of reasonably good presentation without unnecessary detail. Such specifications as heights of a seal leg, locked open valve, or other information not summarized elsewhere must be recorded on the flowsheets.

## 1.15 WORKING SCHEDULES

As a direct companion of the completed flowsheet, the line schedule sheet transmits the process and mechanically necessary details for proper interpretation of the piping aspects of the flowsheet (see Figures 1.31a-d). These schedules are initiated by the process engineer to further explain the requirements of the process as shown on the flowsheets. They are often and perhaps usually cooperatively completed by other engineers, particularly the piping, mechanical, and instrumentation groups.

A schedule similar to Figure 1.31a is used to summarize insulation process code or class, and pressure test information to the erection contractor. The process code is the complete code specification (as a separate fluid process service detailed for each fluid) tabulation for the required piping materials, fittings, valves, gaskets, thread lubricant, and so on for a specific process or utility fluid (see Figures 1-32 and 1-33). For example, it identifies the



**Figure 1-27c** Flow diagram symbols: Valves, fittings, and miscellaneous piping. (Compiled from several sources and, in particular, Fluor Corp., Ltd.)

type of gate, globe, plug, check, and needle valves to be used in the fluid by specific catalog figure number of a manufacturer or its equivalent. This requires attention to materials of construction, pressure-temperature ratings, and connections (flanged, screwed, weld-end), bonnet type, packing, seat type (removable or non removable), stem, and any other details affecting the selection of a valve for the process fluid conditions. It also contains the specifications for pipe, fittings, flanges, unions, couplings, gaskets, thread

compound, bolting, and any special materials needed to properly complete the piping requirements.

Other schedules and summaries include vessels (tanks and drums), towers or columns, heat exchangers, pumps, compressors, motors, trip, utility summary, hazardous chemicals and conditions, safety analysis table, and so on. These are often developed by the process engineer for organizational uses by the process designers as well as by other engineering groups. Again, these are often

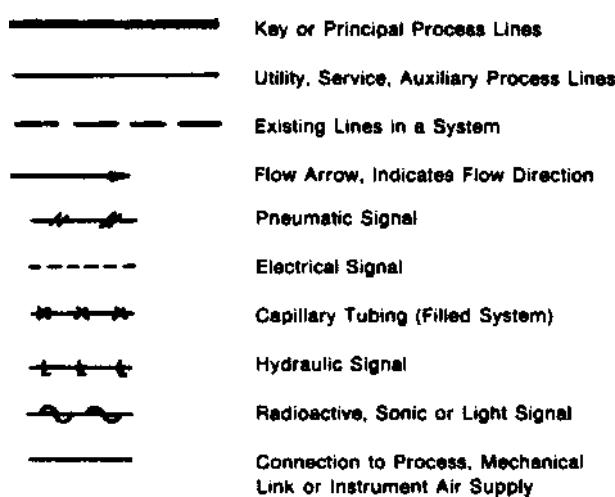


Figure 1-28 Line symbols. (By permission from ISA Std. S51 – 1973 and 1984.)

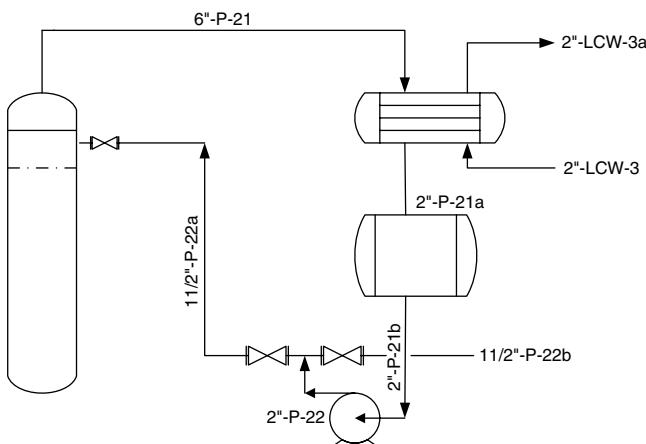


Figure 1-29 Use of alphabetical suffixes with line symbols.

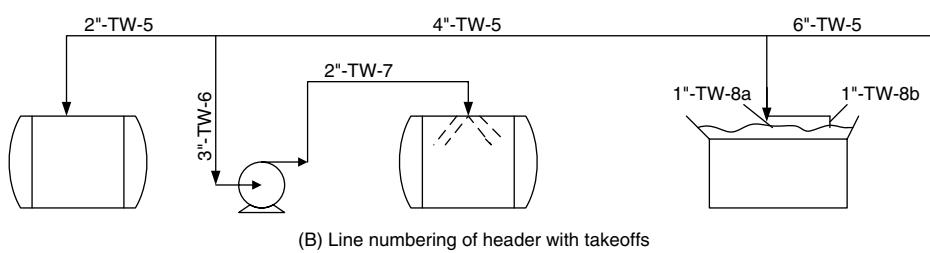
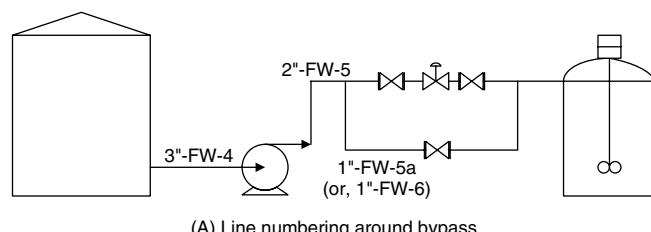


Figure 1-30 Examples of line numbering.

cooperatively and sometimes completely prepared by a particular specialty group after interpreting and designing for the needs of the process (Figures 1-34–1-37). These figures and many others (process data sheets) can be downloaded from the companion website. Appendix A shows some process flow diagrams and engineering process data sheets/schedules.

Two types of schedules are in use:

1. The summary sheet, which summarizes process conditions and equipment selection.
2. The schedule sheet, which summarizes the key reference data for a particular class of equipment such as pumps, but contains no process data. The latter type is prepared for job coordination with and in the various departments, that is, engineering, construction, purchasing, and production. It primarily serves for the construction period but, of course, does have lasting cross-reference value.

From a construction viewpoint these summaries are a necessary checklist to aid in keeping the construction program organized. Individuals who have no real knowledge of the scope of the job, and in particular the process, can properly tie the project together in the field by use of these schedules.

## 1.16 INFORMATION CHECKLISTS

The process engineer must summarize in some form the raw material and utility requirements for use by others. For example, the civil engineer is interested in wastewater and sanitary sewer flows for proper layout studies. He is also in need of special requirements for site development as well as railroads. The checklist of Figure 1-38 is an example of a helpful form. Others can be developed to suit the project or general plant situation.

For immediate job reference as well as for estimating requirements of a process for expansion purposes, the form shown in Figure 1-39 is convenient and can be expanded to suit the process under consideration.

## ENGINEERING ETHICS, STANDARDS, AND CODES

In 1954, the National Society of Professional Engineers (NSPE) adopted the following statement, known as the Engineers' Creed:

**Figure 1-31a** Line Schedule.

*Note:* 1. Flow Units: G = gpm; C = acfm; P = lb/h.

**Figure 1-31b** Pipe line list.

**Figure 1-31c** Line Schedule.

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**Figure 1-31d** Line summary table.

**TABLE 1-6 A System of Equipment Designations**

AD	- Air Drier
AF	- Air Filter
Ag	- Agitator
B	- Blower
BR	- Barometric Refrigeration Unit
C	- Compressor
CP	- Car Puller
CT	- Cooling Tower
CV	- Conveyor
D	- Drum or tank
DS	- Desuperheater
E	- Heat Exchanger, condenser, reboiler, and so on.
Ej	- Jet Ejector
Ex	- Expansion Joint
F	- Fan
FA	- Flame Arrestor
Fi	- Filter (line type, tank, centrifugal)
GT	- Gas Turbine
MB	- Motor for Blower
MC	- Motor for Compressor
MF	- Motor for Fan
MP	- Motor for Pump
P	- Pump
PH	- Process Heater or Furnace
R	- Reactor
S	- Separator
St	- Strainer
ST	- Steam Turbine
Str	- Steam Trap
SV	- Safety Valve
Tr	- Trap
V	- Valve
VRV	- Vacuum Relief Valve

**TABLE 1-7 Typical Identification for Flowsheet Process Functions**

AS	- Air Supply
BD	- Blowdown
BF	- Blind Flange
CBD	- Continuous Blowdown
CD	- Closed Drain
CH-O	- Chained Operated
CSO	- Car Seal Open
CSC	- Car Seal Closed
DC	- Drain Connection
EBD	- Emergency Blowdown Valve
ESD	- Emergency Shutdown
FC	- Fail Closed
FO	- Fail Open
HC	- Hose Connection
IBD	- Intermittent Blowdown
LO	- Lock Open
ML	- Manual Loading
NC	- Normally Closed
NO	- Normally Open
OD	- Open Drain
P	- Personnel Protection
QO	- Quick Opening
SC	- Sample Protection
SO	- Steam Out
TSO	- Tight Shut Off
VB	- Vacuum Breaker

V-No.				
MATL.				
SIZE 8	CHECK VALVE			
CONN.	DESCRIPTION			
V-11  1 1/4 Cr. 2 1/2"-14" 600# RF	PISTON LIFT, PRESS. SEAL RATING: 600 psig @ 975° F BODY: C.A.S. A-217 GR WC6 STEM: SEATS: Integral Stel Alloy DISC: Body-Guided	ROCKWELL EDWARDS 690 WC6 1/2"-6"	ROCKWELL EDWARDS 694 WC6 8"-14"	
V-12  CS 1/2"-2" 2500# SW	HORIZ. PISTON, WELD CAP RATING: 2500 psig @ 650° F BODY: C.S. A-216 Gr. WCB STEM: STEATS: Integral, Stellited DISC:	ROCKWELL EDWARDS 6674	VOGT SW-6933	1/2"-2"
V-13  CS 2 1/2"-12" 2500# BW	HORIZ. PISTON, PRESS. SEAL RATING: 2500@ 650° F BODY: C.S.A.-216 Gr. WCB STEM: SEATS: Integral, Stellited DISC: Piston Stellited	ROCKWELL EDWARDS 3994Y WCB 2 1/2"-12"	POWELL 125065 WE	2 1/2"-10" (5) (6)
V	Add additional valves of all types As needed for project			

Note: 1. Vertical columns indicate valves acceptable as equivalent to the specification description.  
 2. V-11 is a typical valve code to use on flowsheets and piping drawings.

**Figure 1-32** Typical valve codes and specifications. (By permission from Borden Chemicals and Plastics Operating Ltd Partnership.)

## GENERAL PIPING MATERIAL SPECIFICATIONS

<b>GENERAL MATERIAL</b>		<b>: Carbon Steel</b>
<b>MAXIMUM DESIGN PRESSURE and TEMPERATURE LIMITS</b>		<b>: 275 psig AT 20/100° F; 100 psig at 750° F</b>
<b>LIMITED BY</b>		<b>: 150# Flanges</b>
<b>CORROSION ALLOWANCE</b>		<b>: See Table, This Spec.</b>
<b>CONSTRUCTION</b>		<b>: 1½" and Smaller – Socket Welded : 2" and Larger – Flanged and Butt-Welded</b>
<b>TYPE</b>	<b>SIZE</b>	<b>DESCRIPTION</b>
<b>PIPE:</b>	1½" and smaller  2" through 10"  12" through 24"	Schedule 80, ASTM-A106 Gr. B Seamless P.E. (Plain End). Nipples: Sch. 80 ASTM-A106 Gr. B Schedule 40, Standard Weight, ASTM-A53 Gr. B, Seamless, B.E. (Bevel End). Standard Weight (.375" ASTM-A53, Gr.B., Seamless, B.E.
<b>FITTINGS:</b>	1½" and smaller 2" through 10"  12" through 24" Full Half header Size and larger Less half header Size down through 2"	3000# F.S., Socket Weld, ASTM-A105 Gr. I or II Schedule 40, Standard Weight, Butt-Weld ASTM -A234 Gr. WPB, Seamless Same Except Use Standard Weight (.375") Use Tees
<b>BRANCHES:</b>		Straight Tee and Reducer or Reducing Tee
<b>FLANGES:</b>	1½" and smaller 1½" and smaller  2" and larger	Sockolets, Elbolets and Nipoletes 150# ASA, 1/16" R.F., Socket Weld ASTM-A181 Gr. I 150# ASA 1/16" R.F. Weld Neck, ASTM-A181 Gr. I
<b>UNIONS:</b> (6)	1½" and smaller	3000# F.S. Union ASTM-A105 Gr. II, Socket Weld ASA B16.11. Steel to Steel Seats, Ground Joint. No Bronze
<b>BOLTING:</b>	All	ASTM-A193 Gr. B7, Alloy Steel Stud Bolts, with ASTM-A194, Class 2H Heavy Series, Hex. Nuts 1/16" Thick, Compressed Asbestos Flat Ring Type. (JM 60 or Equal) 500° F and above, use Flexitallic CG.
<b>GASKETS:</b>		
<b>THREAD LUBRICANT:</b>	450° F and under Over 450° F	Use Teflon Tape Use "Molykote" G Paste
<b>GATE VALVES:</b> (4)	1½" and smaller  ¾" and smaller (1) 2" and larger (2) (7)	VGA-112, 800#, Socket Weld Ends, Welded Bonnet, F.S., ASTM-A105 Gr. II VGA-113, 800#, Screwed Ends, Welded Bonnet, F.S., ASTM-A105 Gr. II VGA-101, 150#, Flanged O.S. & Y., Cast Steel Body, ASTM-A216 WCB
<b>GLOBE VALVE:</b> (4)	1½" and smaller  2" through 12" (7)	VGL-215, 800#, Socket Weld Ends, Welded Bonnet, F.S., ASTM-105 Gr. II VGL-200, 150#, Flanged, O.S. & Y., Cast Steel Body, ASTM-A216 WCB
<b>CHECK VALVES:</b> (4)	1½" and smaller (3)  2" through 16"	VCH-314, 800#, Horizontal Piston Type Socket Weld Ends, F.S., ASTM-A105 Gr. II VCH-312, 800#, Combination Horizontal & Vertical Ball Type, Socket Weld Ends, F.S., ASTM-A105, Gr. II VCH-302, 150#, Horizontal Swing Check, Flanged, Cast Steel Body, ASTM-A216 WCB
<b>DRAINS, VENTS and INSTRUMENTS:</b> (4)	1" and smaller	VGA-120, 800#, Male Socket Weld x Female Thread Ends, Welded Bonnet, F.S., ASTM-A105, Gr. II.

**Figure 1-33** Partial presentation of piping materials specifications for a specific process service. (By permission from Borden Chemicals and Plastics, Operating Ltd Partnership.)

**PIPING MATERIAL SPECIFICATIONS**  
Alternative Process Service

Press./Temp. Limits: 175 psig/-20 to 150° F      125 psig/350° F Corrosion Allowance: 0.05 inches			
ITEMS	RATING & TYPE	MATERIAL OR MANUFACTURER	NOTE
<b>SIZE INCHES</b>			
<b>PIPE</b>			
2 and smaller	Sch. 40 Seamless	Carbon steel ASTM A-53, Gr. B, T& C	
3 through 6	Sch. 40 ERW	Carbon steel ASTM A-53, Gr. B, beveled	
8 through 12	Sch. 20 ERW	Carbon steel ASTM A-53, Gr. B, beveled	
14 through 20	Sch. 10 ERW	Carbon steel ASTM A-53, Gr. B, beveled	
<b>FITTINGS</b>			
2 and smaller	150# Screwed Buttweld-Sch. To Match pipe	Mal. Iron ASTM A-197 Carbon steel ASTM A-234, Gr. WPB.	
<b>FLANGES</b>			
2 and smaller	150# RF or FF Screwed	Carbon steel ASTM A-105	
3 and larger	150# RF or FF Slip on or weld neck	Carbon steel ASTM A-105	
<b>ORIFICE FLANGES</b>			
1 and larger	300# RF Weld Neck	Carbon steel ASTM A-105	
<b>UNIONS</b>			
2 and smaller	300# Screwed	Mal. Iron, ground joint, brass to iron seats ASTM A-197	
<b>BRANCH CONN.</b>			
2 and smaller	3000# Threadolet Std. Wt. Weldolet	Forged steel ASTM A-105	
3 and larger		Forged steel ASTM A-105	
<b>REDUCERS</b>			
2 and smaller	150# Screwed Sch. 80 Swage Buttweld-Sch. to match pipe	Mal. Iron ASTM A-197 Carbon steel ASTM A-234, Gr. WPB Carbon steel ASTM A-234, Gr. WPB	
3 and larger			
<b>STRAINERS</b>			
2 and smaller	150 screwed	Bronze with 30 mesh monel screen-Mueller # 351 or equal.	
<b>GASKETS</b>			
All sizes	1/16 in. ring	Compressed	
All sizes	1/16 in. full face	Compressed	
<b>VALVES</b> (Alternate, for 2 and smaller)	different process liquid/150# screwed gate	vapor service) Bronze, ISRS, union bonnet, Powell 2714 or equal	
3 or larger	125# FF gate	IBBM, OS&Y, bolted bonnet, Powell 1793 or equal	
2 and smaller	300# screwed ball	CS body, Teflon seats & seals CS ball, Hill McCanna Fig. S302-CSTCS	
3 and larger	150# RF ball	CS body, Teflon seats & seals CS ball, Hill McCanna Fig. S151-CSTCS	
3-6	150# butterfly w/locking handle	Cast iron body, Buna N seat & seals, Al-Brz. Disc, 316 SS stem. Keystone Fig. 100/122 or equal	

Figure 1-33—(continued)

**TABLE 1-8 Instrumentation Nomenclature – Complete General Identification**

Modification of ISA Standards															
First Letter  * Process Variable or Actuation	Second and Third Letters														
	Controlling Devices							Measuring Devices		Alarm Devices			Primary Elements	Wells	Glass Devices for Visual Observation
	Recording	Indicating	Nonindicating (Blind)	Valves	Self-Actuated valve	Safety Valve	Recording	Indicating	Recording	Indicating	Nonindicating (Blind)				
	RC	IC	C	CV	V	SV	R	I	RA	IA	A	E	W	G	
Flow	F	FRC	FIC			FV		FR	FI	FRA	FIA	FA	FE		FG
Level	L	LRC	LIC	LC	LCV	LV		LR	LI	LRA	LIA	LA	LE		LG
Pressure	P	PRC	PIC	PC	PCV	PV	PSV	PR	PI	PRA	PIA	PA	PE		
Speed	S	SRC	SIC	SC	SCV		SSV	SR	SI	SRA	SIA	SA			
Weight	W	WRC	WIC					WR	WI	WRA	WIA		WE		
Analysis	A	ARC	AIC	AC	ACV		ASV	AR	AI	ARA	AIA	AA	AE		
Hand	H		HIC	HC	HCV										
Temperature	T	TRC	TIC	TC	TCV	TV	TSV	TR	TI	TRA	TIA	TA	TE	TW	
Special	X	XRC	XIC	XC	XCV		XSV	XR	XI	XRA	XIA	XA	XE		

**Note:** Blank spaces are impossible or improbable combinations.

(Source: By permission from Oriolo D.J., Oil Gas J., Nov 17, 1958; also see ISA Stds Latest edition.)

As a Professional Engineer, I dedicate my professional knowledge and skill to the advancement and betterment of human welfare.

I pledge:

To give the utmost of performance;

To participate in none but honest enterprise;

To live and work according to the laws of man and the highest standards of professional conduct;

To place service before profit, the honor and standing of the profession before personal advantage, and the public welfare above all other considerations.

In humility and with need for Divine Guidance, I make this pledge.

In 1977, the Accreditation Board for Engineering and Technology (ABET) approved a similar statement as follows:

Engineers uphold and advance the integrity, honor, and dignity of the engineering profession by:

1. Using their knowledge and skill for the enhancement of human welfare;
2. Being honest and impartial, and serving with fidelity the publics, their employers;
3. Striving to increase the competence and prestige of the engineering profession; and
4. Supporting the professional and technical societies of their disciplines.

These statements relate to ethics or moral philosophy, which is derived from the Greek *ethika*, meaning character. Therefore, ethics deals with standards of morals or conduct, and engineering ethics is concerned with the personal conduct of engineers as they uphold and advance the integrity, honor, and dignity of engineering while practicing their profession. This conduct of behavior has obligations to

1. self
2. employer and/or client
3. colleagues and co-workers
4. public, and
5. environment.



## CENTRIFUGAL PUMP SUMMARY

**Figure 1-34** Centrifugal pump summary.

**Figure 1-35a** Centrifugal pump schedule.

**Figure 1-35b** Centrifugal pump schedule.

**Figure 1-36** Vessel and tank summary sheet.

**Figure 1-37** Vessel and tank schedule.



## GENERAL SERVICES AND UTILITIES CHECK LIST

\*ASSUME 1 HORSE POWER = 1 KVA

**Figure 1-38** General services and utilities checklist.

		PROCESS ENGINEERING JOB ANALYSIS SUMMARY		
Job Title _____				
Job No. _____	Charge No. _____	Date _____		
Based upon Cost Estimated Dated _____ or Actual Construction Cost _____				
Summary Prepared By _____	Information Dated _____			
Production Basis (lbs/day, tons/day, lbs/month)				
Service Requirements:		Unit Rate	Unit Rate/ Production Basis	
1	Steam (30 lbs.)	lbs/hr.		
2	Steam (150 lbs.)	lbs/hr.		
3	Steam (400 lbs.)	lbs/hr.		
4	Steam (    lbs.)	lbs/hr.		
5	Treated R.W.	gpm		
6	Untreated R.W.	gpm		
7	Fresh Water	gpm		
8	Sea Water	gpm		
9	Fuel Gas (    psi)	cfm (60°F & 1 atm)		
10	Air (    psi)	cfm (60°F & 1 atm)		
11	Power (    )			
12	Horsepower			
13	Condensate	lbs/hr.		
14				
Raw Materials		Unit Rate		
1	Chlorine			
2	Hydrogen (    %)			
3	Caustic (    %)			
4	Salt			
5	Sat. Brine			
6	Natural Gas			
7	Air			
8	Ethylene			
9				
10				
11				
Products and By-Products		Unit Rate		
1	Chlorine			
2	HCl (    %)			
3	Salt (    %)			
4	Caustic (    %)			
5	Ammonia (    %)			
6	H <sub>2</sub> SO <sub>4</sub> (    %)			
7	Gas (    )			
8				
9				
10				
11				

**Figure 1-39** Process engineering job analysis summary.

Appendix B provides specific examples of these obligations given in individual codes of ethics, which are adopted by the various engineering societies (e.g., ASME, AIChE, NSPE, and IChemE).

The requirement for standardization started early in the evolution of the modern engineering industry, where engineering standards cover a much wider function such as the following.

1. materials, properties, and composition
2. testing procedures for performance, compositions, and quality
3. preferred sizes (e.g., tubes, plates, sections)

4. design methods, inspection, and fabrication
5. codes of practice, for plant operation and safety.

The terms Standard and Code are often used interchangeably, though Code should be reserved for a Code of practice covering, for example, a recommended design or operating procedure, and Standard for preferred sizes, compositions, and so on. All the developed countries and many of the developing countries have national standards organizations, which are responsible for the issue and maintenance of standards for the manufacturing industries and for the protection of consumers.

In the United States, the government organization responsible for coordinating information on standards is the National Bureau of Standards; standards are issued by the Federal, States, and various commercial organizations. The principal ones of interest to the chemical/process engineers are those issued by the American National Standards Institute (ANSI), the American Petroleum Institute (API), the American Society for Testing Materials (ASTM), and the American Society of Mechanical Engineers (ASME) (pressure vessels). In the United Kingdom, preparation and promulgation of national standards are the sole responsibilities of the British Standards Institution (BSI). The Institution has a secretariat and a number of technical personnel, but the preparation of the standards is the responsibility of committees of persons from the appropriate industry, such as the professional engineering institutions and other interested organizations. The International Organization for Standardization (ISO) coordinates the publication of international standards.

The process design engineer must in effect become a good general-purpose engineer who recognizes the need for integrating the various engineering disciplines into the process details as may be required. The engineer becomes what might be termed a "pseudo-mechanical, corrosion, and metallurgical engineer" as well as a "basic chemical engineer." The design engineer must, or should soon, be knowledgeable of all types of information and specifications necessary to totally perform the process design functions in all detail and scope. A partial list of these specifications follows.

It is recommended that all pressure vessels and atmospheric vessels be designed, fabricated, tested, and code-stamped according to the most applicable code as ASME or API, regardless of service application (nuclear is excluded from any discussion in these chapters):

1. American Society of Mechanical Engineers (ASME), website: [www.asme.org](http://www.asme.org), codes are as follows.
  - ASME Unfired Pressure Vessel Code, Section 8, Division 1
  - ASME Code, Materials Specification, Part A, Ferrous Materials.
  - ASME Code, Materials Specification, Part B, Non-Ferrous Materials.
  - ASME Section V, Non-Destructive Examination.
2. American Society For Testing Materials, Part 10, Annual Book of ASTM Standards: Metals-Physical, Mechanical, and Corrosion Testing.
3. General Recommendations for Spacing in Refineries, Petrochemical Plants, Gasoline Plants, Terminals, Oil Pump Stations and Offshore Properties, Industrial Risk Insurance, Hartford, CT (see [3]).
4. American Standards Association, Petroleum Refinery Piping ASA B31.3 (latest edition).
5. Standards of the Tubular Exchangers Manufacturers Association (TEMA Standards, latest editions).
6. National Fire Protection Association Codes are as follows:
7. Standards and Recommendation Practices of American Petroleum Institute (API), website: [www.api.org](http://www.api.org).
  - 520 Design and Installation of Pressure-Relieving systems in Refineries.
    - Part I Design
    - Part II Installation
  - 521 Guide for Pressure Relief of Depressuring Systems.
  - 525 Testing Procedures for Pressure-Relieving Devices Discharging Against Variable Back Pressure.

<b>Standard</b>	<b>Code</b>
Blower and Exhaust Systems	91
Chemical Reaction, Hazardous	491M
Chemical Data, Hazardous	49
Chimneys, Vents, Fireplaces, and Solid Fuel Burning Appliances	211
Coding, Uniform for Fire Protection	901
Dry Chemical Extinguishing Systems	17
Electrical Code, National	70
Electrical Equipment in Hazardous (Class.) Locations, Gases, Vapors, Dusts	497M
Electrical Equipment, Purged and Pressurized Enclosures for Electrical Installations	496
Classification of Class 1 Hazardous Locations	497
Explosion Prevention Systems	69
Explosion Venting	68
Explosive Materials, Code for	495
Fire Hazards of Materials, Identification	704
Fire Pumps, Centrifugal	20
Fire Pumps, Steam	21
Flammable and Combustible Liquids, Class	321
Flammable and Combustible Liquids Code	30
Flammable and Combustible Liquids, Farm Storage of	395
Flammable and Combustible Liquids, Portable Shipping Tanks	386
Flammable and Combustible Liquids, Tank Vehicles for	385

- 526 Flanged Steel Safety Relief Valves for Use in Petroleum Refineries.
- 527 Commercial Seat Tightness of Safety Relief Valves with Metal-to-Metal Seats.
- 540 Recommended Practice for Electrical Installations in Petroleum Refineries.
- 550 Installation of Refinery Instruments and Control Systems
  - Part I Process Instrumentation Control
  - Part II Process Stream Analyzers
- 1101 Measurement of Petroleum Liquid Hydrocarbons by Positive Displacement Meter.
- 2000 Venting Atmospheric and Low Pressure Storage tanks (Non-refrigerated and Refrigerated).
- 2545 Method of Gauging Petroleum and Petroleum Products.
- 2217 Guidelines for Confined Space Work in the Petroleum Industry.
- 2513 Evaporation Loss in the Petroleum Industry – Causes and Control.
- 2516 Evaporation Loss from Low-Pressure Tanks.
- 2517 Evaporation Loss from External Floating Roof Tanks.
- 2518 Evaporation Loss from Fixed-Roof Tanks.
- Chapter II Guide for Inspection of Refinery Equipment – Conditions Causing Deterioration or Failures.
- Chapter IV Guide for Inspection of Refinery Equipment – Inspection Tools.
- Chapter V Preparation of Equipment for Safe Entry and Work.
- Chapter VI Pressure Vessels (Tower, Drums, and Reactors).
- Chapter VII Heat Exchangers, Condensers, and Cooler Boxes.
- Chapter IX Fired/Heaters and Stacks.
- Chapter IXX Atmospheric and Low Pressure Storage Tanks.
- Chapter XIX Inspection for Accident Prevention.
- Chapter XX Inspection for Fire Protection.
- Std. 620 Recommended Rules for Design and Construction of Large, Welded, Low Pressure Storage Tanks.

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- RP-2003 Recommended Practice for Protection Against Ignitions Arising Out of Static, Lightning and Stray Currents.
- 2521 Use of Pressure–Vacuum Vent Valves for Atmospheric Pressure Tanks to Reduce Evaporation Loss.
- 2523 Petrochemical Evaporation Loss from Storage Tanks.

### 8. Steel Structures Painting Council, Ref. SSPC-Vis 1–67, Standards and Practices:

- No. 1 “Pictorial Surface Preparation Standards for Painting Steel Structures.”

Occupational Safety and Health Administration (OSHA) Regulations Environmental Protection Agency (EPA Regulatory section), *Metals Handbook*, ASM International.

- Volume 1 Properties and Selection: Irons and Steels (latest edition).
- Volume 2 Properties and Selection: Nonferrous Alloys and Pure Metals (latest edition).
- Volume 4 Heat Treating (1981).
- Volume 8 Mechanical Testing (1985).
- Volume 9 Metallurgy and Microstructures (1985).
- Volume 11 Failure Analysis and Prevention (1986).
- Volume 13 Corrosion (latest edition).
- Volume 17 Nondestructive Evaluation and Quality Control (1989).

### 9. Instrument Society of America, Standards and Practices:

- RP1.1-7 Thermocouples and Thermocouple Extension Wires.
- RP3.1 Flowmeter Installations. Seal and Condensate Chambers.
- RP3.2 Flange Mounted Sharp Edged Orifice Plates for flow Measurement.
- RP4.1 Uniform Face-to-Face Dimensions for Flanged Control Valve Bodies.
- RP4.2 Standard Control Valve Manifold Designs.
- S5.1 Instrumentation Flow Plan Symbols.
- RP7.1 Pneumatic Control Circuit Pressure Test.
- RP7.2 Color Code for Panel Tubing.
- RP8.1 Instrument Enclosures for Industrial Environments.
- RP12.1 Electrical Instruments in Hazardous Atmospheres.
- RP12.2 Intrinsically Safe and Non-Incendive Electrical Instruments.
- S12.4 Instrument Purging for Reduction of Hazardous Area Classification.
- RP18.1 Specifications and Guides for the Use of General Purpose Annunciators.
- RP20.1 and 20.2 Specification Forms for Instruments.

### Federal Safety Standards for Pipelines; Part 195—Transportation of Liquids by Pipelines:

Often the process design engineer will become involved in managing a project, especially if he/she designed the specifications for fabrication and purchase of the equipment for the project. It is necessary that the process engineer participates in equipment layout/arrangement decisions from the early stages of the plant development. With all this background, the process engineer is the logical person to handle or coordinate the interrelationships of the various engineering disciplines, to review and evaluate the equipment purchase proposals from the purchasing department. The role of a project engineer often grows from the process design engineer's responsibilities [3].

The process engineer should be responsible for understanding the following regulations:

### 1. Occupational Safety and Health Administration regulations as they relate to (a) safety of design related to injury to personnel (includes such matters as latest vessel design [15], noise level

from operating equipment, and so on. [16–24]). (b) safety of the plant layout environment, which might influence the safety of the plant facilities.

2. Environmental Protection Agency regulations related to air, water, solid waste, and land contamination with toxic substances that a plant might emit/release into immediate plant area, or discharge as waste into public streams, or inject into underground aquifers, or dump or store [25–27].

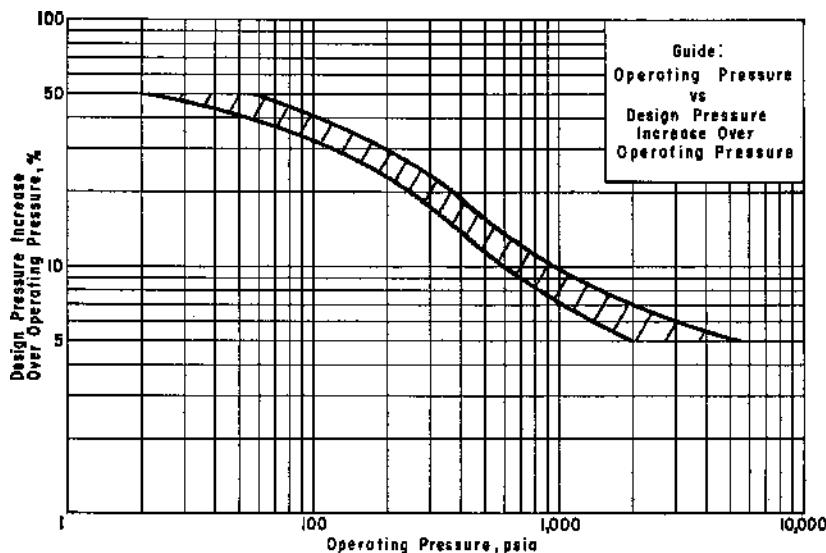
## 1.17 SYSTEM OF UNITS

Although the US chemical industry is committed to converting from American Engineering Standard units to the metric standards, or SI units, the actual progress in this conversion has been slow. This is primarily due to the fact that engineers are more familiar with the “more practical” engineering units, and also few textbooks using SI units are available. The conversion in the industry is awkward and confusing because there is no “feel” for the practical meaning of the SI terms. However, it is desirable for industry worldwide to adopt one consistent set of units, such as SI, although it is unlikely to happen for many years; the designer must contend with whatever system, or combination of systems, his/her organization adopts. For those in the contracting industry, the designer must adopt whatever system of units the client requires. It is usually the best practice to work through design calculations in the units in which the result is to be presented; but if working in SI units is preferred, data can be converted to SI units. *Tables of conversion factors to the SI system from most of the scientific and engineering units used in process design calculations are available in the companion website. The SI units will be incorporated in some of the process design calculations in this book.*

## 1.18 SYSTEM DESIGN PRESSURES

In order to coordinate the design pressures for the various vessels in a given process system, it is necessary to establish the relationship between the operating and the design conditions. Figure 1-40 and Tables 1-9 and 1-10 are guides to setting the percentage for the design pressure over the operating pressure. This type of relationship can be established according to the preferences of the responsible engineer or company policy. In the range near atmospheric pressure the preferences vary, however, for small diameter (less than 8 ft) vessels operating in a definite pressure system. Thus, the effect of a reasonable overpressure for design (as suggested by Figure 1-40) on the vessel wall thickness is usually negligible. For the larger diameter storage vessels operating with a few ounces of water to 1 psig, the selection of a design pressure must also consider the system surges in relation to the normal conditions. For example, a storage tank 20 ft in diameter, which will operate less than 6 oz of water, might be designed for 12 oz while arbitrarily selecting a design pressure of 10 psig would be quite uneconomical. A 40-foot diameter tank for atmospheric storage would normally be designed for 2–3 oz of water (see previous listing of API codes and ASME codes). The bottom shell of a 40-ft diameter × 40-ft tall vessel must withstand the greater pressure of the height of water or process liquid when the vessel is full to the vents. For a 6-ft diameter vessel operating at 3 psig, a reasonable design pressure might be 10 psig.

In some low-pressure processes it is good practice to set a minimum design pressure of 10 psig or 25 psig for all vessels operating below 5 psig and no larger in diameter than 8–10 ft. The minimum design pressures for a vessel will be influenced by the fact that the minimum vessel wall thickness for carbon steel is usually 3/16–1/4 in. Economics of the situation dictate where the



**Figure 1-40** Guide: operating pressure vs design pressure increase over operating pressure.

**TABLE 1-9 Suggested System Design Pressures (Based upon Condensation at 100° F with 10° F Approach)**

System	Design Pressure (psig)
Freon – 11 (or equivalent)	50
Freon – 12 (or equivalent)	200
Freon – 22 (or equivalent)	300
Ammonia	250
Chlorine	300

cutoff pressures and/or diameters lie, as these will vary with the type of metal under consideration.

Vessels operating below atmospheric pressure are designed for full vacuum regardless of the actual vacuum. If it is extremely uneconomical to design at this point, then proper vacuum control must be installed. However, this is not the usual approach to the design. If the equipment can operate alternately under vacuum or positive pressure, it must be designed for the worst or controlling condition.

**TABLE 1-10 Suggested Maximum Operating Pressure**

System	Usual Condition of Maximum Operating Pressure
Refrigeration Systems	Refrigerant vapor pressure at temperature 10–15° F above condensing water.
Storage Vessels	Vapor pressure of liquid at maximum ambient temperature plus 30° F (usually 110–140° F)
Process Vessels	Depends upon operating conditions, surge conditions, insulation, toxicity, explosion hazard, and so on.
A. In a compressor or pump system: (1) Centrifugal Type (2) Reciprocating Type	Shut-off pressure plus 5 psi Normal operating pressure plus 15 psi for low pressures to plus 50 psi for 200–3000 psi system.
B. Direct Injection of Steam, Air, Methane, Cooling Water, and so on.	Supply line pressure plus 5–15 psi.

## 1.19 TIME PLANNING AND SCHEDULING

Scheduling of work in process engineering or design is a near impossibility as far as pinpoint accuracy is concerned. The very developmental and planning nature of the early phases, as well as the continuous follow-through and follow-up, makes this difficult. One can seldom foresee specific changes, delays, and so on. Very few projects are clear-cut and well-defined ("frozen") as to scope or design conditions except for small jobs and repeat or duplicate projects.

With new processes and/or products, the collection of physical property data (either from pilot or laboratory operations, or from the literature) and consideration and evaluation of alternate conditions and flow schemes with the corresponding decisions often become a significant portion of the time required to complete the actual process calculations and preparation of design specifications. So that this early phase of work does not unnecessarily slow down the project, it is important that close guidance and supervision be given to the individual designer and the use of experience, judgment, and approximations (e.g., *rules of thumb* or *heuristics*) be encouraged. In this way many unnecessarily detailed or time-consuming calculations can be avoided, or reduced to a reasonable minimum.

On the other hand there are many situations which require the detailed work before a sound decision can be made. Additionally, it is often necessary to obtain reasonably accurate prices for various items of equipment and their assembly before the final decision can be made.

For groups specializing in this type of design work, it is good to maintain records of the time requirements, job conditions, and

so on in order to build a history upon which to base future estimating. It will be recognized that no two projects or problems are exactly alike. However, with time certain basic similarities can be recognized, with good judgment these records can be used to advantage. Thus, average information can have some value. The size of a project does not always have a significant bearing on the schedule. Weighted judgment, taking the type of job, type of process, and type and nature of the men with the engineering and process responsibility into account is necessary to align a balanced and smooth working team.

### ACTIVITY ANALYSIS

A time study of eight graduate process engineers with a minimum of five years experience is shown in Tables 1-11 and 1-12. The time includes process calculations, preparation of specifications, discussions with vendors, and handling the complete scope of small and large projects, and is helpful in accounting for legitimate time which was obviously not spent in performing process calculations. This does not include total project coordination or project engineering (for expanded reference also see [27]).

It should be recognized that the data in these tables may not necessarily fit other situations; however, it can serve as a guide. Since it is based upon engineers associated with an engineering department located at an operating company plant site, there is a basic difference in contacts, availability of production experience, and perhaps even philosophy between this type of group and one centered at an engineering office remote from plant contacts. The interruptions and requirements for data and results although similar in many respects are certainly different in other respects. The use of this type of activity information will be combined with detailed calculation data and discussed later.

**TABLE 1-11 Time Study**

Activity of Engineers	Percent of Single Study	Time (Avg. Range)
Process design calculations	34.69	35-52
Conferences, consultation, unscheduled urgent assignments, information assembly	28.98	13-29
Supervision and administrative, including time schedules, discussions with salesmen, preparation of outside correspondence	4.45	4-15
Preparation of charts, forms, methods for benefit of overall group	1.95	1-3
Marketing, checking, and reviewing flowsheets (no drafting)	10.94	9-12
Group meetings, training periods, overall department and company development	1.80	1-3
Literature review (current magazines, and so on)	1.80	0.5-2
Coffee breaks, and so on	5.55	4-6
Unaccounted, including vacation	9.84	5-10

**TABLE 1-12 Time Study**

Activity of Engineers	Percentage of Time
Consulting outside of scheduled jobs	4.4
Section supervision duties	4.7
Meetings related to scheduled jobs	13.7
Discussion with vendors	2.6
Special technical assignments	2.4
Communications within section	5.9
Process design calculations (original)	51.0
Process design calculations (checking)	3.7
Equipment schedules, line schedules, and so on	3.1
Flowsheet development, checking, revising (no drafting)	2.5
Coffee breaks, miscellaneous activity	6.0
Total	100.0

### COLLECTION AND ASSEMBLY OF PHYSICAL PROPERTY DATA

An important but time-consuming factor in practically every design situation and in development of flowsheets is the collection and assembly of physical property data for the components of the system in question. Often it is not sufficient to obtain single data points from various tables, since many designs cover rather wide ranges of temperature and pressure and the effects of these on the properties must be taken into account.

Data may be located in many useful handbooks as well as published technical papers and company compilations. However, experience indicates that extensive literature searches may be necessary to locate specific data on a particular compound. It is surprising to find so many common compounds for which the data are incomplete and sometimes inaccurate. Empirical correlations must often be utilized, sometimes to generate a value and sometimes to check a questionable literature value.

Therefore, when developing an estimate of process engineering time required, it is important to recognize the amount of effort that may be necessary to collect physical property data before any real work can commence. This same concern exists when evaluating *K* values and activity data for systems. Chapter 3 reviews physical property data of many components within the minimum and maximum temperature range.

### ESTIMATED EQUIPMENT CALCULATION MAN-HOURS

The required man-hours for a specific calculation vary with the process system, availability of physical property data, and the relative familiarity of the process design engineer. Records collected over a period of years on a wide cross-section of organic and inorganic process equipment calculations are summarized in Table 1-13. It is impossible to accurately define the limits of the calculations represented, but on an average they have been found to be helpful in establishing the *order of magnitude* of the calculation time, as well as the basis for approximating the overall extent of the process engineering of the project.

Electronic computers, both digital and analog, can be used to great advantage in design studies and calculations. In evaluating reactor designs it is extremely helpful to develop a family

**TABLE 1-13 Estimated Man-Hours Required for Equipment Design (updated)**

Type of Equipment	*Design	**Computer	†Checking	Total	M-H
<b>HEAT EXCHANGERS:</b>					
Solvent cooler	30	—	3	33	
Tank heating coil	4	—	2	6	
Caustic cross exchanger	32	(1)	6	38	
Caustic cooler	8	(2)	2	10	(2)
Oil cross exchanger	32	(3)	5	37	(3)
Gas cooler	8	(3)	4	12	(3)
Compressor gas aftercooler	8	(2)	1	9	(2)
Slurry cooler	32	(4)	8	40	(4)
Finned tube exchanger	15	—	4	19	
Gas cooler	4	(1)	1	5	(1)
<b>CONDENSERS:</b>					
Steam	7	(2)	—	7	(2)
Organic	6	(2)	2	8	(2)
HCl organic	10	(5)	11	21	(5)
Organic	6	(2)	2	8	(2)
Finishing	4	(1.5)	1.5	5.5	(1.5)
<b>PARTIAL CONDENSERS:</b>					
Organic-air	10	(3)	2	12	(3)
Organic-air	20	(3)	4	24	(3)
Organic-air	30	(4)	14	44	(4)
Inorganic-air	50	(4)	20	70	(4)
<b>REBOILERS (THERMOSIPHON):</b>					
Organic-steam	16	—	—	16	
Organic-steam	20	(3)	5	25	(3)
Organic-steam	14	(3)	—	14	(3)
Organic-steam	10	(3)	5	15	(3)
Organic-steam	16	(3)	6	22	(3)
Organic-steam	14	(3)	5	24	(3)
Organic-steam	4	(0.5)	—	4.5	(1.5)
Organic-steam	5	(1)	—	6	(1)
Organic-steam	4	(1)	—	5	(1)
Organic-steam	5	(0.5)	1	6	(0.5)
Organic-steam	5	(0.5)	1	6	(0.5)
<b>REBOILERS (FORCED CIRCULATION):</b>					
Organic-steam	25	(4)	10	35	(4)
Organic-steam	19	(4)	8	27	(4)
Organic-steam	6	(1)	3	9	(1)
Organic-steam	6	(2)	3	9	(2)
<b>DISTILLATION (PACKED):</b>					
Carbonating tower	25	(4)	10.5	35.5	(4)
Gas cooler	20	(4)	8	28	(4)
Gas cooler	25	(4)	7	32	(6)
Cooling	16	(5)	22	38	(5)
Gas scrubber	24	(6)	4	28	(6)
Gas scrubber	12	(6)	8	20	(6)
Vent gas scrubber	5	(2)	1.5	6.5	(2)
<b>DISTILLATION (TRAY):</b>					
Organic-tray-by-tray	50	(12)	25	75	(12)
Organic-tray-by-tray	40	(12)	64	104	(12)
Demethanizer-tray-by-tray	31	(15)	22	53	(15)
Organic-tray-by-tray	35	(7)	4	46	(7)
Organic-tray-by-tray	10	(5)	5	15	(5)
Organic-tray-by-tray	5	(6)	3	11	(6)
Organic-tray-by-tray	2	(2)	2	4	(2)
De-ethanizer	24	(12)	15	39	(12)
Demethanizer	30	(15)	15	45	(15)
Organic – includes tray layout	24	(15)	28	52	(15)
Organic – includes tray layout	38	(20)	10	48	(20)

(continued)

**TABLE 1-13—(continued)**

Type of Equipment	*Design	**Computer	† Checking	Total	M-H
PUMPS					
System	8	(2)	6	14	(2)
Single	1.5	(2)	0.5	2	(2)
Single	1	(1)	1	2	(1)
Single	3	(1)	3	6	(1)
RECIPROCATING COMPRESSOR:					
BHP, temperature and	3	—	—	3	
Data for vendor rating	6	—	2	8	
CENTRIFUGAL COMPRESSOR: About the same as Reciprocating above.					
PROCESS LINE SIZES:					
Single	1	(0.5)	0.5	1.5	(0.5)
Single	0.5	(0.5)	0.5	1	(0.5)
System, 22 lines	20	(3)	9	29	(3)
Air header for plant	4.5	(3)	2	6.5	(3)
SAFETY VALVES:					
Single	2	—	2	4	
Single	1	—	1	2	
STEAM TRAPS:					
System of 4	3	—	1	4	
Single	0.75	—	0.25	1	
Single	1	—	1	2	
MISCELLANEOUS TANKS, DRUMS, ETC.:					
Condensate level drum	0.5	—	0.5	1	
Steam flash drum	6	—	3	9	
Storage tank	2	—	—	2	
MATERIAL BALANCES:					
Depends on size of system.					
<i>Note: The man-hours listed in this table included collection of needed physical property and other data and preparation of a specification or summary of the requirements.</i>					

(Source: Ludwig [2].)

\* Using desk electronic calculators, not programmed.

\*\* Programmed computer. Represents data input plus calculation time, sometimes multiple.

† Checking only for "Design" calculations.

( ) Alternate calculations by programmed computer.

of performance curves for variables involved in the system. Usually this type of calculation becomes too time-consuming with the desk electronic calculator, and is a good problem for the computer.

After investing time and talent into a program for the computer, it is usually only a matter of minutes or hours before a complete series of results can be calculated. Computers are quite adaptable to the following calculations: distillation tray-by-tray and short-cut methods; tray hydraulics for bubble cap, sieve or perforated and "dual flow"; absorption, heat exchanger including condensation, partial condensation, cooler-condensing, reboiling; drying; compression; equilibrium flash; fluid flow (e.g., incompressible and compressible), including two-phase and many others. It is important to remember that good results cannot be obtained from a poor or inadequate computer program. Thus, it is wise to invest the effort into the development of basically sound general-purpose programs. With these many variations can be arranged to suit the special case. In order to have confidence in the results of any computer program (whether self-developed or purchased) it must be audited and tested against extreme conditions or limits. To purchase and use a program without testing is inviting errors.

Some programs require only a few days to be completed for general-purpose use, while some others require several months of continuous effort. Whenever more than one individual is expected to use the computer program, it is good practice to obtain the several views on attacking the problem, that is, type of input data, solution approach, range of variables, fixed conditions, and type and form of output or results.

Table 1-14 illustrates some reasonable time requirements for solution of problems or designs when using a medium-sized digital computer, using existing programs. A very high speed machine might reduce the pure calculation time to a matter of minutes; however, the time required for (1) data collection (specific problem conditions as well as physical property data, (2) data input to the computer, and (3) evaluation of results and preparation of design specification sheets all remain essentially fixed. In some situations the complexity of the calculations requires the capabilities of the large machines, and in these cases the time advantage can be the difference between a good result and none at all. Total plant material and heat balances are a good example. This can be accomplished by a flowsheeting software (e.g., ASPEN, CHEMCAD, HYSYS, PRO II).

**TABLE 1-14 Time Calculation Using Medium-sized Digital Computer**

Calculation	Total elapsed time preparation + calculation (h)
Preliminary distillation	
Number of trays, reflux ratio	1–3
Tray-by-tray distillation	
to 40 trays	2–3
to 100 trays	3–5
Trays hydraulics	
Bubble cap, sieve, perforated	1–3
Heat exchangers	
Condenser, exchangers	0.5–1.0
Separators	0.5
Flash vaporization	0.5
Oil absorbers	1–3
Safety valves	0.25

**ESTIMATED TOTAL PROCESS MAN-HOURS**

After the man-hours have been estimated for all of the individual items of equipment on the project, a guide to total man-hours is the following.

$$\text{Total estimated job process} = \frac{\text{Estimated equipment man-hours (including checking)}}{\text{engineering man-hours}} \quad 0.45$$

This applies to work where at least 50% of the time is calculated by electronic desk calculator for the numerical calculations. It may not apply well to projects of less than 200 process man-hours.

When limited time is available to complete a project, this may be used to determine the estimates of manpower:

$$\text{Average number of men} = \frac{\text{Estimated man-hours (process)}}{(\text{Total elapsed weeks}) (30–33)}$$

Where 30–33 represents the actual usable job-related man-hours per 40-h week per man, allowing for average sickness, vacation, jury duty, and so on.

$$\text{Approximate maximum number required} = (1.67) (\text{Avg number of engineers})$$

**EXAMPLE 1-1****Man-Hour Evaluation**

From an examination of the process flowsheet the total man-hours is 685\* for the significant equipment. Items such as steam traps and miscellaneous small time-items can be omitted from the total.

$$\text{Total estimated job man-hours} = \frac{685}{0.45} = 1525$$

If the work must be completed, including flowsheet supervision, and so on, in 3 weeks,

$$\text{Average no. of engineers required} = \frac{1525}{(3)(32)} = 15.9$$

This is impractical since a job of this magnitude cannot be planned and decisions reached in this time. Therefore, the men could not be kept busy. It will be necessary to spread out the time, using fewer engineers.

For a 12-week program:

$$\text{Average no. of engineers required} = \frac{1525}{(12)(32)} = 3.97 \cong 4$$

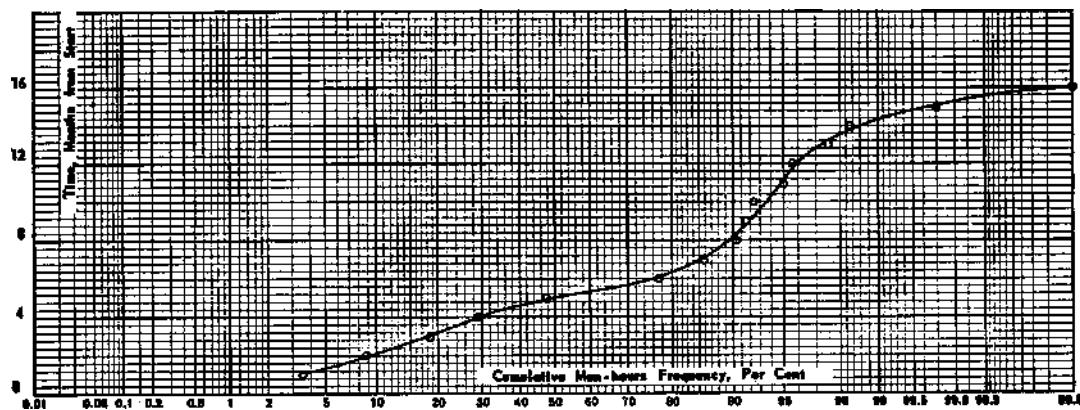
Peak manpower  $\cong 3.97 \times 1.67 \cong 6.6$ , use 7 men

Near peak manpower requirements will be needed when 30–50% of the total time scheduled has been crossed, unless other factors influence the timing.

**TYPICAL MAN-HOUR PATTERNS**

Figures 1-41a–c illustrate accumulation patterns for the process engineering man-hours of a few typical projects. In general the smaller the project and the better defined the scope, the more the pattern of Project B is approached. Projects A and C represent the larger projects where there may be changes in plant capacity or location, as well as a concurrent pilot plant research program to continually obtain a better answer. The slowdown portions of the

\* Includes 75 man-hours for pipeline sizing.



**Figure 1-41a** Process engineering man-hours accumulation pattern: Project A. (By permission from Ludwig [2].)

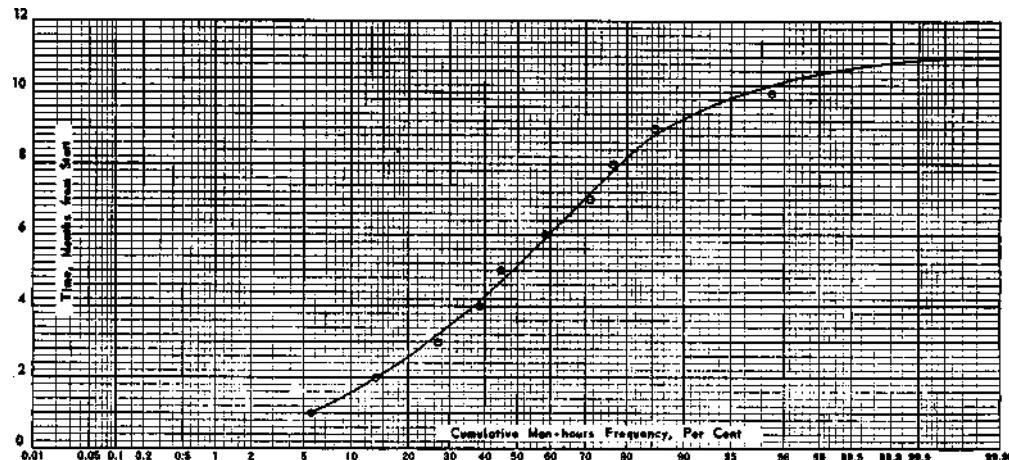


Figure 1-41b Process engineering man-hours accumulation pattern: Project B. (By permission from Ludwig [2].)

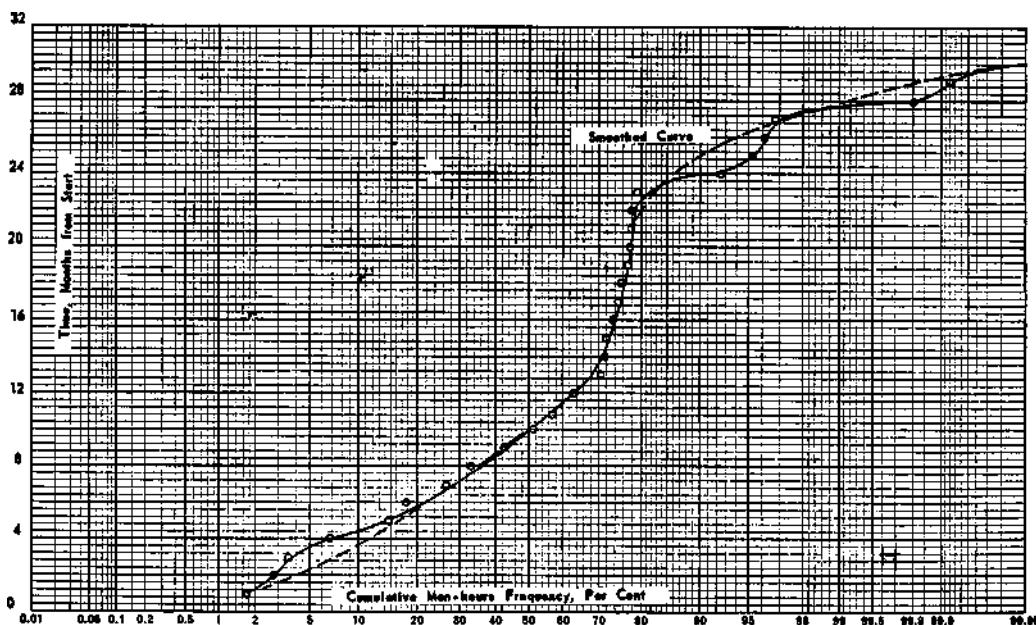


Figure 1-41c Process engineering man-hours accumulation pattern: Project C. (By permission from Ludwig [2].)

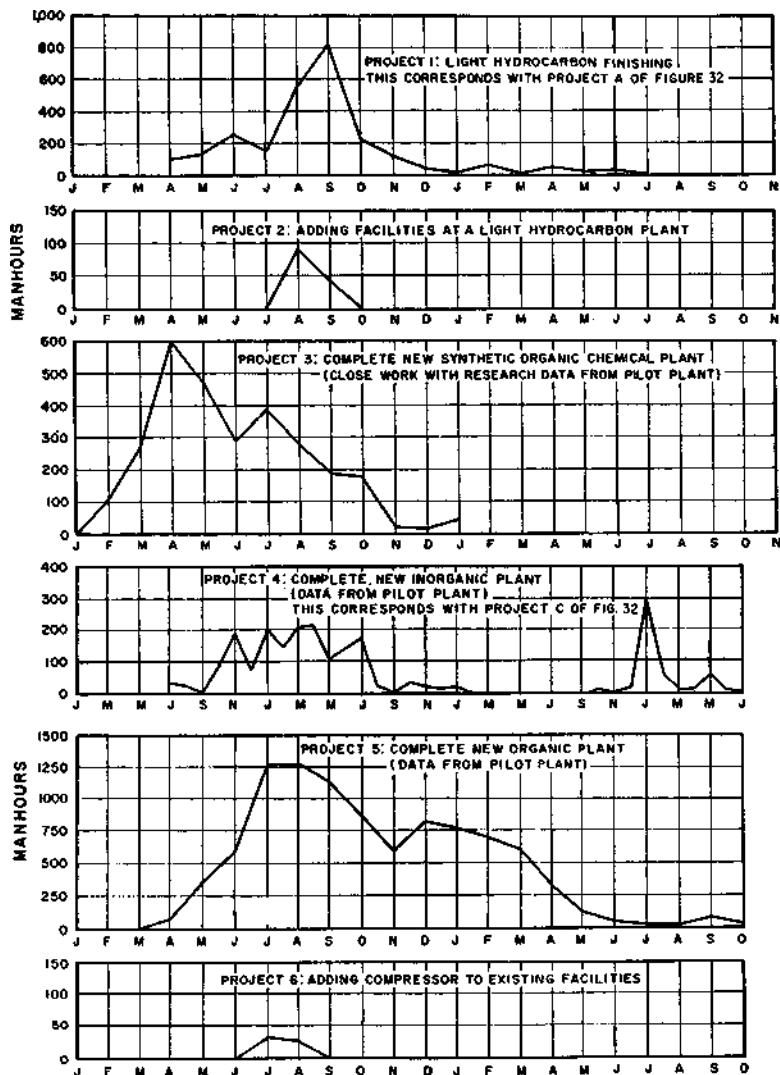
curves can be accounted for as significant changes in the process or process-related factors. In general, most large (6 months or longer) process engineering projects undergo significant changes by the time 50% of the project has been completed. These changes may not be setbacks, but they are reflected in the ability of the project to properly utilize the available engineering manpower in the "normal" manner.

Figure 1-42 presents some typical monthly requirements of process engineering for projects of different magnitudes. In some organizations, the schedule is set by the available manpower, and does not always represent all that could be accomplished if a limitless supply of qualified manpower were available.

A summary of process engineering costs as they are related to total erected plant costs is shown in Figure 1-43. The process engineering man-hour requirements are related to total engineering man-hours for the project in Figure 1-44. These data are based on the operation of a complete process engineering section in the engineering

department of a relatively large petrochemical plant complex. Since the assignment of responsibility varies with company policy and types of processes, this information is reasonably valid only for the particular plant relationship. It should establish order-of-magnitude information for other related operations. By studying the progress history of the individual projects, the major deviations from a so-called "average straightforward job" can be recognized.

Figure 1-45 is reasonably typical of fixed-fee costs as charged by contract engineering organizations. The top curve representing the total engineering and related costs include complete process engineering, equipment specifications, flowsheets, detailed complete plant drawings, purchasing, and expediting. The lower curve represents only the process engineering including material and heat balances, equipment specifications, flowsheets, plot plan and elevations, and cost estimate. The middle curve covers the balance of all engineering detailing, purchasing, and expediting.



**Figure 1-42** Process engineering manpower requirements by project and by months from start. (By permission from Ludwig [2].)

In some cases, they may be anticipated by knowledge of the status of the process data prior to the start of engineering activity. The larger projects are somewhat easier to group than the smaller ones. Process engineering is not always handled as completely for the small jobs. This is to say that flowsheets may be simplified, detailed equipment, and Line Schedules may not be required, and the overall project can be completely visualized at the outset, which is not the case with large projects.

Figure 1-46 illustrates that for average capital expenditures of \$10–30 million per year covering the very small hundred thousand dollars to very large \$5–8-million projects, the process engineering work leads the expenditures in a somewhat regular pattern by about three calendar quarters. This actual lead interval is a function of a company policy in scheduling its projects. The curves are believed representative for an aggressive program.

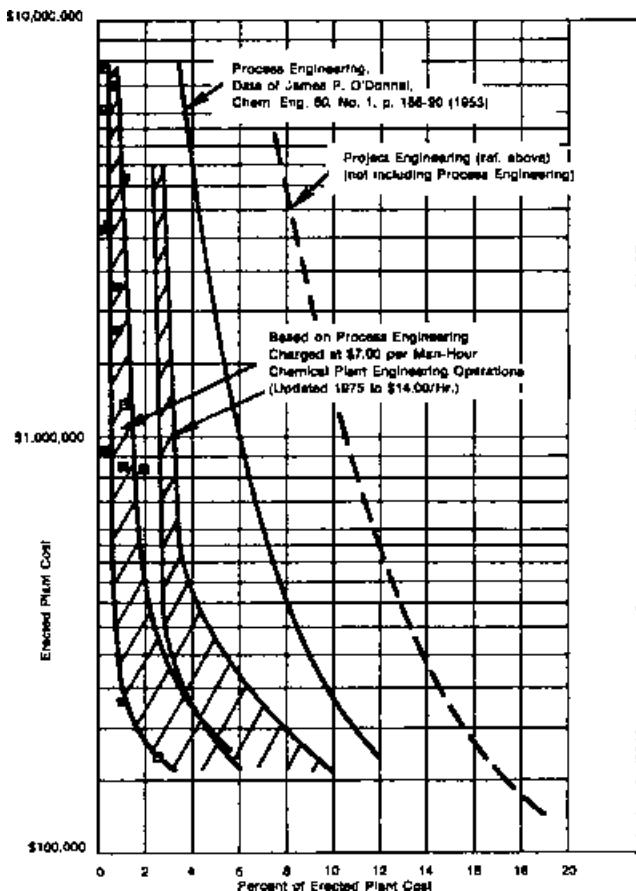
## INFLUENCES

The principal factor, which runs the process engineering man-hours over expected time for a “straight-through” project is the compar-

ison studies of equipment or process schemes when compared to the relatively simple and limited work after the decisions are made.

Any rushed program of process engineering development will usually be inefficient in manpower for certain parts of the work. Thus lead time for proper thinking and evaluation of significant process schemes and types of equipment will usually be reflected in efficient handling for the project when the bulk of the general engineering manpower is assigned to the detailed work. When decisions are made at the time of the need, all concerned can produce to the most benefit of the project. If the basic process can be designed and the flowsheet approved prior to initiating the detailed mechanical, structural, and electrical engineering, the project usually runs well throughout the department. This situation is more likely to occur in a contractor organization than an engineering department of a producing company.

The schedule of projects must often be adjusted to reflect the influence of the key decision-maker assigned to the work. If he/she requires complete detailed figures before reaching any decision, time will necessarily be consumed. On the other hand if he/she applies judgment and experience to the basic factors (less details), then the overall direction of the project can be continually pointed



**Figure 1-43** Process engineering costs (1975), based on process engineering charged at \$14 per man-hours. Chemical plant engineering operations include flowsheet development and drafting, material and heat balances, equipment designs, ratings, checking, and bid reviews and selection of equipment. (By permission from Ludwig [2].)

in a profitable direction in the minimum of time. In reality actual "multipliers" are often applied to the time schedule of a project to reflect the type of decision-maker involved.

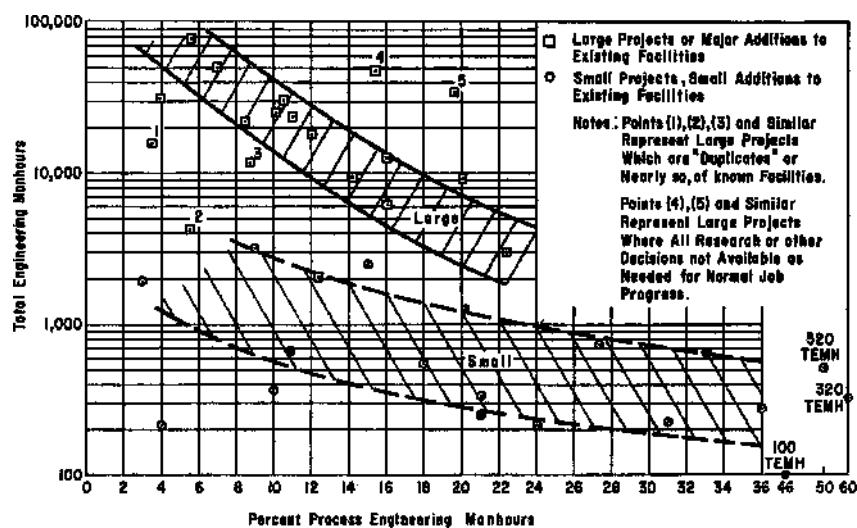
#### ASSIGNMENT OF PERSONNEL

It is important to plan ahead for the proper assignment of qualified engineers to various projects as they arise. Jobs cannot be assigned on an unconsidered basis; that is, each lead or principal process engineer and others in his/her group on a project must be selected for their (1) basic ability to understand the process under consideration, (2) background know-how, (3) design ability for the equipment involved, and (4) compatibility with the project engineer and other key decision-making representatives with whom they will be in daily contact.

There are two approaches to developing qualified personnel:

1. *The generalist approach* – each process engineer becomes competent overall with preferential areas of specialization. With this approach, all personnel are urged to study and keep up to date in order to handle any type of project. This simplifies the assignment of the men, since there are more chances of having some available who are relatively strong in the needed specialties of a particular job. This does not require that projects or specific designs be lined up waiting for the specialist. With overall good general knowledge by each member of the group there is better appreciation for the exchange of views and understanding of specific design problems.
2. *The specialist approach* – each process engineer becomes a specialist in one or more related fields. Even in this arrangement some overall general process engineers are needed to cover and tie together the areas handled by the specialist. Each specialist becomes an expert in a single field or, if reasonable, in a broad range of related topics. Each problem for design or study for every project of the particular type passes to the specialist for detailed handling of design, specification, and evaluation. His/her work passes into the project and he/she turns to another assignment on the same or some other project.

In general, the specialist may often be much more of an expert in a particular subject under this system than under the generalist



**Figure 1-44** Process engineering man-hours for new construction or major additions to existing facilities and small projects. (By permission from Ludwig [2].)

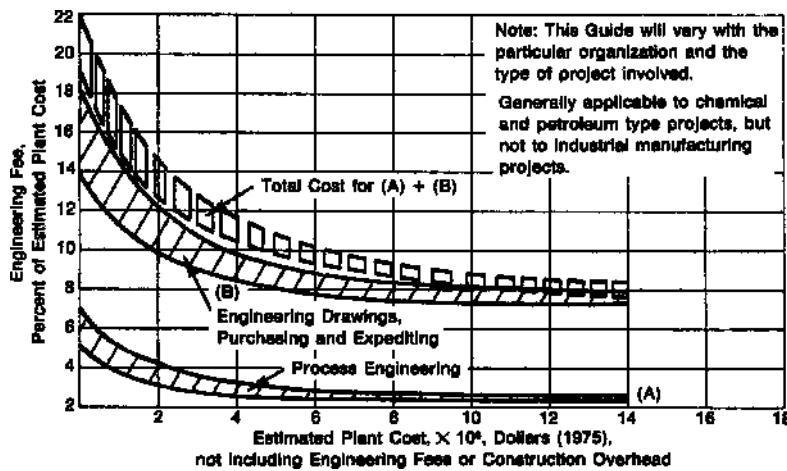


Figure 1-45 Estimating engineering fees.

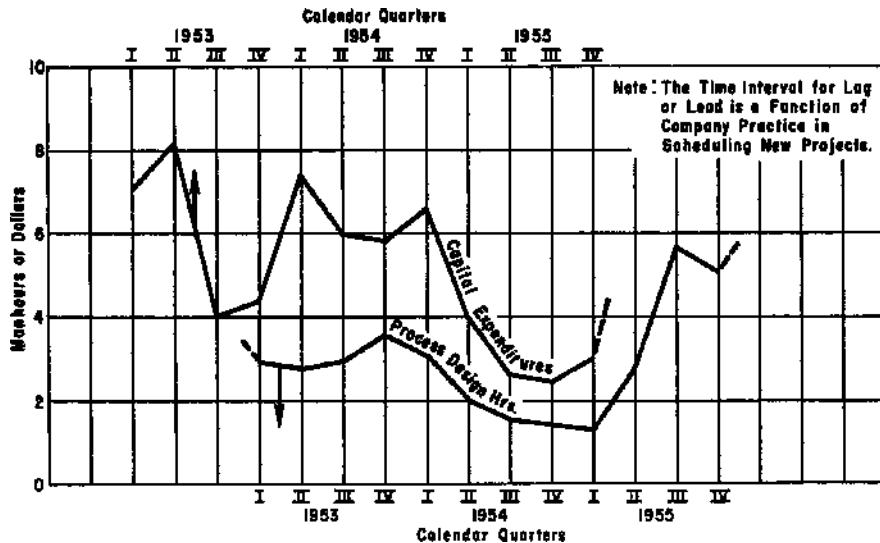


Figure 1-46 Process design man-hours versus capital expenditures.

approach, and consequently the depths of the pertinent factors of a problem may come to light for evaluation. By contrast, it is easy to make a career of even a small assignment when the field of interest is narrow and the overall project perspective is not clearly in view.

When the workload is low, it is important to have other assignments for these men. This is the time to develop standards for

1. design of various types and items of equipment;
2. methods of practice and general details;
3. electronic computer programs for these design standards;
4. evaluation of field data.

These should all be viewed from the long range and with repetitive value to the group effort. The individuals who develop these standards in effect become specialists if they handle the assignment in good detail.

## 1.20 PLANT LAYOUT

The final plant layout combines the various engineering considerations for soil conditions: drainage; railroad, truck and services

access; receiving raw materials; waste materials removal; effect of climate on outdoor versus indoor operations and on types of structures; prevailing wind direction for vent as well as climate moisture; corrosion; plant expansion and growth; access to public; and many other general evaluation points. From these broad considerations the details are developed to suit the particular plant process and the combined effects of the location.

The process engineer has an important responsibility in site selection as well as plant layout, since many of the decisions regarding physical location of buildings and associated equipment require knowledge of what is taking place in the operation as well as the hazardous factors of explosion, fire, toxicity, and so on. The process engineer is usually called upon to describe the process requirements and limitation to the other engineers – civil, structural, mechanical, electrical, and instrument. By progressively discussing the process others can note the requirements, which might affect the normal or routine design approach to each phase of the project. This review must not be limited to the design aspects of the engineering but rather must describe how the plant is to operate and how product is to be shipped, stored, and so on.

**TABLE 1-15 Layout and Process Development Engineering Checklist**

<b>SITE (ASSUMES SITE SELECTED)</b>
1. Ground contour and its relation to general orientation of buildings and equipment.
2. Drainage and waste disposal, details to prevent erosion.
3. Set plant elevations: floor elevations of buildings and bottom of steel footings for equipment and large storage tanks.
4. Location of any existing or probable locations for new railroads, roads, power lines and power sources, telephone lines, water supply, residential, and/or industrial buildings or structures.
5. Legal Requirements and Permits.
a. Rights of way for pipe crossing of road, highway, railroad, rivers, canals, etc.
b. Easements of pipe lines, power lines, etc.
c. C.A.A. approval on airports, and for construction and painting of structures in certain areas in airport vicinity.
d. Underground storage wells for chemical and hydrocarbon products.
e. Railroad approval of road crossings, additions to existing facilities, automatic railroad gates, required state, and railroad clearances.
f. Navigable stream requirements and permits.

**CLIMATE**

1. Prevailing wind; locate hazardous vents, burning flares, waste burning pits, waste settling ponds down-wind of plant proper.
2. Nature of climate. Consider seasonal and daily temperature variations, dust, fog, tornados, hurricanes, earthquakes. Define duration of conditions for design. Determine from US Weather Bureau yearly statistics for above, as well as rainfall. Establish if conditions for earthquakes, hurricanes prevail. For stormy conditions, structural design for 100 mi/h winds usually sufficient. For hurricanes, winds of 125 mi/h may be design basis.
3. Corrosion. Plants located close (within 100 ft) to seas, oceans, bays, lakes encounter more severe corrosion than if located one-fourth mile or more away. Some highly industrial areas are more corrosive than rural or non-industrial locations. Additional details are presented by Mears [28].
4. Pollution of Air and water. Determine allowable limits for atmospheric vent as well as liquid wastes. Consider neutralization. Determine federal, state, and local regulations and effect of climatic conditions on dispersion.

**UTILITIES AND RAW MATERIALS**

1. Sources and methods of transportation and packaging.
  - a. Water: potable, service, brackish, sea or ocean, cooling tower.
  - b. Steam: condensate disposal, feed-water make-up
  - c. Gas: (1) Process; may not be odorized  
(2) Fuel; odorized
  - d. Oil: fuel, lubrication (or Liquefied Petroleum Gas)
  - e. Air: (1) Utility  
(2) Instrument; must be dry below lowest equivalent dew point to prevent moisture condensation and freezing.
  - f. Power
2. Warehouse receiving and storage: drum, boxes, carboys for raw processing materials as well as laboratory control and testing chemicals.

**TABLE 1-15—(continued)**

<b>PRODUCT SHIPMENTS</b>
1. Conditions for pipe line transfer of product to user or customer.
2. Warehouse conditions for bagging, boxing, crating, palletizing, and methods of shipment (trailer truck, box car, tank car, hopper or special car). Consider in-transit and turnaround time to determine number in use.

**GENERAL LAYOUT**

1. Use of models
2. Maintenance considerations associated with each building, process area and equipment. Consider (a) access for cranes and trucks (b) work space for local repairs (c) operating conditions of adjacent parts of process to allow local repairs.
3. Initial construction sequence and problems.
4. Materials of construction for buildings.
5. Roads: paving, width.
6. Basic pattern for concrete, gravel or asphalt paving or work floors in operating and adjacent areas.
7. Fencing.
8. Plant guard or security system.

**ELECTRICAL AND FREE HAZARDS**

1. Define plant areas handling hazardous and lethal materials and set rules for design considerations, such as ventilation explosion, walls, etc. Flammable storage materials may require enclosed dikes, foam systems and the like. Refer to National Board of Fire Underwriters or specific insurance company to coordinate recommended protection. Attaway [29] has details on many points to consider.
2. Define plant areas requiring explosion-proof, drip-proof and Open motor and associated electrical components. Refer to National Electrical Code and National Electrical Manufacturer's Association Standards.
3. Define areas and building to use wet and dry sprinklers systems, foam systems, location to hand and hose fire extinguishers, fire carts, fire engines.
4. Define location of fire walls, fire hydrants.
5. Review layout for fire equipment access, and secondary and emergency exit roads from each area.
6. Review entire fire and other hazards program with insurance representatives. Industrial insurance companies have excellent facilities for evaluating the associated problems.

**Safety Requirements**

1. Special design problems for emergency handling of dangerous or lethal materials.
2. Safety as it is reflected in factors of safety in design of pressure vessels, pressure testing of piping and vessels, use of API, ASME and ASA Codes; Code Stamps on equipment.
3. Areas requiring safety showers and eye wash stations.
4. Design and selection philosophy for use of safety devices for pressure relief and alarm.
5. Inside block valves on acid and caustic storage vessels.
6. Emergency power and other facilities to control safe operation or shut-down.

**FUTURE GROWTH**

1. Define areas of future growth and associated space requirements.
2. Correlate future expansion plans to required utilities and raw materials as related to economics of required installation.
3. Consider spare equipment, present and future.

After the project begins to take shape and preliminary layouts of the overall as well as sections of the plant are partially completed, design work by the other phases of engineering will require the answering of questions as well as the evaluation of details of a particular phase as they are related to the process performance. Some useful considerations for selected details are given by Thompson [30] and Ludwig [2]. A general checklist of factors, which usually needs reviewing for the proper layout considerations of chemical and petrochemical plants, is given in Table 1-15.

There are many other factors which affect project planning as it is related to process engineering. However, these are usually peculiar to the process or objective of the project. On first glance some of the items listed in Table 1-15 may appear to be unrelated to the process engineering requirements, and this can be the case for some types of projects. In these situations they become more of a project engineering responsibility. However, in many cases these have a relationship either to the process engineering requirements or the decisions which must take this into account.

## 1.21 RULES OF THUMB ESTIMATING

The quality of the solution of an engineering problem depends on the experience, data, and time that can be devoted to it. In many cases, the best possible solutions are absolutely necessary. However, in some circumstances less than the best is acceptable or preferred.

Tentative solutions may be finals in themselves or they may serve as starting points for detailed analysis. They may originate on the basis of related experience or analogy with known situations. Another class of solutions is called engineering rules of thumb, which are outright statements regarding suitable sizes or performances of equipment that may obviate all need for calculations. Such rules often apply to features of equipment that are arbitrary or whose improvement is likely to result in only marginal economic benefit. Rules of thumb may afford a possible course of action when circumstances do not warrant development of the optimum course of action.

Everyday activities are also governed to some extent by rules of thumb or shortcut methods. They serve us when we wish to take a course of action but are not in a position to find the optimum course of action. These shortcut calculations methods and guidelines are essential for [31]:

1. checking new process designs;
2. providing equipment size and performance estimates;
3. helping to troubleshoot problems with operating systems;
4. verifying the reasonableness of results of computer calculations and simulations;
5. providing reasonable initial values for process simulation parameters in order to achieve program convergence;
6. obtaining approximate costs for process units;
7. developing preliminary process layouts;
8. pipe sizing and insulation requirements.

These short-cut methods are forms of rules or heuristics that are helpful to the practicing engineer. However, all heuristics can be erroneous and sometimes difficult to ascertain. They merely serve as plausible aids or directions toward the solution of any design problem. The rules of thumb or heuristics as presented by Walas [32] are illustrated in Chapter 0.

## NOMENCLATURE

$a_{i,j,k}$	Coefficients
$A, B, C, D$	Regression constants
$f_1, f_2, f_3, f_4$	Functions
$g_{11}, g_{21}, g_{31}, g_{32}$	Functions
$k_i$	Number of input streams to unit $i$
$N_D$	Number of degrees of freedom
$N_{\text{Equations}}$	Number of independent equations
$N_{\text{Externally Defined}}$	Number of externally defined variables
$N_{\text{Variables}}$	Number of variables
$N_{\text{Manipulated}}$	Number of manipulated variables
$S_1, S_2, S_3, S_4$	Streams
$\Delta H_f^\circ$	Enthalpy of formation, kJ/kmol
$\Delta H_r$	Enthalpy of reaction, kJ/kmol

## ABBREVIATION

AIChE	American Institute of Chemical Engineers
ANSI	American National Standard Institute
API	American Petroleum Institute
ASME	American Society of Mechanical Engineers
BSI	British Standard Institute
CAD	Computer-Aided Design
EPA	Environment Protection Agency
DCS	Distributed Control Systems
ISA	Instrument Society of America
OSHA	Occupational Safety and Health Administration
P&ID	Piping and Instrumentation Diagram
ROI	Return on Investment

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## COST ESTIMATION AND ECONOMIC EVALUATION

### 2.1 INTRODUCTION

Recently, mergers and downsizing in the various chemical process industries (CPIs) have required management to make effective decisions regarding investments in strategic assets, sometimes with limited engineering participation. Due to limited capital budgets, potential projects are scrutinized stringently before funds are allocated to proceed with them; management requires reasonably accurate cost estimates at each stage of the funding. The choice as to which estimation method to use at each stage depends on the information available at the time of preparation, its desired accuracy, and the end use of the estimate.

Many industries use some form of classification system to identify the various types of estimate that may be prepared during the life cycle of a project, and also to indicate the overall maturity and quality of the estimates produced. However, the process industries in general and individual companies and organizations have not been consistent and tend not to have a firm understanding of the terminology used during classification. The Association for Advancement of Cost Engineering International (AACE) recently developed recommended practices for cost estimate classification for the process industries [1]. Table 2-1 shows a list of recommended practices, which are intended to be the principal products and services requiring AACE certification.

The AACE 18R-97 is the recommended-practice cost estimate classification for the process industries. It is a reference document that provides extensions and additional detail for applying the principles of estimate classification specifically to project estimates for engineering, procurement, and construction (EPC) work. It also describes and differentiates between various types of project estimate by identifying five classes of estimates as shown in Table 2-2. Class 1 estimate is associated with the highest level of project definition or maturity; a Class 5 estimate is assigned with the lowest level. Five characteristics are used to distinguish one class of estimate from another, namely degree of project definition, end use of the estimate, estimating methodology, estimating accuracy, and effort to prepare the estimate. However, the degree of project definition is the primary characteristic used to identify an estimate class [2].

The CPI depends on process flow diagrams (PFDs) and piping and instrumentation diagrams (P&IDs) as the primary scope-defining documents. These documents form the principal deliverables in determining the level of project definition, the maturity of the information used to perform the estimate, and subsequently the estimate class. An estimate input checklist, including items such as PFDs, process and utility equipment lists, and instrumentation and control system diagrams, is incorporated into AACE 18R-97, which identifies the engineering deliverables used to prepare a project estimate. Further information on ACCE 18R-97 can be accessed at [www.aacei.org](http://www.aacei.org).

### 2.2 CAPITAL COST ESTIMATION

Capital cost estimation is an essential part of investment appraisal. Many types of capital cost estimates are made, ranging from initial order-of-magnitude estimates to detailed estimates which require

**TABLE 2-1 AACE-Recommended Practices for Cost Estimate Classification**

Documents	Recommended Practices
Cost Engineering Terminology	10S-90
Required Skills and Knowledge of a Cost Engineer	11R-88
Model Master's Degree Program with Emphasis in Cost Engineering	12R-89
Recommended Method for Determining Building Area	13S-90
Roles and Duties of a Planning and Scheduling Engineer	14R-90
Profitability Methods	15R-81
Conducting Technical and Economic Evaluations in the Process and Utility Industries	16R-90
Cost Estimation Classification System	17R-97
Cost Estimation Classification System – As Applied in Engineering, Procurement, and Construction for the Process Industries	18R-97
Estimate Preparation Costs in the Process Industries	19R-97
Project Code of Accounts	20R-98
Project Code of Accounts – As Applied in Engineering, Procurement, and Construction for the Process Industries	21R-98
Direct Labor Productivity Measurement as Applied on Construction and Major Maintenance Projects	22R-01
Planning and Scheduling – Identification of Activities	23R-02
Planning and Scheduling – Developing Activity Logic	24R-03
Estimating Lost Labor Productivity in Construction Claims	25R-03

the collection of accurate technical data. The American Association of Cost Engineers (AACE) defines five types of cost estimates as follows:

1. *Order-of-magnitude estimate (ratio estimate)* is based on similar previous cost data, where an approximate forecast of fixed investment can be determined without flowsheet, layout, or equipment analysis by applying overall ratios (to account for differences in scale of production), and applying appropriate escalation factors to update from previous installations. The probable accuracy of estimate is within  $\pm 30\%$ .
2. *Study estimate (factored estimate)* is based on knowledge of major equipment items. This is applicable when a scheme has been developed to the stage of preliminary flowsheets, with the duty rating of principal items of equipment specified, and a geographical location for the construction of the facility is known. A cost estimate can be prepared based on estimates for each main plant item or group of items. The factored estimate accuracy is within  $\pm 30\%$ .

**TABLE 2-2 Cost Estimate Classification Matrix**

Estimate Class	Project Definition (% of Complete Definition)	Purpose of Estimate	Estimating Method	Accuracy Range (Variation in Low and High Ranges)	Preparation Effort (Index Relative to Project Cost)
Class 5	0–2	Screening	Capacity-factored, parametric models	L: –20 to –50% H: 30 to 100%	1
Class 4	1–15	Feasibility	Equipment-factored, parametric models	L: –15 to –30% H: 20 to 50%	2–4
Class 3	10–40	Budget authorization or cost control	Semi-detailed unit cost estimation with assembly-level line items	L: –10 to –20% H: 10 to 30%	3–10
Class 2	30–70	Control of bid or tender	Detailed unit-cost estimation with forced, detailed take-off	L: –5 to –15% H: 5 to 20%	4–20
Class 1	50–100	Check estimate, bid or tender	Semi-detailed unit cost estimation with detailed take-off.	L: –3 to –10% H: 3 to 15%	5–100

The state of process technology and the availability of cost data strongly affect the accuracy range of an estimate. Plus or minus high (H) and low (L) values represent the variation in actual costs vs estimated costs, after applying contingency factors. The “preparation effort” uses an index to describe the cost required to prepare an estimate, relative to that for preparing a Class 5 estimate. For example, if it costs 0.005% of the project cost to develop a Class 5 estimate, then a Class 1 estimate could require as much as 100 times that, or 0.5% of the total project cost.

(Source: Dysert, L., By permission of Chem. Eng., Oct 2001, pp. 70–81.)

3. *Preliminary estimate (sanction, budget authorization estimate, or scope estimate)* is applied after a study estimate has been accepted. The further engineering work, which has been authorized, aims to obtain and present information such as preliminary material and energy balances, P&IDs, equipment lists and material specifications, duty rating and sizing of all process equipment, and instrumentation and control devices. The cost estimate which may be achieved will have a probable error of less than  $\pm 20\%$ . For several items such as insulation, electrical services, instrumentation, and pipework, estimates are derived by applying factors to the estimated costs of the main equipment items.
4. *Definitive estimate (project control estimate)* is based on almost complete data, but before completion of drawings and specifications. Initially derived from project funds allocation, which is based upon the preliminary stage estimate and the authorization to proceed, estimates can be refined as design work proceeds and decisions are made. A definitive estimate can be determined shortly before complete drawings and specifications have been discussed with vendors. The estimate is aimed to achieve a probable error in the limited range of  $\pm 10\%$ .

5. *Detailed estimate (tender or contractor's final cost estimate)* requires completed engineering drawings, specifications, and site surveys. Process and mechanical datasheets are prepared and vendors' quotations for selected equipment items are obtained such that competitive prices, which are compatible with quality and delivery, are available for the estimate. The probable accuracy of estimate is within  $\pm 5\%$ . Each type of estimate and its probable range of accuracy are summarized in Table 2-3.

Process plant designs commence with preliminary designs based on approximate technical data, calculations, and cost data and proceed to final designs that require detailed and accurate data, calculations, and quotations. Cost estimates of a proposed plant are continuously carried out during the development of a process from laboratory to

**TABLE 2-3 Probable Accuracy Relative to the Cost of the Estimate**

Recommended Nomenclature	Probable Range of Accuracy (%)	Cost as % of Project Expenditure (%)
Detailed estimate	$\pm 2$ to $\pm 5$	5 to 10
Definitive estimate	$\pm 5$ to 15	1 to 3
Preliminary estimate	$\pm 10$ to 25	0.4 to 0.8
Study estimate	$\pm 20$ to $\pm 30$	0.1 to 0.2
Order of magnitude estimate	$\pm 30$ to $\pm 50$	0 to 0.1

(Source: Gerrard [3].)

construction. The total capital cost,  $C_{TC}$ , of a project consists of the fixed capital cost,  $C_{FC}$ , and the working capital,  $C_{WC}$ , plus the cost of land and any other non-depreciable assets,  $C_L$ . This is given by

$$C_{TC} = C_{FC} + C_{WC} + C_L \quad (2-1)$$

The fixed capital cost is the capital required to provide all the depreciable facilities;  $C_{FC}$  may be divided into two classes known as the battery limits and auxiliary facilities. The boundary of battery limits includes all manufacturing and processing equipment. The auxiliary facilities are the storage areas, administration offices, utilities, and other essential and non-essential supporting facilities.

Generally, as the size of the project under consideration increases, the cost of preparing an estimate tends to decrease as a percentage of project total cost. Table 2-3 illustrates a summary of the expected accuracy ranges and approximate costs of the preparation of estimates for the various classifications. The cost of preparing an estimate exceeding the bands shown in Table 2-3 depends on uncertainty of basic data, inadequate

management associated with the project, development in new areas of technology, frequent changes of scope of work, and so forth. The cost of preparing cost estimates is dependent not only on the nature and scale of the project but also on the experience and overhead structure of the organization preparing the estimate [3].

### 2.3 EQUIPMENT COST ESTIMATIONS BY CAPACITY RATIO EXPONENTS

The process engineer often plays a key role in the preparation of cost estimates and in their development. From a first draft flow-sheet and a preliminary plot plan, a preliminary cost estimate can be prepared by the “factoring” or equivalent method. This basically accumulates the individual costs of each item of major equipment and then multiplies by an experience factor to produce one or all of (i) total plant cost installed with or without overhead costs, (ii) piping installed, and (iii) equipment installed. For accuracy, these factors must be developed from actual plant costs, and are often peculiar to a specific construction type or engineering approach to the project. The factor of 2.5–6.0 usually covers most petrochemical processing plants. This factor times the costs of major equipment (pumps, compressors, tanks, columns, exchangers) but not instruments will give total plant costs. The plant will include usual control buildings, structure, foundations, overhead charges, construction fees, engineering costs, and so on. A value of 4.0 may usually be realistic [4].

The process designer must be aware of costs, as reflected in the (i) selection of a basic process route, (ii) the equipment to be used in the process, and (iii) the details incorporated into the equipment. The designer must not arbitrarily select equipment, specify details, or set pressure or temperature levels for design without recognizing the relative effect on the specific cost of an item as well as associated equipment such as relieving devices, instruments, and so on. [4].

With more comprehensive and better information regarding the process and layout plans, estimating engineers can prepare detailed estimates, which are often quite accurate, usually  $\pm 10\%$  for the best. It is the duty of the process designer to supply the best information in order to contribute to better or improved estimates.

Estimating equipment costs is a specialty field in itself. The estimator must therefore have access to continuously updated basic reference costs and to graphical costs relations, which are a function of the capacity of the equipment. Page's *Estimator's Manual of Equipment and Installation Costs* [5] is a helpful reference. Since the equipment is only a portion of the total cost of a plant, or an addition to a chemical project, installation costs, which reflect the labor portion of the total cost, must also be determined. Useful and comprehensive data for such needs are presented for equipment [5], general construction [6], heating, air-conditioning, ventilating, plumbing [7], piping [8], electrical-related areas [9], and all disciplines [10] in the cited references.

Even an inexperienced estimator should be able to establish an approximation of the costs, provided he/she adequately visualizes the work functions and steps involved. From the same type of work reference, an experienced estimator can develop a realistic cost, usually expressed with certain contingencies to allow for unknown factors and changing conditions. Professional estimators will normally develop cost charts and tables peculiar to the nature of their responsibilities and requirements of their employers.

It is often necessary to calculate the cost of a piece of equipment when there are no available cost data for the particular size or capacity. If the cost of a piece of equipment or plant size or

capacity  $Q_1$  is  $C_1$ . The cost  $C_2$  of a similar piece of equipment or plant size or capacity  $Q_2$  can be calculated from the equation

$$C_2 = C_1 \left( \frac{Q_2}{Q_1} \right)^m \quad (2-2)$$

where

$C_1$  = cost of plant or section of plant of original capacity “1”

$C_2$  = cost of plant or section of plant of new capacity “2”

$Q_1$  = capacity of plant or section of original requirements

$Q_2$  = capacity of plant or section of new requirements.

$m$  = cost exponent (or capacity factor)

the value  $m$  depends on the type of equipment or plant. It is generally taken as 0.6, the well-known six-tenths rule [11]. This value can be used to get a rough estimate of the capital cost if there are insufficient data to calculate the index for the particular size of equipment required. Tables 2-4 and 2-5 list values of  $m$  for various types of equipment and products, respectively. The value of  $m$  typically lies between 0.5 and 0.85, depending on the type of plant, and must be assessed carefully for its applicability to each estimating situation. The  $m$  used in the capacity factor equation is actually the slope of the log-curve that has been drawn to reflect

**TABLE 2-4 Typical Exponents for Equipment Cost vs Capacity**

Equipment	Exponent ( $m$ )
Reciprocating compressor	0.75
Turbo blowers compressor	0.5
Electric motors	0.8
Evaporators	0.5
Heat exchangers	0.65–0.95
Piping	0.7–0.9
Pumps	0.7–0.9
Rectangular tanks	0.5
Spherical tanks	0.7
Towers, constant diameter	0.7
Towers, constant height	1.0

(Source: Institution of Chemical Engineers [12].)

**TABLE 2-5 Capacity Factors ( $m$ ) for Process Units**

Product	Factor
Acrylonitrile	0.60
Butadiene	0.68
Chlorine	0.45
Ethanol	0.73
Ethylene oxide	0.78
Hydrochloric acid	0.68
Hydrogen peroxide	0.75
Methanol	0.60
Nitric acid	0.60
Phenol	0.75
Polymerization	0.58
Polypropylene	0.70
Polyvinyl chloride	0.60
Sulfuric acid	0.65
Styrene	0.60
Thermal cracking	0.70
Urea	0.70
Vinyl acetate	0.65
Vinyl chloride	0.80

(Source: Guthrie [13].)

**TABLE 2-6 Percentage Error (in %) if Capacity Factor ( $m$ ) of 0.7 is Used for Estimate**

Actual Exponent	Capacity-increase Multiplier ( $Q_2/Q_1$ )							
	1.5	2	2.5	3	3.5	4	4.5	5
0.20	23	41	58	73	88	100	113	124
0.25	20	36	51	64	75	87	97	106
0.30	18	32	44	55	64	74	83	91
0.35	16	28	38	47	55	63	70	76
0.40	13	23	32	39	46	52	57	63
0.45	11	18	26	32	36	41	46	50
0.50	9	15	20	25	28	32	35	38
0.55	6	11	15	18	21	23	25	28
0.60	4	7	10	12	13	15	16	18
0.65	2	3	5	6	6	7	8	8
0.70	0	0	0	0	0	0	0	0
0.75	-2	-4	-5	-5	-6	-7	-7	-8
0.80	-4	-7	-9	-10	-12	-13	-14	-15
0.85	-6	-10	-13	-15	-17	-19	-20	-21
0.90	-8	-13	-17	-20	-22	-24	-26	-28
0.95	-10	-16	-21	-24	-27	-29	-31	-33
1.00	-11	-19	-24	-28	-31	-34	-36	-38
1.05	-13	-22	-28	-32	-36	-39	-41	-43
1.10	-15	-24	-31	-36	-40	-43	-45	-47
1.15	-16	-27	-34	-39	-43	-46	-49	-52
1.20	-18	-30	-37	-42	-47	-50	-53	-55

(Source: Dysert, L., By permission of Chem. Eng., Oct 2001, pp. 70–81.)

the change in the cost of a plant as it is made larger or smaller. These curves are typically drawn from the data points of the known costs of completed plants. With an exponent less than 1, scales of economy are achieved such that as plant capacity increases by a percentage, say, by 20%, the costs to build the larger plant increases by less than 20%. With an exponent of 0.6, doubling the capacity of a plant invariably increases costs by approximately 50%, and tripling the capacity of a plant increases costs by approximately 10% [2].

Dysert [2] reported that as plant capacities increase, the exponent tends to increase, and as the plant capacity increases to the limits of existing technology, the exponent approaches a value of 1. At this point, it becomes theoretically as economical to build two plants of a similar size rather than one large plant on the same specific site. Table 2-6 shows the percent error that may occur if an assumed capacity factor of 0.7 is used, and the actual value is different. For example, if the new plant is triple the size of an existing plant, and the actual capacity factor is 0.85 instead of the assumed 0.7, the cost of the new plant is underestimated by only 15%. Similarly, for the same threefold scale-up in plant size, if the capacity factor should be 0.6 instead of the assumed 0.7, then the plant cost is overestimated by only 12%. Dysert [2] generated these data from Eq. (2-2); the capacity-increase multiplier is  $Q_2/Q_1$  and in the base  $m$  is 0.7. The error occurs as  $m$  varies from 0.7.

Cost indices should be used to derive the cost data of a desired year. A cost index is a value for a given point in time showing the cost at that time relative to a certain base time. If the cost at some time in the past is known, the equipment cost at the present time can be determined from the equation. This is applicable for any given year of installation but does not correct for the differences in cost from year to year. This is conveniently done as described in the next section.

Experience has indicated that this six-tenths rule is reasonably accurate for capacity scale-up of individual items of equipment. Thus, if the cost of one size of a piece of equipment is known, an estimating figure for one twice as large can be found by multiplying by  $2^{0.6}$ .

The most difficult feature of this method is that for each type of plant or plant product, as well as for each type of equipment, there is a break point where the 0.6 no longer correlates the change in capacity. For small equipment or plants in reasonable pilot or semi-works size, the slope of the cost curve increases and the cost ratio is greater than 0.6, sometimes 0.75, 0.8, or 0.9. From several cost values for respective capacities a log-log plot of capacity versus cost will indicate the proper exponent by the slope of the resultant curve. Extrapolation beyond eight- or ten-fold may usually introduce unacceptable error [4].

## 2.4 YEARLY COST INDICES

The three most-used cost indices for the chemical, petrochemical, and refining industries for relating the cost level of a given year or month to a reference point are the following:

1. *Chemical Engineering Plant Cost Index* [10]. Probably the most commonly used cost-adjusting index printed/updated monthly is in the *Chemical Engineering* magazine and has established continuity over many years. Its breakdown component costs apply to plants and plant equipment/systems. This can be accessed from the website: [www.che.com](http://www.che.com)
2. *Marshall and Swift Equipment Cost Index* [14]. Commonly used for process industry equipment, and the index numbers presented by industries in the *Chemical Engineering* magazine on a monthly basis.
3. *Nelson Farrar Cost Index* [15]. This is generally suited to petroleum refining plants and is referenced to them. It is updated and published regularly in the *Oil & Gas Journal*. It can be accessed from the website: [www.ogj.com](http://www.ogj.com)

These indices are used to update costs when values at some earlier date are known. The new costs are of estimating accuracy and should be verified whenever possible, as should the results of using

**TABLE 2-7 International Plant Cost Indices (Mar 1992–Feb 1993, Base date Jan 1990 = 100)**

Country	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Jan	Feb
Australia	107.7	107.9	107.9	108.2	108.4	108.6	108.7	108.8	108.8	109.0	109.3	109.4
Belgium	108.6	109.2	109.3	109.2	110.4	110.8 <sup>R</sup>	110.8	111.4	111.2	111.2	111.2	111.2
Canada	107.0	107.0	106.9	106.3	105.7	106.3	106.9	107.5	107.5	108.7	108.8	108.8
Denmark	107.0	108.4	108.5	109.3	110.4	108.0	108.5	108.9	108.5	110.4	110.7	—
France	106.5	106.9	107.0	107.1	107.5	107.5	108.1	108.3	108.5	108.6	109.2	109.5
Germany	113.8	114.0	114.0	114.9	115.0	115.0	115.4	115.4	115.4 <sup>F</sup>	116.2	116.3	116.4
Italy	109.5	109.6	109.6	109.9	110.1	110.0	110.2	110.5	110.5	110.7	—	—
Japan	103.0	102.8	102.8	110.4	102.7	101.3	103.1	103.3	103.4	102.5	100.3	103.0
Netherlands	106.8	108.0	108.1	108.1	108.8	108.8	108.8	108.7	108.7	108.7	109.5	109.5
New Zealand	103.9	104.4	104.4	104.4	105.0	105.0	105.0	105.3	105.4	105.4	—	—
Spain	111.1	113.2	113.2	113.3	114.4	114.4	114.4	114.7	114.8	114.8	—	—
Sweden	106.2	107.9	108.8	109.2	109.6	107.9	108.4	109.0	109.5	111.3	—	—
USA	101.7	101.1	101.6	101.4	100.4	100.8	100.7	100.9	100.5	100.7	100.7	100.7
UK	117.2	115.0	115.1	117.0	116.8	117.1	118.2	116.2	114.4	114.6	114.9	115.8

(Source: *Process Eng.*, Jun 1993.)<sup>F</sup> forecast value<sup>R</sup> revised value

the 0.6 power for correlating cost and capacity. The new cost can be obtained by

$$EC_2 = EC_1 \left( \frac{I_2}{I_1} \right) \quad (2-3)$$

where

$I_2$  = index value for year represented by 2 (usually current)

$I_1$  = index value for earlier year represented by 1

$EC_2$  = equipment estimated cost for year represented by 2

$EC_1$  = equipment purchased cost (when available) for year represented by 1.

Cost indices are used to give a general estimate, but no index can account for all the factors. Many different types of cost index are published in journals such as *Chemical Engineering*, *Engineering News-Record*, the *Oil & Gas Journal*, and *Process Engineering*. Table 2-7 shows the international plant cost indices from March 1992 to February 1993 and Table 2-8 gives the annual plant cost indices for the years 1963–2000 [16]. Figure 2-1 shows historic trends and values of cost indices pertinent to the chemical process industry. The Chemical Engineering Plant Cost Index (CEPCI) can

be used to account for changes that result from inflation, and is accessed from the website: [www.che.org](http://www.che.org).

#### EXAMPLE 2-1

A plant to produce 100,000 bbl/d of ethanol is to be completed in Houston in 2004. A similar plant with a capacity of 150,000 bbl/d and a final cost of \$50 million was completed in 2002. Determine the cost of the 100,000 bbl/d plant.

*Solution*

Using the capacity factor algorithm in Eq. (2-2), with  $m = 0.73$  gives:

$$\begin{aligned} C_2 &= C_1 \left( \frac{Q_2}{Q_1} \right)^m \\ C_2 &= \$50M \left( \frac{100,000}{150,000} \right)^{0.73} \\ &= \$37.2M \end{aligned}$$

#### EXAMPLE 2-2

A 100,000 bbl/day ethanol plant is to be completed in Houston in 2004. A similar plant in Indonesia with a capacity of 150,000 bbl/day and a final cost of \$50 million was completed in 2002. Estimate the cost of the 100,000 bbl/day plant in Houston.

*Solution*

A better estimate of the cost of the plant in Houston is obtained by adjustments for differences in scope, location, and time. The plant

in Indonesia includes piling, tankage, and owner's costs, which are included in the plant to be built in Houston. However, construction costs in Houston are expected to be 1.25 times those in Indonesia. Escalation will be included as a 1.06 multiplier from 2002 to 2004. Additional costs for the Houston plant (not included in the cost estimate of the Indonesian plant) will probably involve stringent pollution control measures.

The revised estimate is as follows:

Plant in Indonesia	= \$50M	Escalate to 2004 \$50M × 1.06	= \$53M
Deduct \$10M for piling, tankage, and owner costs	= \$40M	Capacity factor estimate	= \$53M $\left( \frac{100,000}{150,000} \right)^{0.73}$ = \$39.4M
Indonesia to Houston adjustment (\$40M × 1.25)	= \$50M	Add \$5M for pollution-control requirements	= \$44.4M

**TABLE 2-8 Annual Plant Cost Indices**

Year	Composite CE Index	Equipment	Construction Labor	Buildings	Engineering and Supervision
1963	102.4	100.5	107.2	102.1	103.4
1964	103.3	101.2	108.5	103.3	104.2
1965	104.2	102.1	109.7	104.5	104.8
1966	107.2	105.3	112.4	107.9	106.8
1967	109.7	107.7	115.8	110.3	108.0
1968	113.7	109.9	121.0	115.7	108.6
1969	119.0	116.6	128.3	122.5	109.9
1970	125.7	123.8	137.3	127.2	110.6
1971	132.3	130.4	146.2	135.5	111.4
1972	137.2	135.4	152.2	142.0	111.9
1973	144.1	141.8	157.9	150.9	122.8
1974	165.4	171.2	163.3	165.8	134.4
1975	182.4	194.7	168.6	177.0	141.8
1976	192.1	205.8	174.2	187.3	150.8
1977	204.1	220.9	178.2	199.1	162.1
1978	218.8	240.3	185.9	213.7	161.9
1979	238.7	264.7	194.9	228.4	185.9
1980	261.2	292.6	204.3	238.3	214.0
1981	297.0	323.9	242.4	274.9	268.5
1982	314.0	336.2	263.9	290.1	304.9
1983	317.0	336.0	267.6	295.6	323.3
1984	322.7	344.0	264.5	300.3	336.3
1985	325.3	347.2	265.3	304.4	338.9
1986	318.4	336.3	263.0	303.9	341.2
1987	323.8	343.9	262.6	309.1	346.0
1988	342.5	372.7	265.6	319.2	343.3
1989	355.4	391.0	270.4	327.6	344.8
1990	357.6	392.2	271.4	329.5	355.9
1991	361.3	396.9	274.8	332.9	354.5
1992	358.2	392.2	273.0	334.6	354.1
1993	359.2	391.3	270.9	341.6	352.3
1994	368.1	406.9	272.9	353.8	351.1
1995	381.1	427.3	274.3	362.4	347.6
1996	381.7	427.4	277.5	356.1	344.2
1997	386.5	433.2	281.9	371.4	342.5
1998	389.5	436.0	287.4	374.2	341.2
1999	390.6	435.5	292.5	380.2	339.9
2000	394.1	438.0	299.2	385.6	340.6

(Source: Vatavuk [16], W.M., By permission of Chem. Eng., Jan 2002, pp. 62–70.)

## 2.5 FACTORED COST ESTIMATE

The purchased cost of an item of equipment, free on board (FOB), is quoted by a supplier, and may be multiplied by a factor of 1.1 to give the approximate delivered cost. The factorial methods for estimating the total installed cost of a process plant are based on a combination of materials, labor, and overhead cost components. The fixed capital cost,  $C_{FC}$ , of a plant based on design can be estimated using the Lang factor method [17] given by the equation

$$C_{FC} = f_L \sum C_{EQ} \quad (2-4)$$

or

$$\text{Total plant cost (TPC)} = \text{total equipment cost (TEC)} \times \text{equipment factor} \quad (2-5)$$

where

$f_L = 3.10$  for solid processing

$f_L = 3.63$  for mixed solid–fluid processing

$f_L = 4.74$  for fluid processing

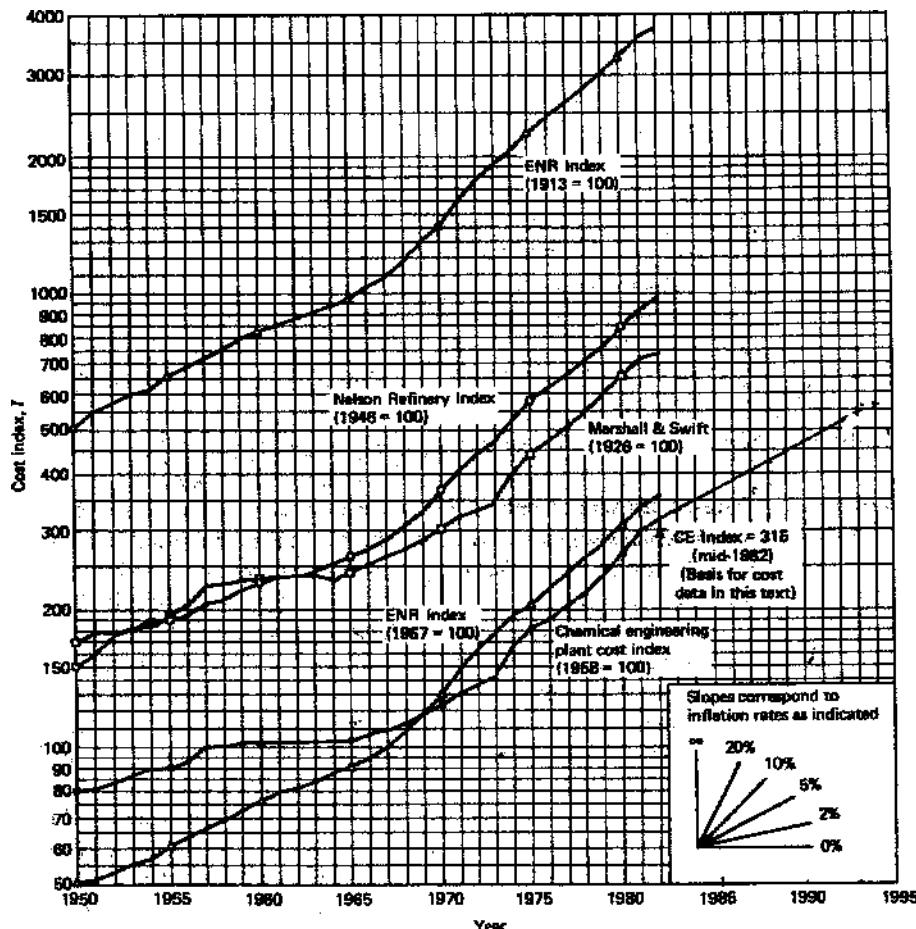
$\sum C_{EQ}$  is the sum of the delivered costs of all the major items of process equipment.

The major advantage of the Lang method is that it benefits from having the cost of equipment available, which is reflected in increased accuracy.

## 2.6 DETAILED FACTORIAL COST ESTIMATES

To increase accuracy, the cost factors that are compounded into the Lang factor are considered individually. Direct-cost items due to which cost is incurred in the construction of a plant, in addition to equipment items, are the following:

- Equipment erection, including foundations and minor structural work.
- Electrical-related, power and lighting, including stand-by provisions.
- Process buildings, including control rooms, and structures.
- Piping, including insulation and painting.
- Instruments, local and control room.
- Storages, raw materials, and finished products.



**Figure 2-1** Selected cost indices pertinent to chemical process construction.

- Ancillary buildings, offices, laboratory buildings, workshops, gatehouses.
- Utilities (services), provision of plant for steam, cooling water, air, firefighting services, inert gas, and effluent treatment (if not in plant costs), for example, lagoons, holding pits, process water supplies.
- Site, site preparation, landscaping, internal roads, and fencing.

Table 2-9 shows typical factors for the components of the capital cost, and these can be used to make an approximate estimate of it using equipment cost data published in the literature. In addition to the direct cost of the purchase and installation of equipment, the capital cost of a project will include the indirect costs listed in Table 2-9, which can be estimated as a function of direct costs [18].

Other methods for estimating capital investment consider the fixed-capital investment required as a separate unit. These are known as the functional-unit estimates, the process step scoring method, and the modular estimate.

The functional unit may be characterized as a unit operation, unit process, or separation method, which involves energy transfer, moving parts, or a high level of internals. The unit includes all process streams together with side or recycle streams. Bridgwater [19] proposed seven functional units, namely compressor, reactor, absorber, solvent extractor, solvent recovery column, main distillation column, and furnace and waste heat boiler. Taylor [20] developed the step counting method based on a system in which a

complexity score accounting for factors such as throughput, corrosion problems, and reaction time is estimated for each process step. The modular estimate considers individual modules in the total system, with each module consisting of a group of similar items. For the modular estimate, all heat exchangers are classified in one module, all furnaces in another, all vertical process vessels in another, and so on. The total cost estimate considers fewer than six general groupings. These are chemical processing, solids handling, site development, industrial buildings, offsite facilities, and project indirects [21]. Table 2-10 gives a more detailed explanation and definition of a functional unit.

The principle of the step count functional unit estimating methods is that the average cost of a functional unit in a process is a function of various process parameters:

$$\text{Capital cost per functional unit} = f(Q, T, P, M) \quad (2-6)$$

where

$Q$  = capacity or throughput

$T$  = temperature

$P$  = pressure

$M$  = material of construction.

A number of authors have published correlations based on a step counting approach: Zevnik and Buchanan [22], Taylor [20], Timms [23], Wilson [24], Bridgwater [25], and so on. These and other

**TABLE 2-9 Typical Factors for Estimation of Fixed Capital Cost of a Project**

Item	Process Type		
	Liquids	Liquids-Solids	Solids
1. Major equipment, total purchase cost	PCE	PCE	PCE
$f_1$ Equipment erection	0.40	0.45	0.50
$f_2$ Piping	0.70	0.45	0.20
$f_3$ Instrumentation	0.20	0.15	0.10
$f_4$ Electrical	0.10	0.10	0.10
$f_5$ Building, process	0.15	0.10	0.05
* $f_6$ Utilities	0.50	0.45	0.25
* $f_7$ Storages	0.15	0.20	0.25
* $f_8$ Site development	0.05	0.05	0.05
* $f_9$ Ancillary buildings	0.15	0.20	0.30
2. Total physical plant cost (PPC)			
$PPC = PCE (1 + f_1 + \dots + f_9) = PCE_x$	3.40	3.15	2.80
$f_{10}$ Design and engineering	0.30	0.25	0.20
$f_{11}$ Contractor's fee	0.05	0.05	0.05
$f_{12}$ Contingency	0.10	0.10	0.10
Fixed capital = $PPC (1 + f_{10} + f_{11} + f_{12})$ = $PC_x$	1.45	1.40	1.35

(Source: Sinnott [18].)

\* Omitted for minor extensions or additions to existing sites.

**TABLE 2-10 Explanation and Definition of a Functional Unit**

- A functional unit is a significant step in a process and includes all equipment and ancillaries necessary for operation of that unit. Thus, the sum of the costs of all functional units in a process gives the total capital cost.
- Generally, a functional unit may be characterized as a unit operation, unit process, or separation method that has energy transfer, moving parts, and/or a high level of "internals".
- Pumping and heat exchanger are ignored as they are considered as part of a functional unit unless substantial special loads such as refrigeration are involved.
- Storage "in process" is ignored, unless mechanical handling is involved – that is, for solids – as the cost of storage is relatively low and tends to be a constant function of the process. Large storages of raw materials, intermediates, or products are usually treated separately from "the process" in the estimate.
- Multi-stream operation is taken as one unit.
- Simple "mechanical" separation where there are no moving parts is ignored – that is cyclone, gravity settler – as the cost is usually relatively insignificant.

(Source: Gerrard [3].)

correlations have been reviewed and compared by Gerrard [3]. Correlations by some of these authors are as follows.

#### ZEVNIK AND BUCHANAN'S METHOD

$$CF = 2 \times 10^{(F_t + F_p + F_m)} \quad (2-7)$$

where

 $CF$  = complexity factors $F_t$  = temperature factor, read from a graph of  $F_t$  versus maximum (or minimum) process temperature $F_p$  = pressure factor, read from a graph of  $F_p$  versus maximum (or minimum) process pressure $F_m$  = materials of construction factor, read from a table of factors from 0 for mild steel and wood to 0.4 for precious metals.

Gerrard has updated the Zvenik and Buchanan's method to year 2000 with an adjusted *Engineering News-Record* (ENR) construction cost index.

For plant capacity above 10 million pounds per year (4536 tonnes/year) and temperature and pressure above ambient:

$$C = 7470 N Q^{0.6} 10^{[(0.1 \log P_{\max}) + (1.8 \times 10^{-4} (T_{\max} - 300)) + (F_m)]} \quad (2-8)$$

where

 $C$  = estimated capital cost, million pound, in 2000 $Q$  = plant capacity, tonnes per year $N$  = number of functional units $P_{\max}$  = maximum process pressure, atm $T_{\max}$  = maximum process temperature, K $F_m$  = materials of construction factor

0 for mild steel and wood

0.1 for aluminium, brass, lower grade stainless steel

0.2 for monel, nickel, higher grade stainless steel

0.3 for hastelloy

0.4 for precious metals.

For plant capacities below 10 million pounds per year (4536 tonnes/year), the equation is

$$C = 17,280 N Q^{0.5} 10^{[(0.1 \log P_{\max}) + (1.8 \times 10^{-4}(T_{\max} - 300)) + (F_m)]} \quad (2-9)$$

For sub-ambient temperatures, the temperature factor becomes

$$[0.57 - (1.9 \times 10^{-2} T_{\min})] \text{ in place of } [1.8 \times 10^{-4}(T_{\max} - 300)]$$

For sub-ambient pressures, the pressure factor becomes

$$0.1 \log \left( \frac{1}{P_{\min}} \right) \text{ in place of } [0.1 \log (P_{\max})]$$

Zevnik and Buchanan's definition of a functional unit refers to all equipment necessary to carry out a single significant process function.

### TIMMS' METHOD

This involves a simpler approach for gas phase processes only, including both organic and inorganic chemical products. The following equations have been updated from the original work by applying an adjusted US plant cost index and converting at \$2/£ to account for exchange rate and location effects.

$$C = 10560 N Q^{0.615} \quad (2-10)$$

with materials of construction, temperature, and pressure effects.

$$C = 4910 N Q^{0.639} F_m (T_{\max})^{0.066} (P_{\max})^{-0.016} \quad (2-11)$$

where

$C$  = capital cost in UK £<sub>2000</sub>, battery limits

$N$  = number of functional units

$Q$  = plant capacity, tonnes/year

$F_m$  = materials of construction factor

1.0 for carbon steel

1.15 for low grade stainless steel

1.2 for medium grade stainless steel

1.3 for high grade stainless steel

$T_{\max}$  = maximum process temperature, K

$P_{\max}$  = maximum process pressure, bar.

### BRIDGWATER'S METHOD

For processes with predominantly liquid and/or solid handling phases, the correlation equation is

$$C = 1930 N \left( \frac{Q}{s} \right)^{0.675} \quad (2-12)$$

where

$C$  = capital cost £<sub>2000</sub>, battery limits

$N$  = number of functional units

$Q$  = plant capacity, tonnes/year, above 60,000

$s$  = reactor "conversion" =  $\frac{\text{mass of desired product}}{\text{mass of reactor input}}$ .

For plant capacities below 60,000 tonnes/year

$$C = 169,560 N \left( \frac{Q}{s} \right)^{0.30} \quad (2-13)$$

Gerrard [3] developed a generalized approach based upon the principle that the capital cost is a function of a number of steps, and basic process parameters, particularly capacity or throughput, can be applied to any special situation to derive a model for that industry or group of processes.

The correlations are for refuse sorting and separating processes, and for non-biological effluent treatment.

(a) Refuse sorting and separating processes:

$$C = 3250 N (686 + Q) \quad (2-14)$$

where

$C$  = capital cost, £<sub>2000</sub>, battery limits basis

$N$  = number of functional units

$Q$  = plant capacity in tonnes of refuse feed per day.

(b) Non-biological effluent treatment:

$$C = 1900 N_e Q^{0.453} \quad (2-15)$$

where

$C$  = capital cost, £<sub>2000</sub>, battery limits basis, fully automated plant, buildings are not included

$Q$  = design throughput, imperial gal/h

$N_e$  = number of effluent treatment steps.

Effluent treatment steps are

- Acid/alkali neutralization
- Chrome reduction, aqueous (if gaseous sulfur dioxide is used, add a further half step)
- Cyanide oxidations to cyanate, aqueous (if gaseous chlorine is used, add a further half step)
- Demulsification
- Filter press
- Ion exchange
- Lime reagent preparation
- Settlement
- Water recycle system.

In SI units, the correlation is

$$C = 1750 N_e Q^{0.453} \quad (2-16)$$

where

$Q$  = throughput, m<sup>3</sup>/h.

The capital cost includes all equipment, materials, labor, civil installation, commissioning, and cubicle for the control panel. Reagent warehousing would cost about 20% more, and complete enclosure up to 100% more.

Equipment factored estimates (EFEs, Class 4) in Table 2-2 are typically prepared during the feasibility stages of a project, when engineering is approximately 1–15% complete. They are used to determine whether there is sufficient reason for funding the project. This estimate is used to justify the funding required to complete additional engineering and design for a Class 3 or budget estimate. The first steps when preparing an EFE are to estimate the cost for each item of process equipment, to examine the equipment list carefully for completeness, and to further compare it against the PFDs and P&IDs. However, the equipment list is often in a preliminary stage when an EFE is prepared and even when the major equipment is identified, it may be necessary to assume a cost percent for auxiliary equipment that remain to be identified.

Equipment is often sized at 100% of normal operating duty; however, by the time the purchase orders have been issued, some

**TABLE 2-11 Example of Equipment Factored Estimation**

Item Description	Equipment Cost (\$)	Equipment Factor	Total (\$)	Derived Multiplier
Columns	650,000	2.1	1,365,000	
Vertical vessels	540,000	3.2	1,728,000	
Horizontal vessels	110,000	2.4	264,000	
Shell-and-tube heat exchangers	630,000	2.5	1,575,000	
Plate heat exchangers	110,000	2.0	220,000	
Pumps, motors	765,000	3.4	2,601,000	
Raw equipment cost (TEC)	2,805,000			
Direct field cost (DFC)	$2,805,000 \times 2.8$		7,754,000	2.8
Direct field labor (DFL) cost	$DFC \times 25\%$		1,938,000	
Indirect field costs (IFC): Temporary construction facilities; construction services, supplies and consumables; field staff and subsistence expenses; payroll, benefits, insurance; construction equipment and tools				
IFC	$DFL \times 115\%$		2,229,000	
Total field costs (TFC)	$DFC + IFC$		9,982,000	3.6
Home office costs (HOC): Project management, controls and estimating criteria, procurement, construction management, engineering and design, and home-office expenses				
HOC	$DFC \times 30\%$		2,326,000	
Subtotal project cost	$TFC + HOC$		12,308,000	4.4
Other project costs (OTC), including project commissioning costs				
Commissioning	$DFC \times 30\%$		233,000	
Contingency	$(TFC + HOC) \times 15\%$		1,846,000	
Total OTC			2,079,000	
Total Installed project cost (TIPC)			14,387,000	5.1

*Note:* The multiplier is the ratio of DFC, TIPC and other costs to the raw total equipment cost of \$2,805,000.

The cost of each type of equipment was multiplied by a factor to derive the installed DFC for that unit. For instance, the total cost of all vertical vessels (\$540,000) was multiplied by an equipment factor of 3.2 to obtain an installed DFC of \$1,728,000. The total installed cost (TIC) for this project is \$14,387,000.

(Source: Dysert, L., By permission of Chem. Eng., Oct 2001, pp. 70–81.)

percentage of over-sizing would have been added to the design specifications. The purchase cost of the equipment is obtained from purchase orders, published equipment cost data, and vendor quotations. It is essential to accurately determine the equipment costs as the material cost of equipment often represents 20–40% of the total project costs for process plants. Once the equipment cost is determined, the appropriate equipment factors may be generated and applied by applying the necessary adjustments for equipment size, metallurgy and operating conditions. Tables 2-11 shows an example of an equipment factored estimate, and Table 2-12 shows heat exchanger equipment factors. Table 2-13 summarizes the various factors affecting the capital cost of chemical plants.

## 2.7 BARE MODULE COST FOR EQUIPMENT

The bare module equipment cost represents the sum of direct and indirect costs, and are shown in Table 2-13. The conditions specified for the base case are [26]

1. unit fabricated from most common material, usually carbon steel (SS)
2. unit operated at near ambient temperature.

The equation used to calculate the bare module cost for each piece of equipment is

$$C_{BM} = C_p F_{BM} \quad (2-17)$$

**TABLE 2-12 Heat Exchanger Discipline Equipment Factors**

	Factor	Cost (\$)
Equipment cost	1.0	10,000
Installation labor	0.05	500
Concrete	0.11	1,100
Structural steel	0.11	1,100
Piping	1.18	11,800
Electrical parts	0.05	500
Instrumentation	0.24	2,400
Painting	0.01	100
Insulation	0.11	1,100
Total DFC	2.86	28,600

This illustrates equipment factors for a Type 316 stainless steel heat exchanger with a surface area of 2400 ft<sup>2</sup>. The purchase cost of \$10,000 is multiplied by each factor to generate the DFC for that discipline.

(Source: Dysert, L., By permission of Chem. Eng., Oct 2001, pp. 70–81.)

where

$C_{BM}$  = bare module equipment cost – consists of both direct and indirect costs for each unit

**TABLE 2-13 Factors Affecting the Capital Cost of Chemical Plants**

Factors Associated with the Installation of Equipment	Symbol	Comments
1. Direct project expenses		
(a) Equipment FOB cost	$C_p$	Purchased cost of equipment at manufacturer's site (FOB = free on board).
(b) Materials required for installation	$C_M$	Includes all piping, insulation and fire proofing, foundations and structural supports, instrumentation and electrical, and painting associated with the equipment.
(c) Labor to install equipment and material	$C_L$	Includes all labor associated with installing the equipment and materials mentioned in (a) and (b).
2. Indirect project expenses		
(a) Freight, insurance, and taxes	$C_{FIT}$	Includes all transportation costs for shipping equipment and materials to the plant site; all insurance on the items shipped; and any purchase taxes that may be applicable.
(b) Contractor's overhead	$C_O$	Includes all fringe benefits such as vacation, sick leave, retirement benefits, etc.; labor burden such as social security and unemployment insurance, etc.; and salaries and overhead for supervisory personnel.
(c) Contractor engineering expenses	$C_E$	Includes salaries and overhead for the engineering, drafting, and project management personnel on the project.
3. Contingency and fee		
(a) Contingency	$C_{Cont}$	A factor included to cover unforeseen circumstances. These may include loss of time due to storms, strikes, small changes in the design, and unpredicted price increases.
(b) Contractor's fee	$C_{Fee}$	This fee varies depending on the type of plant and a variety of other factors.
4. Auxiliary facilities		
(a) Site development	$C_{site}$	Includes the purchase of land; grading and excavation of the site; installation and hook-up of electrical, water, and sewer systems; and construction of all internal roads, walkways, and parking lots.
(b) Auxiliary buildings	$C_{Aux}$	Includes administration offices, maintenance shop and control rooms, warehouses and service buildings (e.g., cafeteria, dressing rooms, and medical facility).
(c) Offsites and utilities	$C_{Off}$	Includes raw material and final product storage, raw material and final product loading and unloading facilities, all equipment necessary to supply required process utilities (e.g., cooling water, steam generation, fuel distribution systems, etc.), central environmental control facilities (e.g., wastewater treatment, incinerators, flares, etc.), and fire protection systems.

(Source: Turton et al. [25].)

$F_{BM}$  = bare module cost factor – multiplication factor to account for the items in Table 2-14, plus the specific materials of construction and operating pressure

$C_p$  = purchased cost for base conditions – cost of equipment made of the most common material, usually carbon steel, and operating at near ambient pressures.

The entries in Table 2-14 can be explained as follows:

*Column 1:* Lists the factors given in Table 2-13.

*Column 2:* Provides equations used to evaluate each of the costs. These equations introduce multiplication cost factors,  $\alpha_i$ . Each cost item, other than the purchased equipment cost, introduces a separate factor.

*Column 3:* For each factor, the cost is related to the purchased cost,  $C_p$ , by an equation of the form

$$C_{xx} = C_p f(\alpha_i) \quad (2-18)$$

The function,  $f(\alpha_i)$ , for each factor is given in Column 3. Using Table 2-13 and Eqs (2-17) and (2-18), the bare module factor can be expressed by

$$F_{BM} = [1 + \alpha_L + \alpha_{FIT} + \alpha_O \alpha_L + \alpha_E] [1 + \alpha_M] \quad (2-19)$$

## 2.8 SUMMARY OF THE FACTORIAL METHOD

A quick and approximate estimate of the investment which would be required for a project can be achieved from the following procedures:

1. Prepare material and energy balances, draw up preliminary flow-sheets, size major equipment items, and select materials of construction.
2. Estimate the purchase cost of the major equipment items (using the general literature).
3. Calculate the total physical plant cost (PPC) using the factors given in Table 2-9.

$$PPC = PCE (1 + f_1 + \dots + f_9) \quad (2-20)$$

**TABLE 2-14 Equations for Evaluating Direct, Indirect, Contingency, and Fee Costs**

Factor	Basic Equation	Multiplying Factor to be Used with Purchased Cost ( $C_p$ )
1. Direct		
(a) Equipment	$C_p = C_p$	1.0
(b) Materials	$C_M = \alpha_M C_p$	$(1 + \alpha_M) \alpha_L$
(c) Labor	$C_L = \alpha_L (C_p + C_M)$	$(1 + \alpha_M)(1 + \alpha_L)$
Total direct	$C_{DE} = C_p + C_M + C_L$	$(1 + \alpha_M)(1.0 + \alpha_L)$
2. Indirect		
(a) Freight, etc.	$C_{FIT} = \alpha_{FIT} (C_p + C_M)$	$(1.0 + \alpha_M) \alpha_{FIT}$
(b) Overhead	$C_O = \alpha_O C_L$	$(1.0 + \alpha_M) \alpha_L \alpha_O$
(c) Engineering	$C_E = \alpha_E (C_p + C_M)$	$(1.0 + \alpha_M) \alpha_E$
Total indirect	$C_{IDE} = C_{FIT} + C_O + C_E$	$(1.0 + \alpha_M)(\alpha_{FIT} + \alpha_L \alpha_O + \alpha_E)$
Bare module	$C_{BM} = C_{IDE} + C_{DE}$	$(1.0 + \alpha_M)(1.0 + \alpha_L + \alpha_{FIT} + \alpha_L \alpha_O + \alpha_E)$
3. Contingency and fee		
(a) Cont.	$C_{Cont} = \alpha_{Cont} C_{BM}$	$(1.0 + \alpha_M)(1.0 + \alpha_L + \alpha_{FIT} + \alpha_L \alpha_O + \alpha_E) \alpha_{Cont}$
(b) Fee	$C_{Fee} = \alpha_{Fee} C_{BM}$	$(1.0 + \alpha_M)(1.0 + \alpha_L + \alpha_{FIT} + \alpha_L \alpha_O + \alpha_E) \alpha_{Fee}$
Total module	$C_{TM} = C_{BM} + C_{Cont} + C_{Fee}$	$(1.0 + \alpha_M)(1.0 + \alpha_L + \alpha_{FIT} + \alpha_L \alpha_O + \alpha_E)(1.0 + \alpha_{Cont} + \alpha_{Fee})$

(Source: Turton et al. [26].)

4. Calculate the indirect costs from the direct costs using the factors given in Table 2-9.
5. The direct plus indirect costs give the total fixed capital.
6. Estimate the working capital as a percentage of the fixed capital (10–20%).
7. Add the fixed and working capital to obtain the total investment required.

A summary of the costs to consider in the evaluation of the total capital cost of a chemical plant is shown in Table 2-13.

## 2.9 COMPUTER COST ESTIMATING

There are two main methods of using a computer in cost estimates. The first involves calculating the cost given a correlation, a curve fit, or more rigorous model. The second is to use it to analyze a set of historical costs, with the aim of showing important cost trends.

Many simulation software packages with mass/energy balance simulators include costing routines. Some of these packages are developed in-house or alternatively from many vendors of cost estimators. The AACE provides an Excel spreadsheet on cost estimates and website links to vendors who provide software packages on cost estimation. Most spreadsheets have statistical packages and numerical optimizers to analyze records. The power law or exponential equation is expressed by

$$C = kS^n \quad (2-21)$$

where  $n$  represents index of that type of equipment.

This can be extended to a multivariable form as

$$C = kS_1^{n1} S_2^{n2} S_3^{n3} \dots \quad (2-22)$$

where  $S_1, S_2$ , and so on are independent measures of size, for example height and diameter of a vessel. This equation can easily be linearized, and an extension of these forms is to add a further constant,  $k'$ , to represent the fixed cost element.

$$C = k' + kS^n \quad (2-23)$$

and

$$C = k' + kS_1^{n1} S_2^{n2} S_3^{n3} \dots \quad (2-24)$$

Recently, Petley and Edwards [27] have shown the use of fuzzy matching to be useful in costing a complete chemical plant. Gerrard and Brass [28] have used this approach together with neural networks and rational polynomials to provide equipment cost predictions.

## 2.10 PROJECT EVALUATION

### INTRODUCTION

Project evaluation enables the technical and economic feasibility of a chemical process to be assessed using preliminary process design and economic evaluations. Once a process flowsheet is available, these evaluations can be classified into a number of steps: material balance calculations, equipment sizing, equipment cost determination, utilities requirements, investment cost estimation, sales volume forecasting, manufacturing cost estimation, and finally profitability and sensitivity analysis.

The results of an economic evaluation are reviewed together with other relevant aspects, for example competition and likely product life, in arriving at project investment decisions. The decisions are essential in order to plan and allocate the available resources for long-term use. The objectives of an economic appraisal are the following:

- To ensure that the expected future benefits justify the expenditure of resources.
- To choose the best project from among alternatives to achieve future benefits.
- To utilize all available resources.

These aim to ensure so far as is practicable that the right project is undertaken and is likely to attain the desired profitability.

In CPIs, an investment project may arise from any of a range of activities. It may be a minor modification to an existing plant, a major plant expansion or revamping, a completely new plant (on an existing or greenfield site), or the development of an entirely

**TABLE 2-15 Use of Chemical Engineering Economics**

<b>1. Production or plant technical services</b>
(i) Plant equipment continuously need repair, scheduled or unscheduled replacement, or modernization. The engineer responsible should know approximately what the comparative performance, costs, and payout periods are, even if there is a plant engineering group specializing in that type of analysis, or later a firm price quotation will be obtained.
(ii) Plant changes, e.g., initiated by the increasing costs of energy and to meet environmental requirements, necessitate that many energy-saving, and pollution and hazardous waste control possibilities must be considered. As a basis for recommendations, the engineer responsible should personally conduct design and cost estimates, payback, and economic calculations on the alternatives before making even preliminary recommendations.
(iii) Competitors' processing methods, as well as changes suggested by R&D, sales, or management must be continuously examined. Managers or other groups may be responsible, but the engineer can help by making preliminary cost estimates and economic analyses of the changes as a guide to his or her own thinking and assessment of the group's position.
(iv) All engineers should have some feel for their company's business, products, and economics. This requires a basic understanding of company annual reports and of general industry economic news reinforced by regular reading.
<b>2. Research and Development</b>
(i) During the analytical phase of creative thinking many novel ideas require a rapid cost estimation and economic analysis to provide a clearer idea of their merit. Managers or others may be assigned to do this work, but the chances of novel proposals being accepted increase if some economic screening is performed by the originator.
(ii) Many obstacles, or alternative directions that may be taken to attempt to solve the problems, arise during R&D. Brief cost estimates and economic analyses often help in determining which are the optimum.
(iii) Following success during an early or intermediate stage of an R&D program, new funding requests are usually required to continue the study. These requests benefit from support by preliminary economic analyses. In the final stages of a project, the engineer may be part of a team assigned to provide a more definitive preliminary economic projection and analysis.
(iv) Credibility with production, sales, or management personnel is enhanced by demonstrating a reasonable knowledge of the costs and economics of the projects under study, and general industry economics.
<b>3. Sales</b>
(i) A general knowledge of company costs, profits, and competition is essential for more effective salesmanship.
(ii) Salesmen may recommend new products, improvements, or pricing ideas to their management. A cost and economic estimate for these ideas should be helpful in the proposal report.
(iii) Salesmen may perform market surveys. A general economic knowledge of the industries and companies surveyed may be essential, and is always useful.
(iv) Salesmen may progress into management, where economic knowledge is essential.
<b>4. Engineering</b>
(i) As a result of the specialization within most engineering companies and engineering departments, cost estimating and economics may not be required directly in many engineering company or departmental jobs. Other jobs will, however, deal exclusively with cost estimating and economic analysis, and all benefit from a good, fluent knowledge of the basic economic procedures. Engineering departments or companies usually have very well developed in-house methods and data that must be used, but the basics are still applicable.
<b>5. General</b>
(i) All chemical engineers are assumed to know the rudiments of cost estimating, economic evaluation, and the economics of their industry. A high percentage will find this knowledge useful or necessary throughout their careers.
(ii) Progression into management in associated areas will necessitate further understanding of microeconomics.
(iii) All work situations are competitive. One means of enhancing advancement potential is to demonstrate to superiors knowledge and an interest in management, business, and economics. Associated with this is the demonstration of ability, an interest in accepting responsibility, and ability to communicate. Promotion may go to those perceived as able to "manage", as those with MBA (Master of Business Administration) degrees, or equivalent capabilities. A confidence in the ability to acquire managerial skills as required, and a knowledge of economics, should render our chemical engineer an equal or preferred candidate for advancement.

(Source: After Garrett [38].)

new process or product. Economic assessment for a major or a new plant may be carried out with increasing degrees of accuracy at different stages as it progresses. This may be based on research and development (R&D), through various stages of the project (e.g., pilot plants, material evaluation, preliminary plant design), and leads to the decision whether or not to proceed with investment in a full-scale plant for the process. Table 2-15 lists examples of the use of chemical engineering economics. The approach used for economic evaluation depends on the quality of the information, which depends on the stage in the project at which it is being undertaken.

Investment decisions are often based upon several criteria such as annual return on investment (ROI), payback period (PBP), net present value (NPV), the average rate of return (ARR), present value ratio (PVR), and the internal rate of return (IRR). Discounted cash flow rate on return (DCFRR) is another popular means of evaluating the economic viability of a proposed project. Hortwitz [29] recommended the DCFRR as the best means to determine the ROI since it accounts for the time value of money (the cost of money), which states that the value of a sum of money is time dependent. For example, given the choice of receiving say \$100 now or \$100 in a year's time, it is preferred to receive the

money now since it could be invested (at no risk) to obtain interest until it is required. Therefore, the \$100 received now is worth more than the \$100 received in a year's time by the amount of interest it could earn in the intervening year. Conversely, \$100 received in a year's time is worth less than \$100 received now, and \$100 received in two year's time is less than this, and so on. The time value of money is the result of the availability of investment possibilities, which generates returns on any investment made. This time value of money does not account for inflation, which is a separate factor. The internal rate of return as an investment criterion introduces the possibility that given cash flows may result in more than one internal rate of return. Cannaday et al. [30] developed a method for determining the relevance of an internal rate of return. They inferred that an internal rate of return is relevant if its derivative with respect to each of the cash flows is positive.

Powell [31] reviewed the basics of various discounted cash flow techniques for project evaluation. Discounting is a method that accounts for the time value of money to provide either for the capital, or to convert the cash flows to a common point in time so that they are summed. Ward [32] proposed a new concept known as the net return rate (NRR) that provides a better indication of a project's profitability. Techniques and criteria for economic evaluation of projects are widely available in the literature and texts [33–38]. A summary of the conventional decision criteria is given here.

## CASH FLOWS

**Return on Investment.** In engineering economic evaluation, rate of return on investment is the percentage ratio of average yearly profit (net cash flow) over the productive life of the project, divided by the total initial investment. This is calculated after income tax has been deducted from the gross or pre-tax income. The remaining net income may be used either for paying dividends for re-investment or spent for other means. Return on investment is defined by

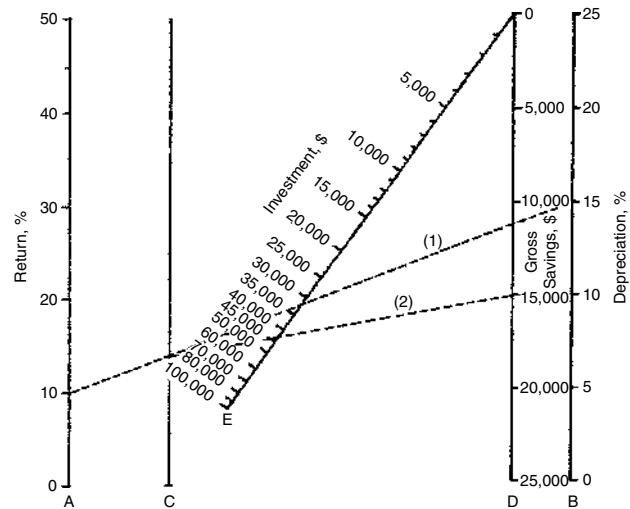
$$\text{ROI} = \frac{\text{Annual return}}{\text{Investment}} \times 100 \quad (2-25)$$

The annual return may be the gross income, net pre-tax income, net after-tax income, cash flow, or profit. These may be calculated for one particular year or as an average over the project life. Investment may be the original total investment, depreciated book-value investment, life time average investment fixed capital investment, or equity investment. The investment includes working capital and sometimes capitalized expenses, for example interest on capital during construction.

The proper evaluation of costs as they affect the selection of processes and equipment is not included in this book. Every process engineer should, however, be cognizant of the relationships. There are several methods to evaluate return on invested money; the nomograph of Figure 2-2 represents one. It is a useful guide [39] to estimate the order of magnitude of a return on expenditure to gain savings in labor and/or material costs. The nomograph is used to determine the investment justified by a gross annual savings, assuming a percent return, a percent annual depreciation charge, and a 50% Federal tax on net savings.

Return

$$= \frac{(\text{gross savings} - \text{depreciation} \times \text{investment})(1 - \text{Federal tax})}{\text{investment}} \quad (2-26)$$



**Figure 2-2** Annual savings, return, and depreciation fix justifiable investment. (By permission from G.A. Larson, Power, Sept. 1995.)

### EXAMPLE 2-3

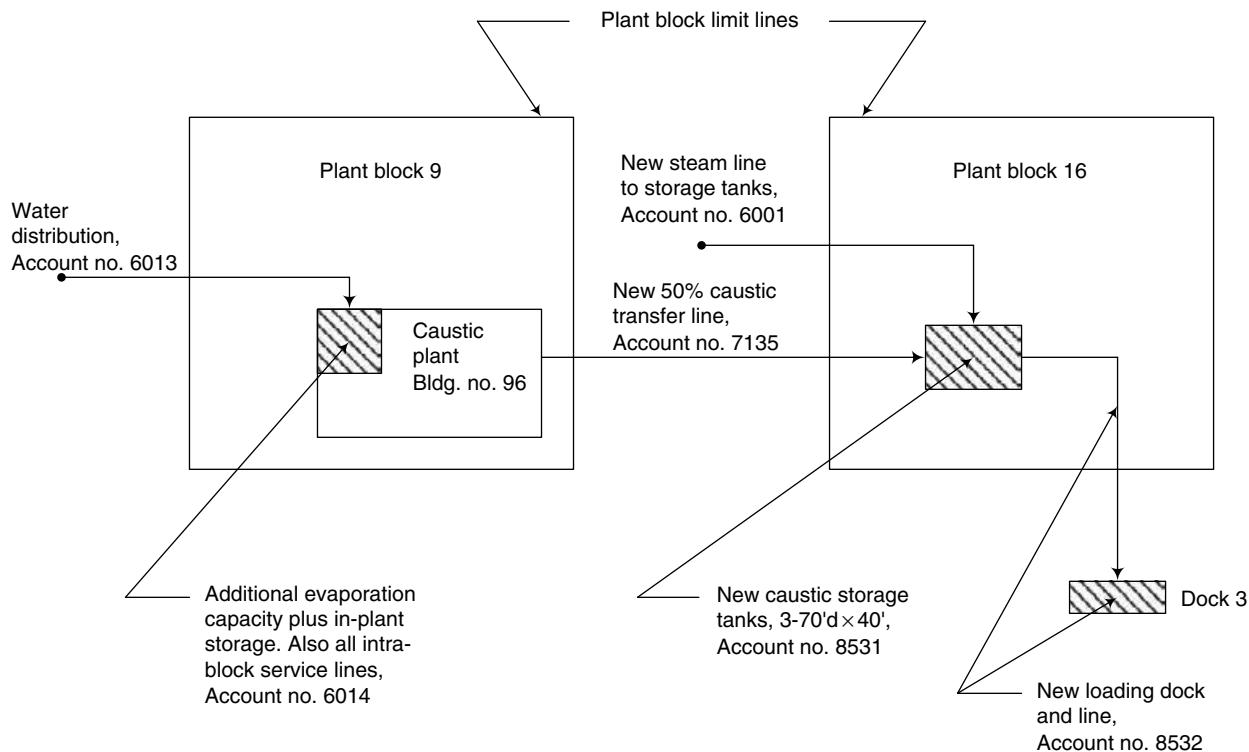
#### Justifiable Investment for Annual Savings [39]

Find the justifiable investment for a gross annual savings of \$15,000 when a return of 10% and a depreciation rate of 15% are specified.

1. From Figure 2-2, connect scales A and B.
2. From the intersection with the C scale, connect a line to the D scale.
3. At the intersection of line (2) with the inclined investment scale, E, we can read that a \$43,000 investment is justified to save a \$15,000 gross per year.

**Accounting Coordination.** All new plants and changes to existing facilities and plants must be coordinated with a cost accounting system. Often costs associated with the building, services, utilities, and site development must be separated from each other. Each company has reason and need for various arrangements in order to present proper information for tax purposes and depreciation. The project engineer is usually responsible for this phase of coordination through the engineering groups; but the process engineer may need to present proper breakdown details; these then serve to coordinate the cost breakdowns. Figure 2-3 is an example of such an accounting diagram.

**Payback Period.** Payback period (PBP) is widely used when long-term cash flows, that is, over a period of several years, are difficult to forecast, since no information is required beyond the breakeven point. It may be used for preliminary evaluation or as a project-screening device for high-risk projects in times of financial uncertainty. Payback period is usually measured as the time from the start of production to recovery of the capital investment. The payback period is the time taken for the cumulative net cash flow from the start-up of the plant to equal the depreciable fixed



**Figure 2-3** Account diagram for accumulation of project costs. Cost estimates must be made to conform to the same scope.

capital investment ( $C_{FC} - S$ ). It is the value of  $t$  that satisfies the equation

$$\sum_{t=0}^{t=(PBP)} C_{CF} = (C_{FC} - S) \quad (2-27)$$

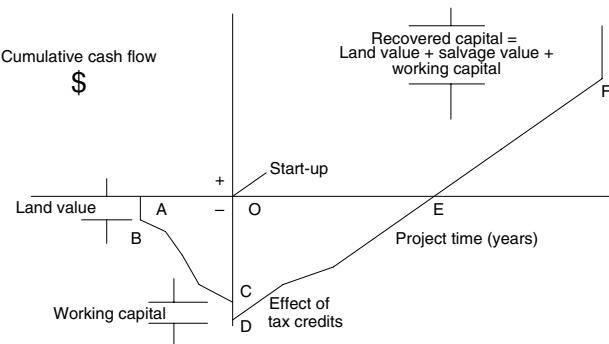
where

$C_{CF}$  = net annual cash flow

$C_{FC}$  = fixed capital cost

$s$  = salvage value.

Figure 2-4 shows the cumulative cash flow diagram for a project. The PBP is the time that elapses from the start of the project A, to the breakeven point E, where the rising part of the curve passes the zero cash position line. The PBP thus measures the time required for the cumulative project investment and other expenditure to be balanced by the cumulative income.



**Figure 2-4** Cumulative cash flow diagram.

#### EXAMPLE 2-4

Consider the following cash flow:

Year	0	1	2	3	4
Cash flow	-\$8000	\$3000	\$4000	\$5000	\$5000

Here, the cumulative cash flow at the end of the second year is  $\$3000 + \$4000 = \$7000$ , which is less than the initial investment, but the cumulative cash flow at the end of the third year is  $\$3000 + \$4000 + \$5000 = \$12,000$ , which is more than the initial investment. The payback period is thus between two and three years, assuming that the cash flow of year 3 is received uniformly throughout the year. The payback period is calculated as follows:

1	Initial investment	\$8000
2	Cash flow recovered to end of second year	\$7000
3	Amount still to be recovered (line 1 – line 2)	\$1000
4	Amount recovered in third year	\$5000
5	Amount in step (3) divided by amount in step (4) ( $\$1000/\$5000$ )	0.2 year
6	Payback period (2 years + number of years in step 5)	2.2 years

In this example, 0.2 is the fraction of year number 3 that it will take to recover \$1000. Adding this fraction to the two years during which \$7000 is recovered yields a payback period of  $2 + 0.2 = 2.2$  years.

## 84 COST ESTIMATION AND ECONOMIC EVALUATION

The payback period method of evaluating investments has a number of flaws and is inferior to other methods. A major disadvantage is that after the payback period, all the cash flows are completely ignored. It also ignores the timing of the cash flows within the payback period.

**Present Worth (or Present Value).** In an economic evaluation of a project, it is often necessary to evaluate the present value of funds that will be received at some definite time in the future. The present value (PV) of a future amount can be considered as the present principal at a given rate and compounded to give the actual amount that will be received at a future date. The relationship between the indicated future amount and the present value is determined by a discount factor. Discounting evaluates each year's flow on an equal basis. It does this by means of the discount, or present value factor, which is the reciprocal of the compound interest factor  $(1+i)^n$ , where

$i$  = interest rate

$n$  = the year in which the interest is compounded.

$$\text{The discount factor} = \frac{1}{(1+i)^n} \quad (2-28)$$

If  $C_n$  represents the amount available after  $n$  interest periods,  $p$  is the initial principal, and the discrete compound interest rate is  $i$ , then PV can be expressed as

$$PV = p = \frac{C_n}{(1+i)^n} \quad (2-29)$$

**Net Present Value.** The net present value (NPV) of a project is the cumulative sum of the discounted cash flows including the investment. The NPV corresponds to the total discounted net return, above and beyond the cost of capital and the recovery of the investment. The NPV represents a discounted return or profit but is not a measure of the profitability. Each cash flow is evaluated by computing its present value. This is done by taking a cash flow of year  $n$  and multiplying it by the discount factor for the  $n$ th year.

$$\text{Present value of } p_n = C_n \left[ \frac{1}{(1+i)^n} \right] \quad (2-30)$$

The present value of  $p$  at year 0 of a cash flow  $C_t$  in year  $t$  at an annual discount rate of  $i$  is

$$p = \frac{C_t}{(1+i)^t} \quad (2-31)$$

For a complete project, the earlier cash flows are usually negative and the later ones positive. The NPV is the sum of the individual present values of the yearly cash flows. This is expressed as

$$NPV = C_0 + \frac{C_1}{(1+i)} + \frac{C_2}{(1+i)^2} + \cdots + \frac{C_n}{(1+i)^n} \quad (2-32)$$

Equation (2-32) can be expressed as

$$NPV = \sum p = \sum_{i=0}^{i=n} \frac{C_n}{(1+i)^n} \quad (2-33)$$

where

NPV = net present value

$C_0$  = initial investment

$C_n$  = cash flow

$n$  = year  $n$

$i$  = interest rate of return (ROI/100).

The life of the project  $n$  years must be specified together with the estimated cash flows in each year up to  $n$ .

If a project includes a series of identical yearly cash flows, such as the net inflow for several years, their present values can be obtained in one calculation instead of obtaining which individually. For yearly cash flows  $C$  from year  $m$  to year  $n$ , the sum of their present values is

$$p = \frac{C}{i(1+i)^{m-1}} \left[ 1 - \frac{1}{(1+i)^{n-m+1}} \right] \quad (2-34)$$

Figure 2-5 shows the cumulative NPV stages in a project, and Table 2-16 shows the discount factors for computing the NPV.

Assuming that the investment is made in year 0 ( $C_0 = I$ ), and the cash flows over the project life are constant, then Eq. (2-32) simplifies to give

$$NPV = \left[ C \sum_{n=1}^n \left( \frac{1}{1+i} \right)^n \right] - I \quad (2-35)$$

The term  $\frac{1}{(1+i)}$  is a geometric progression whose sum can be expressed as the single term

$$\sum_{n=1}^n \frac{1}{(1+i)^n} = \frac{(1+i)^n - 1}{i(1+i)^n} \quad (2-36)$$

where Eq. (2-36) becomes the present value of an annuity. Equation (2-32) can thus be expressed as

$$NPV = C \left[ \frac{(1+i)^n - 1}{i(1+i)^n} \right] - I \quad (2-37)$$

The internal rate of return (IRR) is that value of  $i$  which makes NPV equal to 0. Therefore, if NPV is set to 0, the IRR that makes the future cash flows equal to the investment (the "break-even" point) can be estimated. Equation (2-37) then becomes

$$\frac{I}{C} = \frac{(1+i)^n - 1}{i(1+i)^n} \quad (2-38)$$

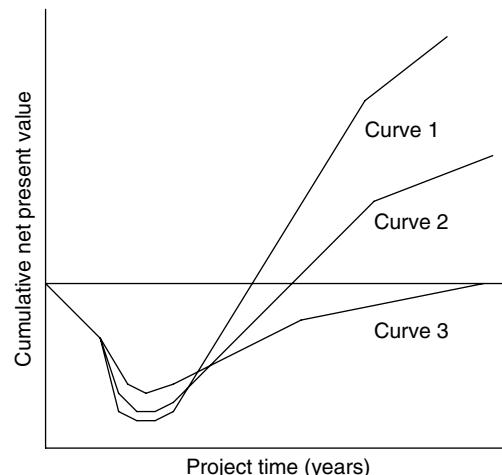


Figure 2-5 Cumulative net present value (NPV) of a project.

**TABLE 2-16** Discount Factors (This table shows the present value of unity discounted for different numbers of years and at different rates of discount.)

*(continued)*

**TABLE 2-16—(continued)**

Rate of Discount \ Year	21%	22%	23%	24%	25%	26%	27%	28%	29%	30%
17	0.0391	0.0340	0.0296	0.0258	0.0225	0.0197	0.0172	0.0150	0.0132	0.0116
18	0.0323	0.0279	0.0241	0.0208	0.0180	0.0156	0.0135	0.01180	0.0102	0.0089
19	0.0267	0.0229	0.0196	0.0168	0.0144	0.0124	0.0107	0.0092	0.0079	0.0068
20	0.0221	0.0187	0.0159	0.0135	0.0115	0.0098	0.0084	0.0072	0.0061	0.0053
	31%	32%	33%	34%	35%	36%	37%	38%	39%	40%
0	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
1	0.7634	0.7576	0.7519	0.7463	0.7407	0.7353	0.7300	0.7246	0.7194	0.7143
2	0.5827	0.5739	0.5653	0.5569	0.5487	0.5407	0.5328	0.5251	0.5176	0.5102
3	0.4448	0.4348	0.4251	0.4156	0.4064	0.3975	0.3889	0.3805	0.3724	0.3644
4	0.3396	0.3294	0.3196	0.3102	0.3011	0.2923	0.2839	0.2757	0.2679	0.2603
5	0.2592	0.2495	0.2403	0.2315	0.2230	0.2149	0.2072	0.1998	0.1927	0.1859
6	0.1979	0.1890	0.1807	0.1727	0.1652	0.1580	0.1512	0.1448	0.1386	0.1328
7	0.1510	0.1432	0.1358	0.1289	0.1224	0.1162	0.1104	0.1049	0.0997	0.0949
8	0.1153	0.1085	0.1021	0.0962	0.0906	0.0864	0.0806	0.0760	0.0718	0.0678
9	0.0880	0.0822	0.0768	0.0718	0.0671	0.0628	0.0588	0.0551	0.0516	0.0484
10	0.0672	0.0623	0.0577	0.0536	0.0497	0.0462	0.0429	0.0399	0.0371	0.0346
11	0.0513	0.0472	0.0434	0.0400	0.0368	0.0340	0.0313	0.0289	0.0267	0.0247
12	0.0392	0.0357	0.0326	0.0298	0.0273	0.0250	0.0229	0.0210	0.0192	0.0176
13	0.0299	0.0271	0.0245	0.0223	0.0200	0.0184	0.0167	0.0152	0.0138	0.0126
14	0.0228	0.2050	0.0185	0.0166	0.0150	0.0135	0.0122	0.0110	0.0099	0.0090
15	0.0174	0.0155	0.0139	0.0124	0.0111	0.0099	0.0089	0.0080	0.0072	0.0064
16	0.0133	0.0118	0.0104	0.0093	0.0082	0.0073	0.0065	0.0058	0.0051	0.0046
17	0.0101	0.0089	0.0078	0.0069	0.0061	0.0054	0.0047	0.0042	0.0037	0.0033
18	0.0077	0.0068	0.0059	0.0052	0.0045	0.0039	0.0035	0.0030	0.0027	0.0023
19	0.0059	0.0051	0.0044	0.0038	0.0033	0.0029	0.0025	0.0022	0.0019	0.0017
20	0.0045	0.0039	0.0033	0.0029	0.0025	0.0021	0.0018	0.0016	0.0014	0.0012

where

$I$  = investment

$C$  = cash flow for each year

$i$  = rate of return (IRR/100)

$n$  = years of project life.

Equation (2-38) is defined as the annuity with  $I$  as the present value and  $C$  as the equal payments over  $n$  years at interest  $i$ . The  $I/C$  term is referred to as the payback period. If  $I$  and  $C$  are evaluated and the life of the project is known,  $i$  can be computed by trial and error. The value of  $i$  can be determined from Figure 2.6.

The NPV measures the direct incentive to invest in a given proposal as a bonus or premium over the amount an investor could otherwise earn by investing the same money in a safe alternative, which would yield a return calculated at the rate  $i$ . The resulting NPV from a project's cash flows is a measure of the cash profit that the project will produce after recovering the initial investment and meeting all costs, including the cost of capital. The more positive the NPV is, the more attractive the proposition. If NPV is 0, the viability of the project is marginal; if it is negative, the proposal is unattractive.

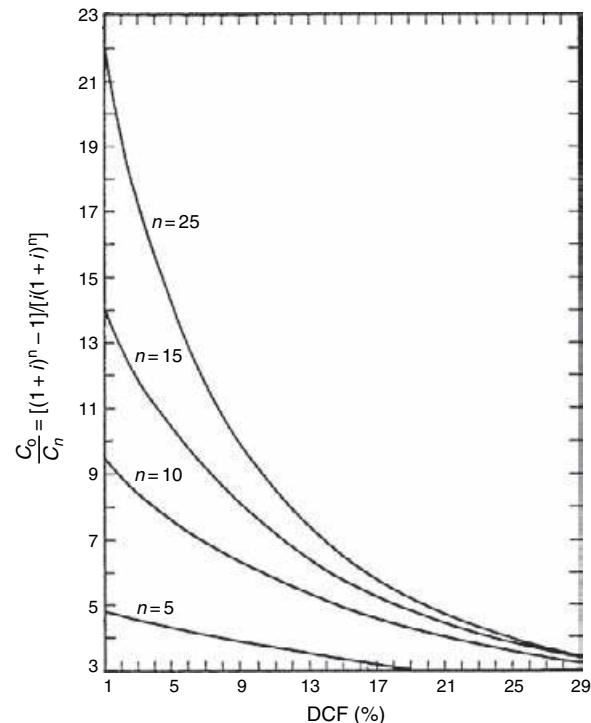
**The Profitability Index.** The profitability index (PI) is defined as the present value of an investment's cash flow divided by the initial outlay (investment). If the present value of the cash flow is PV, then the PI is

$$\text{PI} = \frac{\text{PV}}{I} \quad (2-39)$$

where

$I$  = the initial outlay (investment)

PV = net present value + initial outlay (i.e.,  $\text{NPV} + I$ ).



**Figure 2-6** Graphical solution for simplified DCF calculation (constant cash flow, compounded annually, no salvage working capital return). (Source: Horwitz [29].)

**EXAMPLE 2-5**

Suppose the cost of capital is 15%, and the cash flows on a project are as follows:

Year					NPV (at 10%)	IRR	
0	1	2	3	4			
Cash flow	-\$600	\$250	\$250	\$250	\$250	\$192.47	24.1%

The project's NPV is \$192.47, and the PV is \$792.47 = (192.47 + 600). The profitability index is

$$PI = \frac{PV}{I} = \frac{\$792.47}{\$600} = 1.32$$

This can be interpreted as a 32% return on a \$600 investment. Alternatively, it can also be interpreted that the investment will return the initial amount plus the NPV that is equal to 32% of the initial investment.

The PI method has an advantage over the IRR method in that the latter uses the IRR itself to adjust for the time value of the money whereas the former uses the cost of capital. However, PI is still inferior to the NPV as a criterion for evaluating investment, because like the IRR, it does not directly take into account the difference in investment scale. Table 2-17 summarizes the discussion of the NPV, IRR, PI techniques for project evaluation.

**EXAMPLE 2-6**

A manufacturing company is considering bringing out a new product. The company plans to invest \$12 million at the beginning of year 1 (i.e., at the end of year 0) and another \$7 million one year later. Marketing studies suggest that this investment will generate an irregular series of revenues from year 3 through 12. The cash flow components are summarized below:

End of Year	Revenue (\$)	End of Year	Revenue (\$)
0	-12,000,000	7	5,000,000
1	-7,000,000	8	6,000,000
2	0	9	5,500,000
3	1,000,000	10	3,500,000
4	2,000,000	11	2,500,000
5	3,000,000	12	2,000,000
6	4,000,000	13	1,000,000

Calculate the following:

1. The present value of the proposed investment, assuming the company normally receives a return of 8% per year, compounded annually.
2. The internal rate of return (i.e., the interest rate at which the present value of the cash flow is zero).

**Discounted Cash Flow Rate of Return.** The discounted cash flow rate of return (DCFRR) is known by other terms, for example, the profitability index, the true rate of return, the investor's rate of return, and the internal rate of return. It is defined as the discount rate  $i$  which makes the NPV of a project equal to zero. This can be expressed mathematically by

$$NPV = \sum p = \sum_{t=0}^{t=n} \frac{C_t}{(1+i)^t} = 0 \quad (2-40)$$

**Relationship between Payback Period and Discounted Cash Flow Rate of Return.** For the case of a single lump-sum capital expenditure  $C_{FC}$  which generates a constant annual cash flow  $A_{CF}$  in each subsequent year, the payback period (PBP) is expressed by Perry [40]

$$PBP = \frac{C_{FC}}{A_{CF}} \quad (2-41)$$

If the scrap value of the capital outlay may be taken as zero, then in the limiting case when  $n$  approaches infinity (i.e.,  $n \rightarrow \infty$ ), the maximum DCFRR is defined as

$$DCFRR_{\max} = \frac{1}{PBP} \quad (2-42)$$

This means, for example, that if the PBP is 5 years, the maximum possible discounted cash flow rate of return (DCFRR) which can be reached is 20%. The corresponding DCFRR for a PBP of 10 years is 10%.

The main advantage of DCFRR over NPV is that it is independent of the zero or base year that is chosen. In contrast, the value of NPV varies according to the zero year chosen. In

**Solution**

The cash flow was entered into an Excel worksheet (Example 2-6.xls) to determine the present value of this proposed investment, assuming the company normally receives a return of 8% per year, compounded annually. The initial investments are shown as negative cash flow components at the end of years 0 and 1 respectively. Figure 2-7a shows the Excel worksheet containing the cash flow components, where columns A and B show the project life and the cash flow respectively. A bar graph of the cash flow components (in column D) is also shown. Figure 2-7b shows a plot of the present value (PV) against project life, and Figure 2-7c illustrates the net present value at varying interest rates. The plot indicates that the NPV decreases with increasing interest rate, eventually crossing the abscissa and becoming negative. The internal rate of return is the crossover point, that is, the value of the interest rate at which the net present value is zero. Figure 2-7c shows an internal rate of return of 9%.

Cell B32 is the NPV, which contains the sum of the present values of the cash flow components. The NPV is  $\$1.11 \times 10^6$ , indicating that the present value of the future cash inflows exceeds the present value of the initial investments. Additionally, cell E32 in the spreadsheet is the present value ratio (PVR) and is calculated to equal 1.06, which is greater than 1.0. Hence this new product represents an attractive investment opportunity. Further calculations show that the payback period is 6.4 years and the return on investment is 15.6%. The Excel spreadsheet (Example 2-6.xls) can be downloaded from the companion website.

**TABLE 2-17 Methods of Evaluating Investment Proposals**

	<b>NPV</b>	<b>IRR</b>	<b>PI</b>
Accept-reject decision for independent projects	Accept if $NPV > 0$ Reject if $NPV < 0$	Accept if $IRR > k$ Reject if $IRR < k$	Accept if $PI > 1$ Reject if $PI < 1$
Decision when choosing from mutually exclusive proposals	Choose proposal with high $NPV$ provided it is greater than zero	Choose proposal with highest $IRR$ provided it is higher than the cost of capital, $k$	Choose proposal with highest $PI$ provided it is greater than 1
Advantages	<ul style="list-style-type: none"> <li>1. Is a direct measure of a project's contribution to stockholders' wealth</li> <li>2. Uses the opportunity cost of capital</li> <li>3. Assumes that interim cash flows are reinvested at a rate of return equal to the cost of capital</li> <li>4. Accept-reject decisions and ranking of investments are consistent with maximization of stockholders' wealth</li> </ul>	<ul style="list-style-type: none"> <li>1. Accept-reject decisions for independent projects are consistent with maximization of stockholders' wealth</li> <li>2. Measured in percentage, which is appealing to many people</li> </ul>	<ul style="list-style-type: none"> <li>1. Accept-reject decisions are the same as that of NPV and IRR; therefore they are consistent with maximization of stockholders' wealth</li> <li>2. The assumption about reinvestment rate is as made by the NPV</li> <li>3. Expresses the NPV in relation to the initial investment study</li> </ul>
Disadvantages	<ul style="list-style-type: none"> <li>1. The only real disadvantage occurs when projects with unequal lifetimes are compared. This disadvantage is not unique to NPV</li> <li>2. Although not really a disadvantage, the NPV is not expressed in percent and does not reveal what resources are needed to generate a given NPV</li> </ul>	<ul style="list-style-type: none"> <li>1. Not a direct measure of the effect of a project on stockholders' wealth</li> <li>2. Makes the assumption that cash flows can be reinvested to yield the IRR</li> <li>3. Ranking is inconsistent with maximization of stockholders' wealth</li> <li>4. Some projects may have more than one IRR and others may have no IRR at all</li> </ul>	<ul style="list-style-type: none"> <li>1. Not a direct measure of the effect of a project on stockholders' wealth</li> <li>2. Ranking is inconsistent with maximization of stockholders' wealth</li> </ul>

(Source: Ben-Horim, M., *Essentials of Corporate Finance*, Allyn & Bacon, Inc., Boston, 1987.)

#### EXAMPLE 2-7

The following cash flows describe two competing investment opportunities:

<b>End of Year</b>	<b>Revenue Proposal A (\$)</b>	<b>Revenue Proposal B (\$)</b>
0	-500,000	-500,000
1	150,000	50,000
2	150,000	70,000
3	150,000	100,000
4	150,000	200,000
5	150,000	400,000

Determine which investment is more attractive, based upon

1. present value using an interest rate of 5%, compounded annually;
2. the internal rate of return, NPV, and PVR.

#### Solution

The Excel worksheet (Example 2-7.xls) calculates the net present value and internal rate of return for investment plans A and B. Figure 2-8a shows the cash flows against the project life of investment plans A and B, and Figure 2-8b shows the present value against the project life of the two investment plans A and B respectively. The net present value at 5% interest rate shows that proposed plan B (\$175,446) is higher than that of proposed plan A (\$149,421). However, the calculated internal rate of return (IRR) shows that proposed plan A (15%) is higher than that of proposed plan B (13%). Further calculation on the PVR shows that plan B ( $PVR = 1.35$ ) is higher than plan A ( $PVR = 1.3$ ). This further indicates that proposed investment plan B is more attractive. Figure 2-8c shows plots of NPV as a function of interest rate for plans A and B respectively. These profitability criteria as illustrated in Table 2-18 indicate that proposed plan B is the more attractive investment. The Excel spreadsheet (Example 2-7.xls) can be downloaded from the companion website.

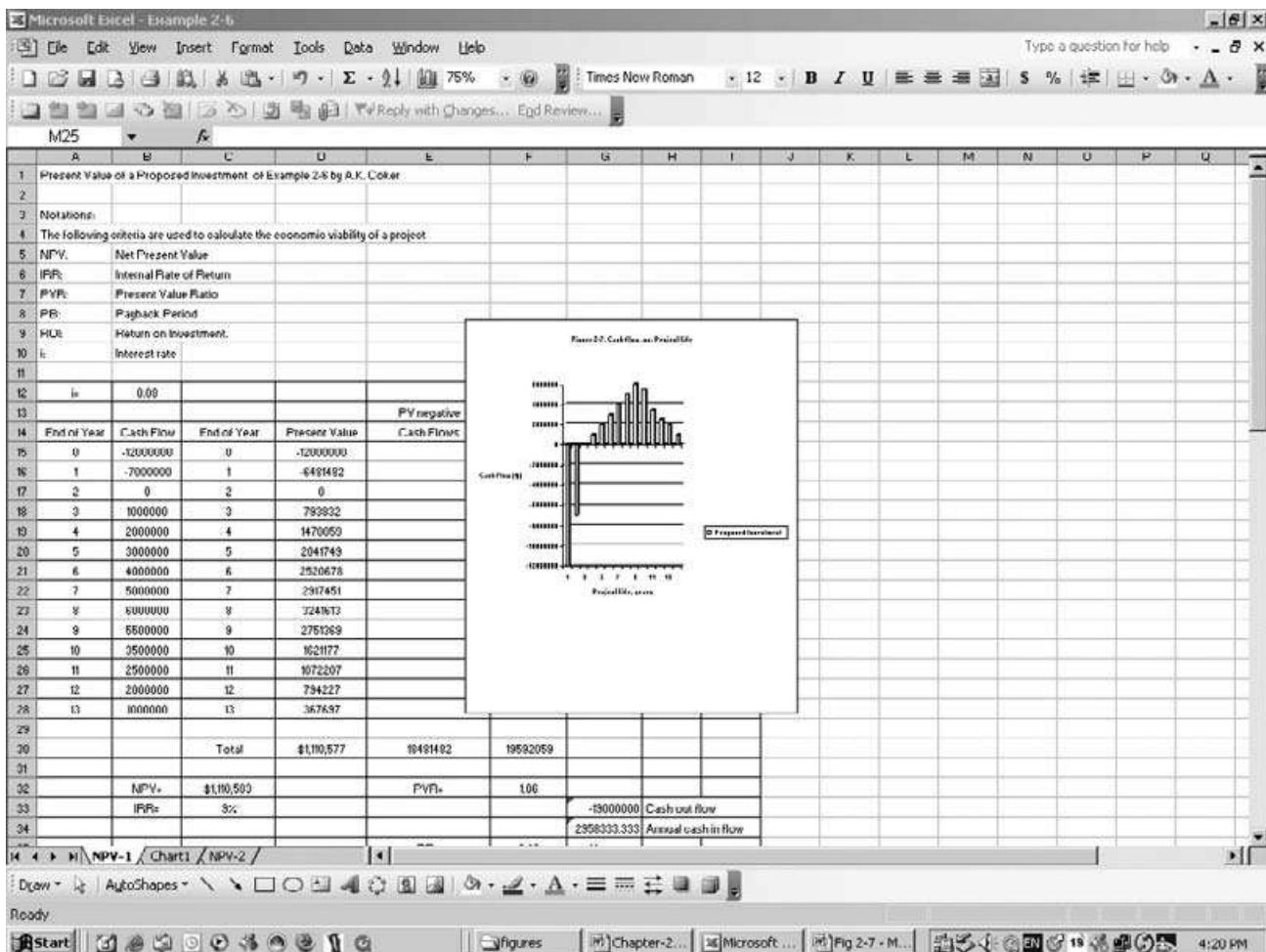


Figure 2-7a Excel worksheet for Example 2-6, with an embedded bar graph of the cash flow components.

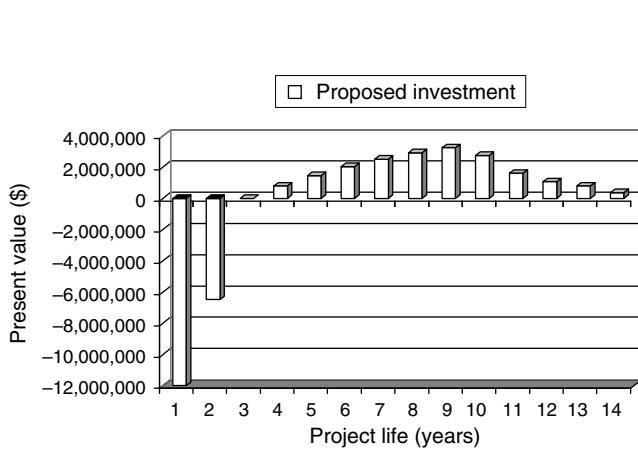


Figure 2-7b Present value vs project life.

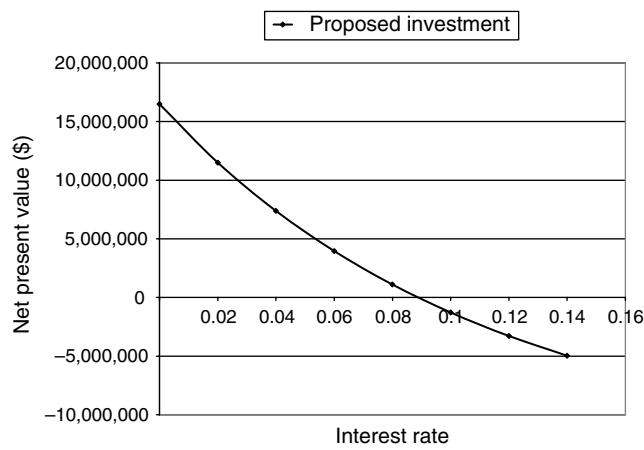


Figure 2-7c Net present value vs interest rate.

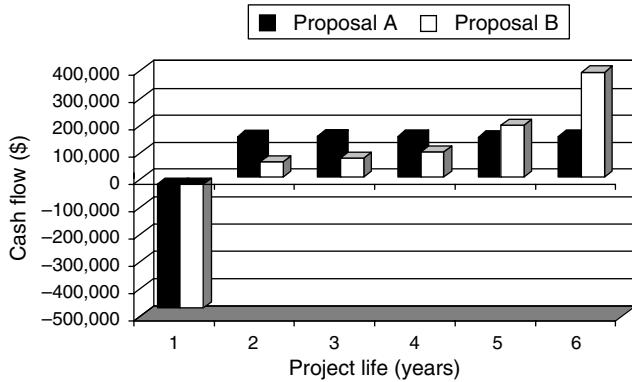


Figure 2-8a Cash flow vs project life.

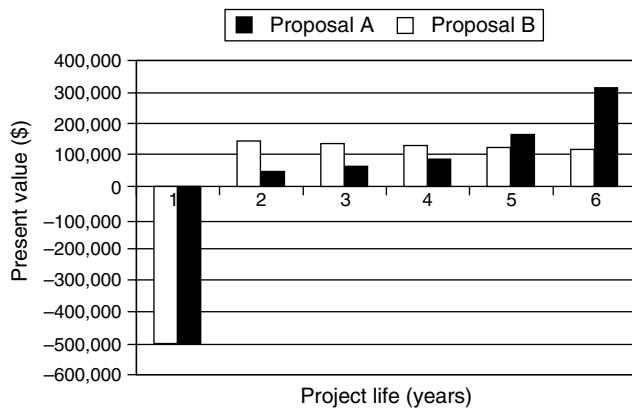


Figure 2-8b Present value vs project life.

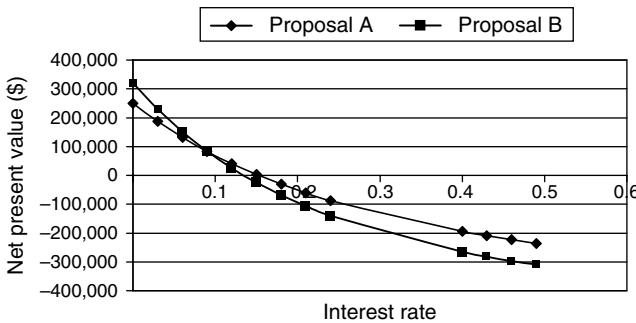


Figure 2-8c Net present value vs interest rate.

TABLE 2-18 Profitability Criteria of Proposals A and B

Profitability Criteria	Proposal A	Proposal B
Net present value, \$	1,149,222	1,75,446
Present value ratio	1.3	1.35
Internal rate of return, %	15	13
Return on investment, %	30	32.8
Payback period, years	3.3	3.05

calculating the NPV, the cost of capital has to be explicitly included as  $i$  in the discounting calculations. In computing the DCFRR, the cost of capital is not included. Instead, the value calculated for  $i$  is compared with the cost of capital to see whether the project is profitable. The DCFRR for a project is the rate of ROI. It measures the efficiency of the capital and determines the earning power of the project investment. Therefore, a DCFRR of 20% implies that 20% per year will be earned on the investment, in addition to which the project generates sufficient money to repay the original investment plus any interest payable on borrowed capital and all taxes and expenses.

**Net Return Rate.** The net return rate (NRR) is analogous to the rate of return and is the net average discounted “return” on the investment over and above the cost of capital. This is defined by

$$\text{NRR} = \frac{\text{NPV}}{\left( \begin{array}{l} \text{discounted} \\ \text{investment} \end{array} \right) \left( \begin{array}{l} \text{project} \\ \text{life} \end{array} \right)} \times 100 \quad (2-43)$$

where the investment is discounted to the same point as the NPV. Holland et al. [35] introduced the NPV/investment ratio as a normalized measure of the total discounted return over the life of the investment. Ward [32] showed that the NPV can be divided by the number of cash flow increments (venture lifetime) such that the NRR corresponds to the average discounted net return on investment. The cost of capital is already accounted for by the discount rate in the NPV computation, and therefore the NRR is the true return rate.

## INCREMENTAL CRITERIA

**Incremental Net Return Rate.** In an incremental-investment analysis, the various options are first ranked by size and the venture with the smallest investment is selected as the base case. Then the incremental net present value (INPV) and incremental net return rate (INRR) can be calculated. The INPV is the net present value of the incremental cash flows and is defined as

$$\text{INRR} = \frac{\text{INPV}}{\left( \begin{array}{l} \text{incremental} \\ \text{investment} \end{array} \right) \left( \begin{array}{l} \text{project} \\ \text{life} \end{array} \right)} \times 100 \quad (2-44)$$

The INRR is recommended as the primary criterion of profitability, and is appropriate when the alternatives being considered have different investment levels.

**Depreciation.** Estimation of depreciation charges may be based on

1. The cost of the operation
2. a tax allowance
3. a means of building up a fund to finance plant replacement or,
4. a measure of falling value.

The annual depreciation charge can be calculated using ‘straight line’ depreciation, and is expressed as

$$D = \frac{C_{\text{FC}} - S}{n} \quad (2-45)$$

where

$D$  = annual depreciation

$C_{FC}$  = initial fixed capital cost

$n$  = number of years of projected life

$S$  = salvage value.

Assuming that  $C_{FC}$  is the initial fixed capital investment, and  $S$  is the projected salvage value at the end of  $n$  years of projected life, the depreciated rate  $d_j$  for any particular year  $j$  is

$$D_j = (C_{FC} - S) d_j \quad (2-46)$$

where

$D_j$  = annual depreciation charge.

With the straight line calculation procedure, where  $d_j$  is constant, combining Eqs (2-45) and (2-46) gives

$$d = \frac{1}{n} \quad (2-47)$$

The various component of a plant such as equipment, buildings, and improvements are characterized by projected life times. During this period, each item depreciates from its initial investment cost  $C_{FC}$  to a salvage value  $S$  over the period of  $n$  years of its projected life time. At the end of any particular year  $k$ , the depreciated value or book value  $V_k$  is

$$V_k = C_{FC} - \sum_1^k D_j \quad (2-48)$$

Substituting Eq. (2-46) into Eq. (2-48) gives

$$\begin{aligned} V_k &= C_{FC} - \sum_1^k (C_{FC} - S) d_j \\ &= C_{FC} - (C_{FC} - S) \sum_1^k d_j \end{aligned} \quad (2-49)$$

For the straight line depreciation procedure,

$$\sum_1^k d_j = \sum_1^k d = kd = \frac{1}{n} \quad (2-50)$$

and

$$V_k = C_{FC} - \frac{k}{n} (C_{FC} - S) \quad (2-51)$$

**Double Declining Balance (DDB) Depreciation.** In practice, equipment and complete plants depreciate and lose value more rapidly in the early stages of life. The depreciation based upon the declining book value balance can be expressed as

$$D_j = d_j \cdot V_{j-1} \quad (2-52)$$

The rate of depreciation  $d_j$  is the same for each year  $j$ ; however, the depreciation charges decrease each year since the book value decreases each year. For a declining balance method, the depreciation rate of decline is up to, but no more than, twice the straight line rate. This is given by

$$d_j = d = \frac{2}{n} \quad (2-53)$$

**Capitalized Cost.** The capitalized cost  $C_K$  of a piece of equipment of a fixed capital cost  $C_{FC}$  having a finite life of  $n$  years and an annual interest rate  $i$  is defined by

$$(C_K - C_{FC}) (1 + i)^n = C_K - S \quad (2-54)$$

where

$S$  = salvage or scrap value.

$C_K$  is in excess of  $C_{FC}$  by an amount which when compounded at an annual interest rate  $i$  for  $n$  years will have a future worth of  $C_K$  less the salvage or scrap value  $S$ . If the renewal cost of the equipment and the interest rate are constants at  $(C_{FC} - S)$  and  $i$ , then  $C_K$  is the amount of the capital required to replace the equipment in perpetuity.

Rearranging Eq. (2-54) gives

$$C_K = \left[ C_{FC} - \frac{S}{(1+i)^n} \right] \left[ \frac{(1+i)^n}{(1+i)^n - 1} \right] \quad (2-55)$$

or

$$C_K = (C_{FC} - S \cdot f_d) f_K \quad (2-56)$$

where

$f_d$  = discount factor

$f_K$  = the capitalized cost factor

$$= \frac{(1+i)^n}{(1+i)^n - 1}.$$

**The Average Rate of Return.** The average rate of return (ARR) method averages out the cash flow over the life of the project. This is defined by

$$ARR = \frac{\text{the average cash flow}}{\text{original Investment}} \times 100 \quad (2-57)$$

The higher the percentage value of the ARR, the better the profitability of the project.

**Present Value Ratio (Present Worth Ratio).** This commonly used profitability index in conjunction with the NPV method shows how closely a project has met the criterion of economic performance. This index is known as the present value ratio (PVR) or present worth ratio (PWR) and is defined as

$$PVR = \frac{\text{present value of all positive cash flows}}{\text{present value of all negative cash flows}} \quad (2-58)$$

The discounted cumulative cash position, which is commonly referred to as the NPV or net present worth (NPW) of the project is defined as

$$NPV = \text{cumulative discounted cash position at the end of the project.}$$

NPV of a project is greatly influenced by the level of fixed capital investment, and a better criterion for comparison of projects with different investment levels may be the present value ratio (PVR).

The PVR gives an indication of how much the project makes relative to the investment. A ratio of unity shows that the income just matches the expected income from capital invested for a given interest rate. A ratio of less than unity indicates that the income does not come up to the minimum expectations. A ratio of more than unity means that the project exceeds the minimum expectations.

## PROFITABILITY

A project is profitable if its net earnings are greater than the cost of capital. In addition, the larger the additional earnings, the more profitable the venture, and therefore the greater the justification for putting the capital at risk. A profitability estimate attempts to quantify the risk taken. The methods used to assess profitability are

1. return on investment
2. payback period
3. net present value
4. discounted cash flow rate of return
5. net return rate
6. equivalent maximum investment period
7. interest recovery period
8. rate of return on depreciated investment
9. rate of return on average investment
10. capitalized cost
11. average rate of return
12. present value ratio.

Abrams [41] has listed other methods for assessing project profitability.

### EXAMPLE 2-8

Consider a plant costing \$1,000,000 to build that produces a product A. For the same capital outlay a different plant can be erected to produce an alternative product B. Conditions are such that each plant will only be in operation for eight years and then both will be scrapped. The cash flows in each of the eight years obtained by selling A and B are shown in Table 2-19. Determine which plant is more profitable.

#### Solution

The Excel worksheet (Example 2-8.xls) and Fortran computer programs (Project1.for and Project2.for) from the companion website calculate the various profitability criteria for investment plans A and B at 10% discount rate. Table 2-20 shows the profitability criteria of products A and B.

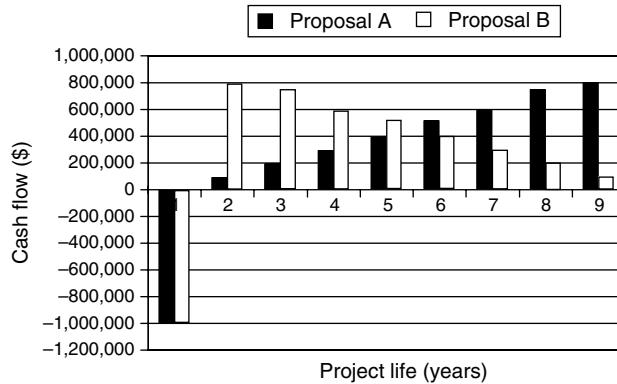
Figures 2-9a and 2-9b show plots of the cash flow and net present value against the project life, and Figure 2-9c illustrates plots of the net present value at varying interest rate for products A and B. The results in Table 2-20 show that product B is a better investment choice than product A. *The Excel spreadsheet (Example 2-8.xls) and the Fortran program source codes (project1.for and project2.for) can be downloaded from the companion website.*

**TABLE 2-19 Annual Cash Flows and Cumulative Cash Flows**

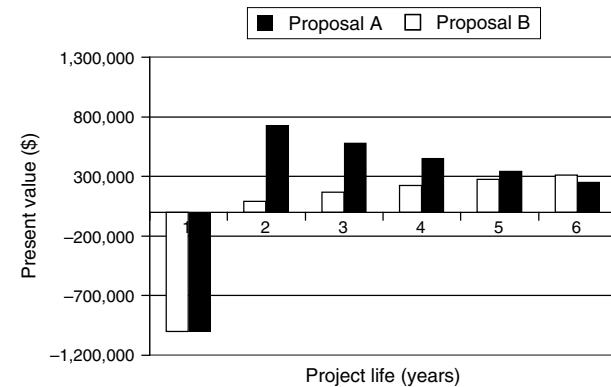
End of Year	Product A Cash Flows (\$)	Product A Cumulative Cash Flows (\$)	Product B Cash Flows (\$)	Product B Cumulative Cash Flows (\$)
0	-1,000,000	-1,000,000	-1,000,000	-1,000,000
1	100,000	-900,000	800,000	-200,000
2	200,000	-700,000	700,000	500,000
3	300,000	-400,000	600,000	1,100,000
4	400,000	0	500,000	1,600,000
5	500,000	500,000	400,000	2,000,000
6	600,000	1,110,000	300,000	2,300,000
7	700,000	1,800,000	200,000	2,500,000
8	800,000	2,600,000	100,000	2,600,000

**TABLE 2-20 Profitability Criteria of Products A and B**

Profitability Criteria	Product A	Product B
Net present value, \$	1,136,360	1,665,074
Present value ratio	2.14	2.67
Net return rate, %	14.20	20.81
Average rate of return, %	45.0	45.0
Internal rate of return, %	28.0	65.0
Discounted cash flow rate of return, %	28.26	64.86
Payback period, years	3.3	3.1
Return on investment, %	30	32.8



**Figure 2-9a** Cash flow vs project life.



**Figure 2-9b** Present value vs project life.

**EXAMPLE 2-9**

A project requires an initial capital expenditure of \$1,000,000 on the plant and \$100,000 on industrial buildings, all incurred at the same time. The project is expected to have constant annual operating costs of \$150,000 and to generate an annual income of \$500,000. The project is expected to have a five-year life. Calculate the after-tax project cash flows, NPV, IRR, and PVR if

the plant will have no final scrap value  
the final plant scrap value is \$200,000  
the final plant scrap value is \$400,000

**Solution**

The Excel worksheet (Example 2-9.xls) provides cells for the initial capital plant investment, expenditure on buildings, operating cost, income, before-tax operating profit, capital allowances, taxable profit, tax at 35%, profit after tax, and after-tax cash flow. The spreadsheet calculates the various profitability criteria before and after tax. Table 2-21 shows the profitability criteria of after-tax cash flows with the plant having no scrap value, and a scrap of \$200,000 and \$400,000 respectively. Figures 2-10a-f show plots of cumulative cash flow vs project life, net present value vs interest rate, and cash flow vs project life for the plant having no scrap value, and with scrap values of \$200,000 and \$400,000, respectively. *The Excel spreadsheet (Example 2-9.xls) can be downloaded from the companion website.*

Year No.	Plant Allowance Qualifying Expenditure (\$)	Allowance (\$)		Industrial Allowance Qualifying Expenditure (\$)	Building Allowance (\$)	Total Capital Allowances (\$)
1	1,000,000	250,000	(25%)	100,000	4,000 (4%)	254,000
2	750,000	187,500		100,000	4,000 (4%)	191,500
3	562,500	140,625		100,000	4,000 (4%)	144,625
4	421,875	105,469		100,000	4,000 (4%)	109,469
5	316,406	316,406	balancing	100,000	4,000 (4%)	320,406
Total allowance		1,000,000			20,000	1,020,000

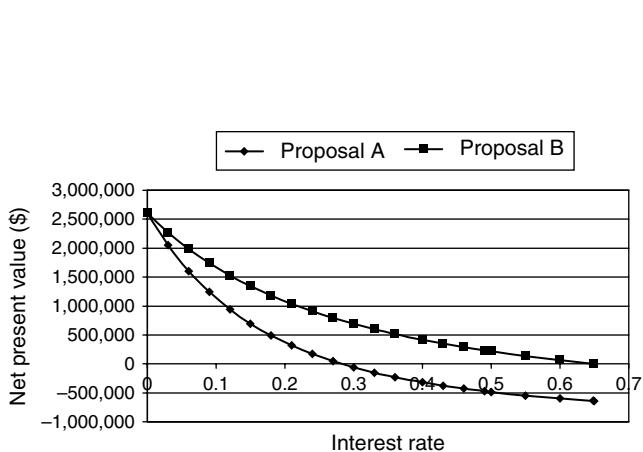


Figure 2-9c Net present value vs interest rate.

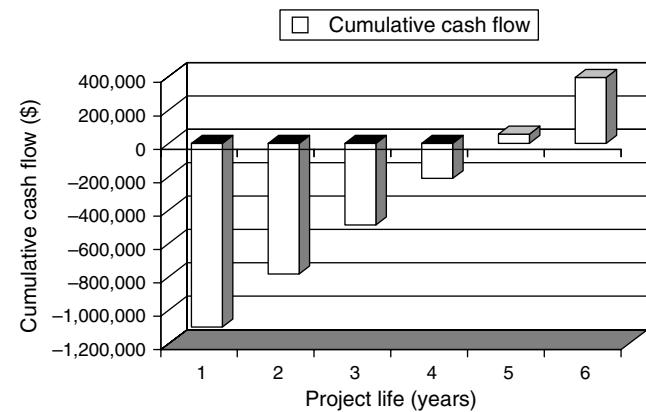


Figure 2-10a Cumulative cash flow (No scrap value).

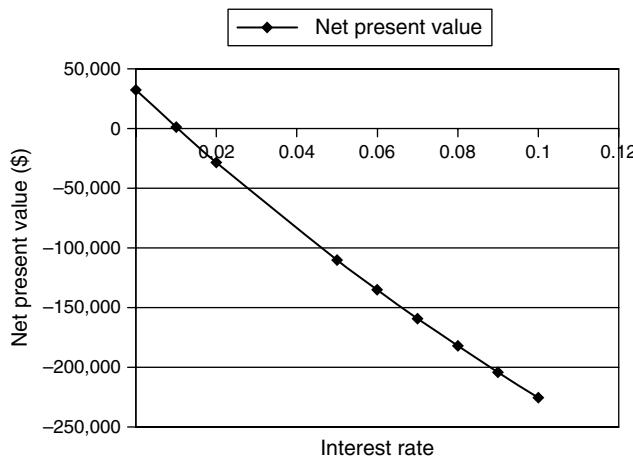
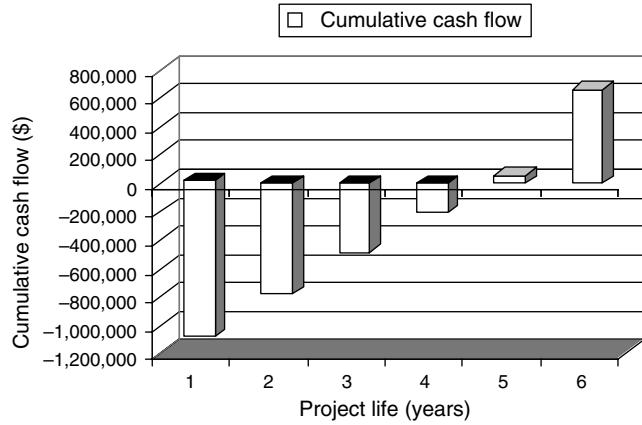
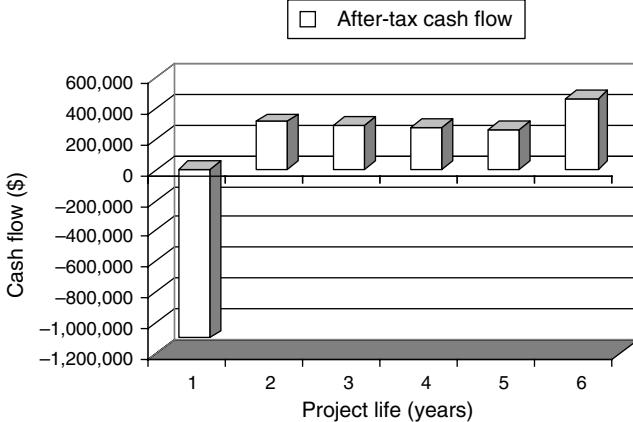
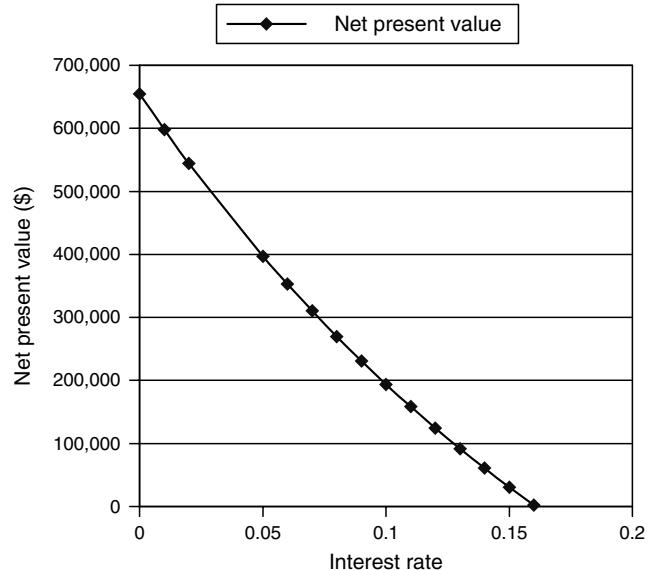
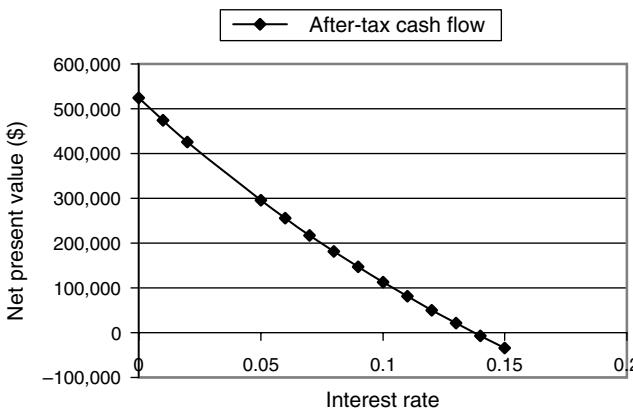
**TABLE 2-21 Profitability Analysis of After-Tax Cash Flows**

Scrap value (\$)	IRR (%)	NPV (\$)	PVR	DCFRR
0	11	32,446	1.03	0.0104
200,000	14	113,166	1.1	0.1376
400,000	16	193,886	1.18	0.1608

**ECONOMIC ANALYSIS**

Computer programs (project1.for and project2.for) have been developed to estimate NPV, PVR, NRR, ARR, PBP, and DCFRR. These analyses are performed for a given cash flow over the operating life of a project. These programs can be incorporated as subroutines into larger programs, if required. In addition, a detailed computer program has been developed to review an economic project using Kirkpatrick's [42] input data. These data are defined as follows:

*Annual revenue, \$:* The money received (sales minus the cost of sales) for one-year production from the plant. This is assumed as being constant for life of the project.

**Figure 2-10b** Net present value vs interest rate (No scrap value).**Figure 2-10e** Cumulative cash flow (Scrap value = \$400,000).**Figure 2-10c** After-tax cash flow (Scrap value = \$200,000).**Figure 2-10f** Net present value vs interest rate (Scrap value = \$400,000).**Figure 2-10d** Net present value vs interest rate (Scrap value = \$200,000).

**Annual operating cost, \$:** The cost of, for example, raw materials, labor, utilities, administration, insurance, and royalties, but does not include debt service payments.

**Depreciating base, \$:** The capitalized cost of the facility, less non-depreciable items, for example land and inventory. No salvage is subtracted since double declining balance depreciation is used.

**Project life, years:** The length of time for which the facility is to be operated. It is also the term of the loan and the depreciation time.

**Initial loan, \$:** The capitalized cost minus owner equity.

**Payments/year:** The number of payments on the loan per year: 1 – annually, 2 – biannually, 4 – quarterly, and 12 – monthly.

**Periodic interest rate:** The annual interest rate divided by the payments per year. For a 10% annual interest rate and monthly repayments, this is  $0.10/12$ .

**Investment tax credit, \$:** The percentage of the initial investment allowed as a tax credit in the year the investment is made.

**Tax rate:** The percentage tax that must be paid on the project's pre-tax income. This rate is assumed to remain constant during the life of the project.

**Debt service/period, \$:** The amount of each loan payment, that is,

$$\frac{(\text{Loan}) \left[ \frac{\text{annual interest}}{\text{payments per year}} \right]}{1 - \left[ 1 + \frac{\text{annual interest}}{\text{payments per year}} \right]^{-(\text{years} \times \frac{\text{payments}}{\text{year}})}}$$

**The salvage value (or scrap value):** The projected salvage value of equipment at the end of the project lifetime is that portion of the fixed capital that cannot be depreciated.

**Land value:** Land is not considered depreciable, on the assumption that it can be used indefinitely for succeeding projects on a specific site, or it can be sold.

**Working capital:** The capital invested in various necessary inventoried times, which are recoverable.

The calculations for every year of the project life are the following:

1. Calculate the depreciation as double declining (that is twice the depreciation base divided by project life) and subtract it from the depreciation base.
2. Calculate the yearly interest and principal of the debt service payments.
3. Subtract from the revenue, the operating cost, annual depreciation, and interest.
4. Subtract from positive pre-tax income, until no negative pre-tax income remains (if the pre-tax income is negative or has been negative in a prior year).
5. Calculate the income tax due at the given rate, and apply the investment tax credit against the income tax due until the credit is exhausted (if after step 4, the pre-tax income is still positive).
6. Deduct the income tax remaining after step 5 from the pre-tax income, leaving the after-tax income.
7. Add the after-tax income to the depreciation for the year, yielding the cash flow.
8. Determine the loan balance at the end of the year.
9. Determine the amount of unused depreciation after the last year of the project's life. Since double declining depreciation does not totally exhaust the depreciation account, the unused depreciation should be added to the cash flow of the last year as salvage value recovered at the termination of the project under consideration.
10. Calculate the NPV for a known discount rate.
11. Determine the PVR.
12. Calculate the NRR.
13. Calculate the ARR.
14. Estimate the PBP.

For a given discount rate, calculate the discount factor during the operating life of the project. Multiply the discount factor by the cash flow and obtain the NPV and the NRR. Valle-Riestra [36] has listed alternative methods of calculating cash flow, as shown in Table 2-22.

### EXAMPLE 2-10

Calculate the yearly return of investment on the financial data shown in Table 2-23

#### Solution

The computer program (project3.for) calculates the following: the yearly cash flows, cumulative cash flows, PV, NPV, PVR, ARR, NRR, and the PBP. Table 2-24 gives both the input data and the computer output for the cash flows during 10 years of the project life, and at a discount rate of 5%. The calculated salvage value of the investment is \$4,294,967. The cash flows generated by the computer program (project3.for) using the financial data in Table 2-23 are then used to determine the DCFRR using the computer program (project2.for). Table 2-25 lists the input data and the computer output for DCFRR of the investment as 13.42%. The Excel worksheet (Example 2-10.xls) calculates NPV, IRR, PVR, PB, and ROI using the cash flows generated from the fortran program (project3.for). Figures 2-11a-c respectively show the various plots of the economic evaluations. *The Fortran program source codes (project2.for and project3.for) and the Excel worksheet (Example 2-10.xls) can be downloaded from the companion website.*

**TABLE 2-22 Methods of Computing Cash Flow from a Project**

Method 1	Method 2
1. Cost of Manufacture (COM) includes depreciation	1. Cost of Manufacture (COM) includes depreciation
2. General expenses	2. Depreciation
3. Total operating expense (3) = (1) + (2)	3. COM less depreciation (3) + (1) - (2)
4. Total sales	4. General expenses
5. Profit before taxes	5. Out of pocket expenditure
6. Income taxes (6) ≈ 1/2(5)	6. Total sales
7. Profit after tax (7) = (5) - (6) ≈ 1/2(5)	7. Cash flow before taxes (7) = (6) - (5)
8. Depreciation	8. Profit before taxes (8) = (7) - (2)
9. Continuous cash flow from project (9) = (7) + (8)	9. Income tax (9) ≈ 1/2(8)
10. Calculate the NPV for a known discount rate.	10. Profit after tax (10) = (8) - (9) ≈ 1/2(8)
11. Determine the PVR.	11. Continuous cash flow from project (11) = (10) + (2)

(Source: Valle-Riestra [36].)

Note: Methods 1 and 2 in this table give identical results.

**TABLE 2-23 Financial Data**

1	Annual revenue, \$	9,000,000
2	Annual operating cost, \$	1,200,000
3	Depreciated base, \$	40,000,000
4	Project life, years	10
5	Initial loan, \$	30,000,000
6	Number of payment per year	12
7	Annual interest, %	10
8	Investment credit, %	10
9	Tax rate, %	50
10	Rate of return, %	5

**TABLE 2-24 Input Data and Computer Results of Example 2-10**

DATA3.DAT

```
-----
9000000
1200000
40000000
10
30000000
12
10
10
50
5
```

## Computer Results

## ECONOMIC EVALUATION OF A PROJECT:

```
*****
1      ANNUAL REVENUE $:          9000000.00
2      ANNUAL OPERATING COST $:  1200000.00
3      DEPRECIATION BASE $:     40000000.00
4      PROJECT LIFE, YEARS:    10.
5      INITIAL LOAN $:         30000000.00
6      NUMBER OF PAYMENTS PER YEAR: 12.
7      ANNUAL INTEREST, %:      10.
8      INVESTMENT CREDIT, %:    10.
9      TAX RATE, %:            50.
10     DISCOUNT RATE %:        5.00
*****
```

## YEAR 1

```
*****
REVENUE $:          9000000.00
OPERATING COST $:   1200000.00
DEPRECIATION $:     8000000.00
INTEREST $:         2917172.00
PRE-TAX INCOME $:   -3117172.00
TAX AT 50.%, $:     0.00
AFTER TAX INCOME $: -3117172.00
CASH FLOW $:        4882828.00
*****
```

## YEAR 2

```
*****
REVENUE $:          9000000.00
OPERATING COST $:   1200000.00
DEPRECIATION $:     6400000.00
INTEREST $:         2724474.40
PRE-TAX INCOME $:   -1324474.40
TAX AT 50.%, $:     0.00
AFTER TAX INCOME $: -1324474.40
CASH FLOW $:        5075526.00
*****
```

(continued)

**TABLE 2-24—(continued)**

## YEAR 3

REVENUE \$:	9000000.00
OPERATING COST \$:	1200000.00
DEPRECIATION \$:	5120000.00
INTEREST \$:	2511598.20
PRE-TAX INCOME \$:	168401.74
TAX AT 50.%, \$:	0.00
AFTER TAX INCOME \$:	168401.74
CASH FLOW \$:	5288402.00

## YEAR 4

REVENUE \$:	9000000.00
OPERATING COST \$:	1200000.00
DEPRECIATION \$:	4096000.00
INTEREST \$:	2276432.20
PRE-TAX INCOME \$:	1427568.00
TAX AT 50.%, \$:	0.00
AFTER TAX INCOME \$:	1427568.00
CASH FLOW \$:	5523568.00

## YEAR 5

REVENUE \$:	9000000.00
OPERATING COST \$:	1200000.00
DEPRECIATION \$:	3276800.00
INTEREST \$:	2016641.00
PRE-TAX INCOME \$:	2506559.20
TAX AT 50.%, \$:	0.00
AFTER TAX INCOME \$:	2506559.20
CASH FLOW \$:	5783359.00

## YEAR 6

REVENUE \$:	9000000.00
OPERATING COST \$:	1200000.00
DEPRECIATION \$:	2621440.00
INTEREST \$:	1729646.00
PRE-TAX INCOME \$:	3109796.00
TAX AT 50.%, \$:	0.00
AFTER TAX INCOME \$:	3109796.00
CASH FLOW \$:	5731236.00

(continued)

**TABLE 2-24—(continued)**

YEAR 7	
*****	
REVENUE \$:	9000000.00
OPERATING COST \$:	1200000.00
DEPRECIATION \$:	2097152.00
INTEREST \$:	1412599.00
PRE-TAX INCOME \$:	4290249.00
TAX AT 50.%, \$:	0.00
AFTER TAX INCOME \$:	4290249.00
CASH FLOW \$:	6387401.00
*****	
YEAR 8	
*****	
REVENUE \$:	9000000.00
OPERATING COST \$:	1200000.00
DEPRECIATION \$:	1677722.00
INTEREST \$:	1062353.00
PRE-TAX INCOME \$:	5059926.00
TAX AT 50.%, \$:	2229985.20
AFTER TAX INCOME \$:	2829940.20
CASH FLOW \$:	4507662.00
*****	
YEAR 9	
*****	
REVENUE \$:	9000000.00
OPERATING COST \$:	1200000.00
DEPRECIATION \$:	1342177.00
INTEREST \$:	675431.50
PRE-TAX INCOME \$:	5782391.00
TAX AT 50.%, \$:	2891195.40
AFTER TAX INCOME \$:	2891195.40
CASH FLOW \$:	4233373.00
*****	
YEAR 10	
*****	
REVENUE \$:	9000000.00
OPERATING COST \$:	1200000.00
DEPRECIATION \$:	1073742.00
INTEREST \$:	247994.34
PRE-TAX INCOME \$:	6478264.00
TAX AT 50.%, \$:	3239132.00
AFTER TAX INCOME \$:	3239132.00
CASH FLOW \$:	4312874.00
*****	
SALVAGE VALUE \$:	4294967.00

(continued)

**TABLE 2-24—(continued)**

ECONOMIC PROJECT EVALUATION NET PRESENT VALUE AT A GIVEN DISCOUNT RATE				
***** 11 YEARLY CASH FLOWS INCLUDING YEAR 0				
5.00 PERCENTAGE ANNUAL DISCOUNT RATE				
YEARLY CASH FLOWS				
YEAR	CASH FLOW	CUMULATIVE CASH FLOW	DISCOUNT FACTOR	PRESENT VALUE
0	-30000000.0	-30000000.0	1.0000	-30000000.0
1	4882828.0	-25117172.0	0.9524	4650313.0
2	5075526.0	-20041650.0	0.9070	4603652.0
3	5288402.0	-14753244.0	0.8638	4568321.0
4	5523568.0	-9229676.0	0.8227	4544254.0
5	5783359.0	-3446317.0	0.7835	4531414.0
6	5731236.0	2284919.0	0.7462	4276738.0
7	6387401.0	8672320.0	0.7107	4539409.0
8	4507662.0	13179982.0	0.6768	3050964.2
9	4233373.0	17413360.0	0.6446	2728871.2
10	4312874.0	21726230.0	0.6139	2647732.2
11	8589934.0	30316163.0	0.5847	5022359.0

THE NET PRESENT VALUE (\$): 15164030.  
 PRESENT VALUE RATIO: 1.505  
 THE NET RETURN RATE: 4.60 %  
 THE AVERAGE RATE OF RETURN: 18.28 %  
 THE PAYBACK PERIOD IS BETWEEN: 5 AND 6 YEARS

**TABLE 2-25 Input Data and Computer Output for the Discount Cash Flow Rate of Return (DCFRR) of Example 2-10**

DATA3.DAT

```

12
-30000000
4882828
5075526
5288402
5523568
5783359
6387401
4507662
4233373
4312874
8589934

```

## DISCOUNT CASH FLOW RATE OF RETURN CALCULATION

***** 12 YEARLY CASH FLOWS INCLUDING YEAR 0	
YEAR	CASH FLOW
0	-30000000.0
1	4882828.0
2	5075526.0
3	5288402.0
4	5523568.0
5	5783359.0
6	5731236.0
7	6387401.0
8	4507662.0
9	4233373.0
10	4312874.0
11	8589934.0

The Discount Cash Flow Rate of Return (%): 13.424

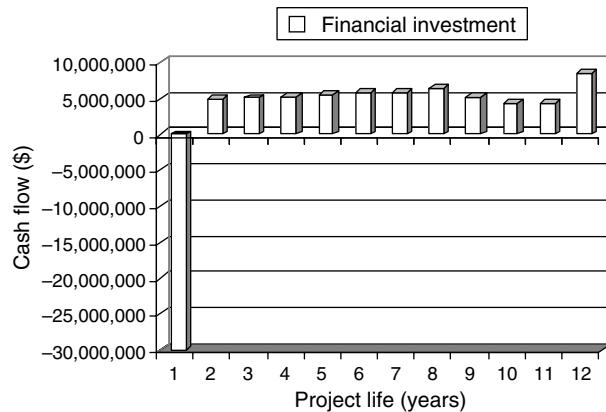


Figure 2-11a Cash flow vs project life.

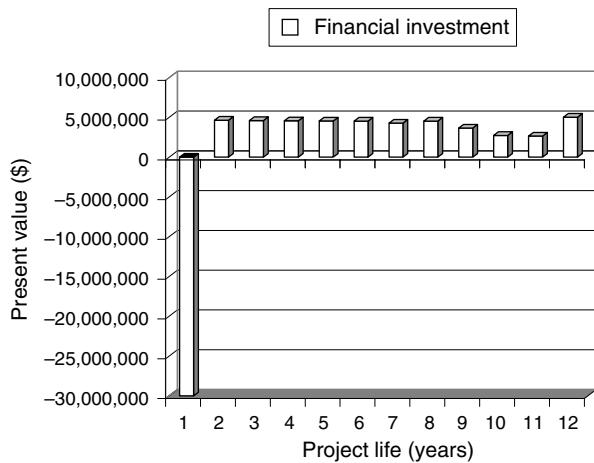


Figure 2-11b Present value vs project life.

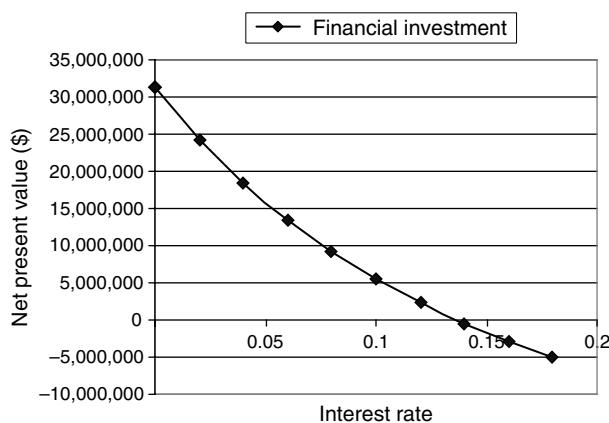


Figure 2-11c Net present value vs interest rate.

Cash flows into a project can be time-dependent, where the cash flows occur in a continuous process rather than on a one-time basis. Table 2-22 shows different methods of calculating cash flow from a project, and Figure 2-12 illustrates a continuous cash flow diagram. Continuous cash flows into the project are from sales revenues, and cash flows out are due to out-of-pocket expenses.

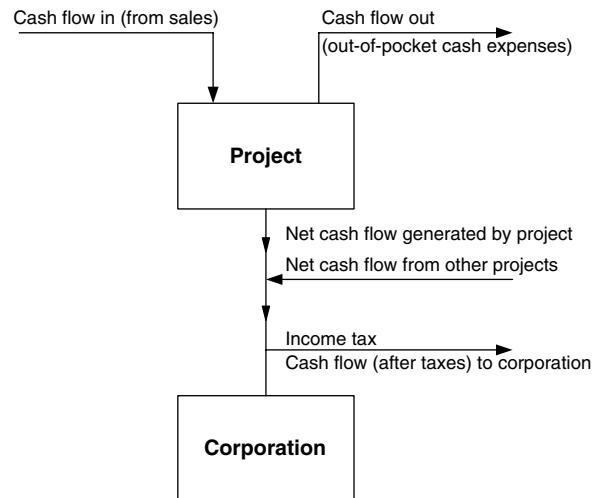


Figure 2-12 A continuous cash flow diagram. (Source: Valle-Riestra [36].)

The difference between the incoming and outgoing cash flow is the net cash flow generated by the project. The combined net cash flow generated by all the company's projects is reduced by the income tax payment to yield the net continuous cash flow. Expressing the definition of profit with the continuous cash flow results in the following:

$$\text{Profit before taxes} = (\text{continuous cash flow before taxes}) - (\text{depreciation}) \quad (2-59)$$

$$\text{Profit after taxes} = (\text{continuous cash flow after taxes}) - (\text{depreciation}) \quad (2-60)$$

It is important to remember what has sometimes been overlooked in unsuccessful ventures;

$$\text{Cash flow} \neq \text{profit} \quad (2-61)$$

Allen [43] has provided a systematic procedure for assessing investment proposals for new plants and equipment, exploiting new technology, and replacing uneconomic, inefficient, and obsolete plants, process, and equipment.

## INFLATION

A decrease in the average purchasing value of currency is referred to as inflation. If a given product cost \$100 last year and now costs \$120, then the product has suffered a 20% rate of inflation. That is, the purchasing power of the currency (i.e., of the \$120) has consequently fallen by a factor of  $(\$120 - \$100)/\$120$  or 16.7%. An inflation rate, for example, 15%, means that the average cost of goods and services will increase 15% in one year. The result is that commencing construction one year early will reduce the amount of money expended by 15%. Since the 1980s, inflation has been considered in most economic project evaluations.

When inflation is used in economic evaluations, all items except interest on a loan and depreciation are considered to increase in value at the same rate as inflation. Generally, interest is set at the time a loan is negotiated and does not change with inflation. In addition, depreciation depends on the method (e.g.,

straight line or double declining balance) used, and the capital charges incurred before start-up, are not affected by the inflation rate after start-up. While determining the profitability of a project (e.g., NPV), the interest rate is assumed to be greater than the inflation rate. Money may be lost on the project while the NPV indicates the opposite if the inflation rate is greater than the interest rate. There are cases where the interest rate is set at the expected inflation rate plus a real expected interest rate. The real expected interest rate is the interest rate that is used to calculate the NPV when there is no inflation. Alternatively, the present value is calculated using the inflation rate as the interest rate, the NPV is then determined using the real expected interest rate.

**Effect of Inflation on NPV.** The effect of inflation on NPV for a proposed project can be expressed by

$$NPV = \sum_{t=0}^{t=n} \frac{C_t}{(1+i)^t (1+i_i)^t} \quad (2-62)$$

where

$C_t$  = cash flow

$i$  = interest rate

$i_i$  = fractional rate of inflation

$n$  = number of years of project life.

Equation (2-62) allows all the net annual cash flows to be corrected to their purchasing power in year 0. If the inflation rate is zero, Eq. (2-62) becomes identical with Eq. (2-40).

For many years, companies and countries have lived with the problem of inflation, or the falling value of money, while other factors (e.g., labor costs) tend to rise each year. Failure to account for these trends in predicting cash flows can lead to serious erroneous results, thereby giving misleading profitability estimates.

## SENSITIVITY ANALYSIS

When an economic evaluation has been carried out using single-value forecasts and estimates for everything which contribute to the yearly cash flows, the information obtained from it for project decision-making purposes can be extended by carrying out an economic evaluation sensitivity analysis. This explores the relative effects on the economic viability of a project of possible changes in the forecast data which contribute to the project cash flows. It focuses on the areas which are most critical in terms of any uncertainty and it indicates where confidence in forecast is most vital. Sensitivity analysis also enables the economic effects of changes in a project to be reviewed; for example, changes in fixed and variable costs resulting from the use of different equipment types, different phasing of investment, delays in plant start-up, and the effect of possible different market growth patterns. It can also be used to explore the effects on the economic viability of a project with uncertainty in different areas, but does not attempt to quantify the uncertainty in an area [37, 43]. In general, it is worthwhile to make tables or plot curves that show the effect of variations in costs and prices on profitability. Its purpose is to determine to which factors the profitability of a project is most sensitive. Sensitivity analysis should always be carried out to observe the effect of departures from expected values.

Sensitivity analysis consists of finding the effects on project NPV, PVR, and IRR of changes in the input factors selected, taken one at a time. Once the Excel worksheets (Examples 2-7.xls and 2-8.xls) have been set up with the series of data tables, cash flow tables, and NPV, IRR, and PVR calculations for the project, the entry in any cell in the input tables can be changed and the

consequent effects will automatically be recalculated and will ripple through the following tables to give the corresponding new project NPV, PVR, and IRR values. Sensitivity analysis thus consists of systematically organizing a sequence of changes in the input data and displaying the resulting changes to the project NPV, IRR, and PVR.

## NOMENCLATURE

ARR	Average rate of return
$C$	Cash flow for each year
$C_{CF}$	Net annual cash flow
$C_{FC}$	Fixed capital cost
$C_K$	Capitalized cost
$C_L$	Land cost
$C_0$	Initial investment cost
$C_n$	Cash flow
$C_{TC}$	Total capital cost
$C_{WC}$	Working capital cost
$C_2$	Capital cost of the designed plant
$C_1$	Capital cost of the existing plant
$D$	Annual depreciation
DCFRR	Discounted cash flow rate of return
$D_j$	Annual depreciation charge
$d_j$	Depreciation rate
$f_d$	Discount factor
$f_K$	Capitalized cost factor
$I$	Investment cost
IRR	Internal rate of return
$i$	Interest rate of return (ROI/100)
$m$	Exponential power for cost capacity relationships, project life
NPV	Net present value
NRR	Net return rate
$n$	Years of project life
$p$	Initial principal
PBP	Payback period
PV	Present value
PVR	Present value ratio
PWR	Present worth ratio
$Q_2$	Capacity of the designed plant
$Q_1$	Capacity of existing plant
ROI	Return of investment
$S$	Salvage value

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4. [www.galorath.com](http://www.galorath.com)
5. [www.ce.cmu.edu/pmbook/05\\_cost\\_estimation.html](http://www.ce.cmu.edu/pmbook/05_cost_estimation.html)

## PHYSICAL PROPERTIES OF LIQUIDS AND GASES

**A**n important but time-consuming factor in practically every design situation and in development of flowsheets is the collection and assembly of physical property data for the components of the system in question. Often it is not sufficient to obtain single data points from various tables, since many designs cover rather wide ranges of temperature and pressure, and the effects of these on the properties must be taken into account.

Data may be located in many useful handbooks [1–4] as well as published technical papers and company compilations. However, experience indicates that extensive literature searches may be necessary to locate specific data on a particular compound. It is surprising to find so many common compounds for which the data are incomplete and sometimes inaccurate. Empirical correlations must often be utilized, sometimes to generate a value and sometimes to check a questionable literature value.

Therefore, when developing an estimate of process engineering it is important to recognize the amount of effort that may be necessary to collect physical property data before the flowsheet calculations can commence. This same concern exists when evaluating K values and activity coefficient data for phase equilibrium.

Physical property data of compounds are frequently required for pilot plant studies or for the design of large process plants. These data are not readily available except in some simulation design packages or in the largest libraries. The data can, of course, be obtained by conducting experiments to measure the properties of individual substances or of mixtures, which may exhibit non-ideal behavior. However, this is often time-consuming and expensive. Therefore the process engineer may resort to correlations and interpolations to obtain data of sufficient accuracy for design purposes. Thermodynamic data of this type are required in most calculations such as sizing vessels, process pipeline sizing, and separation of multicomponents, gas absorption, and chemical reactor design.

Many predictions and correlations of thermodynamic physical data have been published. This chapter reviews physical property data for liquids and gases and presents the Excel spreadsheet programs for determining these properties for a range of temperature and correlation constants, which can be downloaded from the companion website. Appendix C provides tables of some selected physical property data.

### 3.1 DENSITY OF LIQUIDS

Liquid density data are essential in process engineering design such as sizing of storage vessels that contain the basic raw materials and products for a plant, in process piping design involving either single-phase incompressible fluids, compressible fluids or two-phase flow mixtures. In distillation, absorption, or stripping, liquid density data are required in the determination of flooding and sizing of column diameter. Additionally, liquid density usage is encountered in various heat-, mass-, and momentum-transfer operations. Saturated liquid densities at any temperature are based upon the modified form of the Rackett equation as [5]

$$\text{Density} = AB^{-\left(1 - \frac{T}{T_C}\right)^n} \quad (3-1)$$

where

- Density = saturated liquid density, g/mL
- $A$ ,  $B$ , and  $n$  = regression coefficients for chemical compound
- $T$  = temperature, K
- $T_C$  = critical temperature, K
- $T_r$  = Reduced temperature,  $T/T_C$ .

Appendix C gives tabulated values of the correlation constants and results for saturated liquid density between the minimum and the maximum temperature ranges for which the equation is applicable, and denoted by  $T_{\min}$  and  $T_{\max}$ . For many of the compounds, the temperature range corresponds to the freezing point and critical

point. Values for liquid density at 25°C are provided in the last column. The Excel spreadsheet program from the companion website (density-of-liquid.xls) provides calculations for the density of saturated liquids and plots with given correlation constants, temperatures, and at their critical temperatures. Table C-1 in Appendix C lists the density of chemical compounds, and Figure 3-1 shows the plot for saturated liquid density of methanol as a function of temperature.

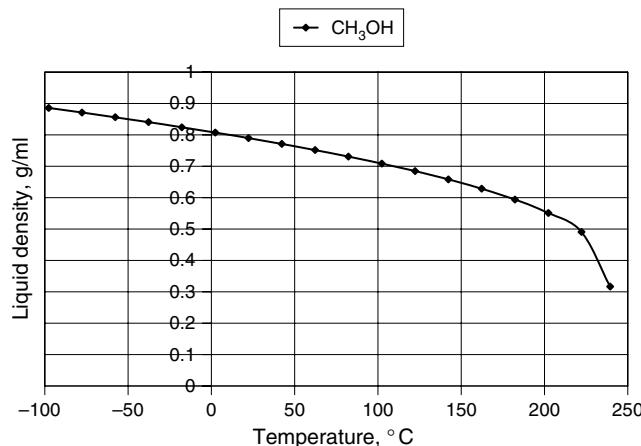


Figure 3-1 Saturated liquid density of Methanol ( $\text{CH}_3\text{OH}$ ).

**EXAMPLE 3-1**

Calculate the liquid density of methanol ( $\text{CH}_3\text{OH}$ ) at a temperature of  $120^\circ\text{C}$  (393.15 K).

*Solution*

Substituting the regression coefficients ( $A$  and  $B$ ) from Table C-1 in Appendix C, and the temperature at 393.15 K into the equation for liquid density yields

$$\begin{aligned} \text{density} &= AB^{-\left(1 - \frac{T}{T_C}\right)^n} \\ &= (0.27197) (0.27192)^{-\left(1 - \frac{393.15}{512.58}\right)^{0.2331}} \\ &= 0.69 \text{ g/mL} \end{aligned}$$

### 3.2 VISCOSITY OF GAS

Gas phase viscosity data,  $\mu_G$ , are used in the design of compressible fluid flow and in unit operations. For example, the viscosity of a gas is required to determine the maximum permissible flow through a given process pipe size. Alternatively, the pressure loss of a given flow rate can be calculated. Gas viscosity data are needed for the design of process equipment involving heat-, momentum-, and mass-transfer operations. The correlation for gas viscosity as a function of temperature is expressed by [2]:

$$\mu_G = A + BT + CT^2 \quad (3-2)$$

where

\*  $\mu_G$  = viscosity of gas, micropoise ( $\mu\text{P}$ )  
 $A$ ,  $B$ , and  $C$  = regression coefficients for chemical compound  
 $T$  = temperature, K.

The Herring-Zipperer correlation for the calculation of gas mixture viscosity at low pressure is [5]:

$$\mu_m = \frac{\sum MF_i \mu_i (MW_i)^{0.5}}{\sum MF_i (MW_i)^{0.5}} \quad (3-3)$$

where

\*  $\mu_m$  = gas mixture viscosity, micropoise ( $\mu\text{P}$ )  
 $MF_i$  = mole fraction of component  $i$   
 $\mu_i$  = gas viscosity of component  $i$ , micropoise ( $\mu\text{P}$ )  
 $MW_i$  = molecular weight of component  $i$ .

Reid et al. [6] have reviewed other correlation methods for determining  $\mu_G$  at low pressure. The Excel spreadsheet program from the companion website (viscosity-of-gas.xls) calculates the gas viscosity between the range of the minimum and the maximum temperatures denoted by  $T_{\min}$  and  $T_{\max}$  for any chemical compound. Table C-2 in Appendix C lists the gas viscosity of compounds, and Figure 3-2 shows the graph for viscosity of propane as a function of temperature. The plot gives a linear relationship between viscosity and temperature. Low pressure gas viscosities (i.e., at about 1 bar) vary in a nearly linear manner with temperature.

\* Conversion factor  
\* Conversion factor

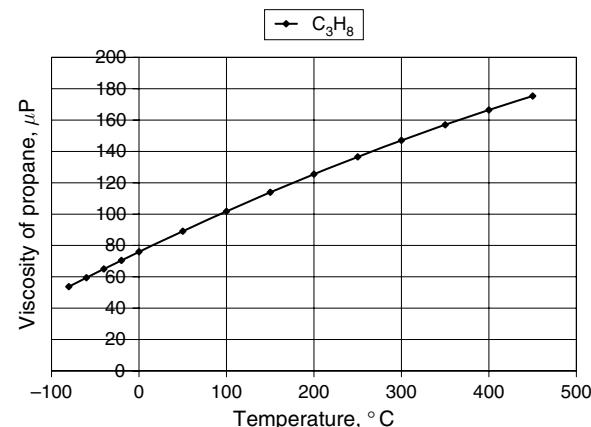


Figure 3-2 Viscosity of propane ( $\text{C}_3\text{H}_8$ ).

**EXAMPLE 3-2**

Calculate the gas viscosity of propane ( $\text{C}_3\text{H}_8$ ) at a temperature of 350 K.

*Solution*

Substituting the regression coefficients ( $A$ ,  $B$ , and  $C$ ) from Table C-2 in Appendix C, and the temperature at 350 K into the correlating equation for the gas viscosity yields

$$\mu_G = (-5.462) + (0.32722)(350) + (-1.0672 \times 10^{-4})(350)^2$$

$$\mu_G = 95.99 \mu\text{P}$$

### 3.3 VISCOSITY OF LIQUIDS

Liquid viscosity data are important in many engineering applications in the petroleum refining and petrochemical industries. It can be applied in fluid mixing: fluid flow, heat- and momentum-transfer operations. The viscosities of liquids are larger than those of gases at the same temperature. Daubert et al. [7] and Yaws et al. [8, 9] have compiled tabulations of constants that allow estimation of liquid viscosities. When these constants are derived from experimental data, they can be used with confidence; otherwise, caution should be exercised when based on estimated viscosities. The correlation for liquid viscosity as a function of temperature is given by

$$\log_{10} \mu_L = A + \frac{B}{T} + CT + DT^2 \quad (3-4)$$

where

\*  $\mu_L$  = viscosity of liquid, centipoises (cP)  
 $A$ ,  $B$ ,  $C$ , and  $D$  = regression coefficients for chemical compound  
 $T$  = temperature, K.

The Excel spreadsheet program from the companion website (viscosity-of-liquid.xls) calculates liquid viscosity of compounds between the range of the minimum and the maximum temperatures, and denoted by  $T_{\min}$  and  $T_{\max}$ , and at  $25^\circ\text{C}$ . Table C-3 in Appendix C shows the viscosity of liquid, and Figure 3-3 shows

\* Conversion factor

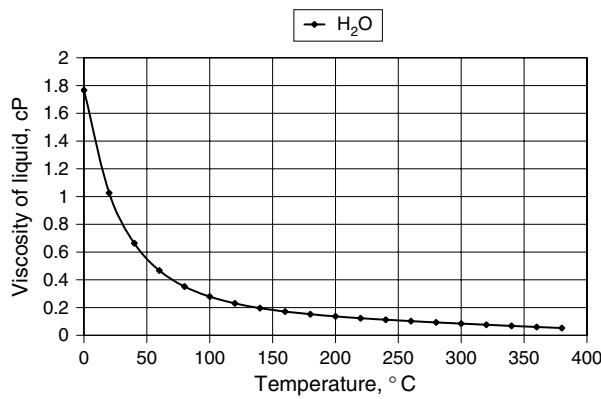


Figure 3-3 Viscosity of water ( $H_2O$ ).

the plot of viscosity of water between 273.15 and 643 K, and it decreases as the temperature increases.

Since viscosity is defined as a shearing stress per unit area divided by a velocity gradient, its dimension is (force)(time)/(length)<sup>2</sup> or mass/(length)(time). Both dimensional groups are used, although for most scientific work, viscosities are expressed in poises, centipoises, micropoises, and so on. A poise (P) denotes a viscosity of 0.1 N s/m<sup>2</sup> and 1.0 cP = 0.01 P. The following conversion factors apply to viscosity units:

$$\begin{aligned} 1 \text{ P} &= 100 \text{ cP} = 1.0 \times 10^6 \mu\text{P} = 0.1 \text{ N s/m}^2 \\ &= 1 \text{ g/(cm s)} = 0.01 \text{ Pa s} \\ &= 6.72 \times 10^{-2} \text{ lb-mass/(ft s)} = 242 \text{ lb-mass/(ft h)} \end{aligned}$$

$$1 \text{ cP} = 1 \text{ mPa s}$$

The *kinematic viscosity* is the ratio of the viscosity to the density. For example, if viscosity is in poise, and the density in grams per cubic centimeter, the unit of kinematic viscosity is the *stoke*, having the unit square centimeters per second (cm<sup>2</sup>/s). In SI system of units, viscosities are expressed in N s/m<sup>2</sup> or (Pa s) and kinematic viscosities in either m<sup>2</sup>/s or cm<sup>2</sup>/s.

#### EXAMPLE 3-3

Calculate the liquid viscosity of water at a temperature of 250°C (523.15 K).

*Solution*

Substituting the regression coefficients ( $A$ ,  $B$ ,  $C$ , and  $D$ ) from Table C-3 in Appendix C, and the temperature at 523.15 K into the correlation equation gives

$$\begin{aligned} \log_{10} \mu_L &= (-10.2158) + \frac{1792.5}{523.15} + (0.01773)(523.15) \\ &\quad + (-1.2631 \times 10^{-5})(523.15)^2 \\ \mu &= 10^{-0.9709} \\ &= 0.1069 \text{ cP} \end{aligned}$$

## 3.4 HEAT CAPACITY OF GAS

The heat capacity of gases  $C_p$  is essential for some process engineering design, for example, in the design of air-cooled heat exchanger and in the engineering design of chemical processes. In gas-phase

chemical reactions, the heat capacity is required to determine the energy (heat) necessary to bring the chemical reactants up to reaction temperature. The heat capacity  $C_p$  is also used in the rating of heat exchangers and energy balance computations. The heat capacity of a mixture of gases may be found from the heat capacities of the individual components contained in the mixture. The correlation for heat capacity of the ideal gas is expressed by Yaws et al. [10]:

$$C_p = A + BT + CT^2 + DT^3 + ET^4 \quad (3-5)$$

where

$$C_p = \text{heat capacity of ideal gas, J/(mol K)}$$

$A$ ,  $B$ ,  $C$ ,  $D$ , and  $E$  = regression coefficients for chemical compound

$$T = \text{temperature, K.}$$

The results for heat capacity of gaseous compounds as given in Table C-4 of Appendix C are based on regression experimental data, and estimates from an extensive literature search. The numerous data points were processed with a generalized least-squares computer program for minimizing the deviations. The developed Excel spreadsheet program from the companion website (heat-capacity-of-gas.xls) calculates the heat capacity of gas between the range of the minimum and the maximum temperatures, and denoted by  $T_{\min}$  to  $T_{\max}$  for various components. Figure 3-4 shows a plot of the heat capacity of 1-butene ( $C_4H_8$ ) as a function of temperature.

#### EXAMPLE 3-4

Calculate the heat capacity of 1-butene at a temperature of 200°C (473.15 K).

*Solution*

Substituting the coefficients ( $A$ ,  $B$ ,  $C$ ,  $D$ , and  $E$ ) from Table C-4 in Appendix C and the temperature into the heat capacity gives

$$\begin{aligned} C_p &= 24.915 + (0.20648)(473.15) + (5.9828 \times 10^{-5})(473.15^2) \\ &\quad + (-1.4166 \times 10^{-7})(473.15^3) \\ &\quad + (4.7053 \times 10^{-11})(473.15^4) \\ &= 123.36 \text{ J/(mol K)} \end{aligned}$$

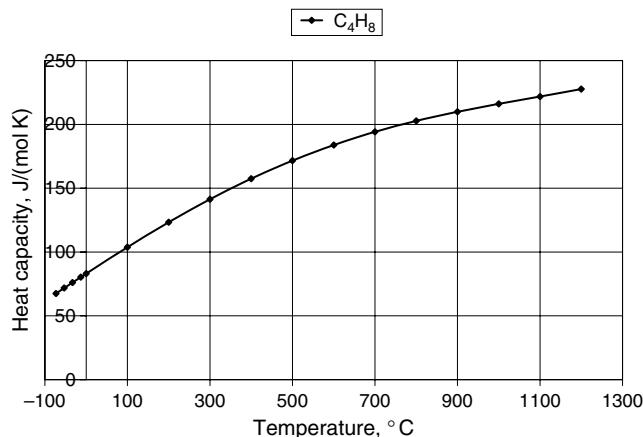


Figure 3-4 Heat capacity of 1-butene ( $C_4H_8$ ).

**EXAMPLE 3-5**

Calculate the energy required to heat gaseous ethyl chloride ( $C_2H_5Cl$ ) from 250 to 500 K.

*Solution*

From thermodynamic property, the change in enthalpy  $\Delta H$ , at constant pressure is

$$\Delta H = \int C_p dT = \int_{T_1}^{T_2} (A + BT + CT^2 + DT^3 + ET^4) dT \quad (3-6)$$

$$\Delta H = \left( A\Delta T + B\frac{\Delta T^2}{2} + C\frac{\Delta T^3}{3} + D\frac{\Delta T^4}{4} + E\frac{\Delta T^5}{5} \right)_{250}^{500} \quad (3-7)$$

$$\begin{aligned} &= 35.946 (500 - 250) + \frac{0.052294}{2} (500^2 - 250^2) \\ &\quad + \frac{0.000203}{3} (500^3 - 250^3) \\ &\quad + \frac{(-2.28 \times 10^{-7}) (500^4 - 250^4)}{4} \\ &\quad + \frac{(6.9123 \times 10^{-11}) (500^5 - 250^5)}{5} \\ &= 18368.8 \text{ J/mol.} \end{aligned}$$

### 3.5 HEAT CAPACITY OF LIQUID

Heat capacity data for liquids are relevant in process design of heat exchanger equipment and chemical reactors. In liquid-phase chemical reactions, the liquid heat capacity is required to determine the energy (heat) necessary to bring the liquid chemical reactants up to reaction temperature. Alternatively, they are essential when sizing reboilers and condensers for distillation plant and when rating heat exchangers in general, and for energy balance design calculations.

The correlation for heat capacity of liquid is expressed in polynomial of the form [2]:

$$C_p = A + BT + CT^2 + DT^3 \quad (3-8)$$

where

$C_p$  = heat capacity of saturated liquid,  
J/mol K

$A, B, C$ , and  $D$  = correlation constants for chemical compound

$T$  = temperature, K.

The Excel spreadsheet program from the companion website (heat-capacity-of-liquid.xls) calculates the heat capacity of liquid as a function of temperature between the range of the minimum and the maximum temperatures denoted by  $T_{\min}$  and  $T_{\max}$ , and at 25°C in Appendix C. Table C-5 in Appendix C lists the liquid heat capacity of compounds, and Figure 3-5 shows a plot for the heat capacity of ethylbenzene ( $C_6H_5C_2H_5$ ) as a function of temperature.

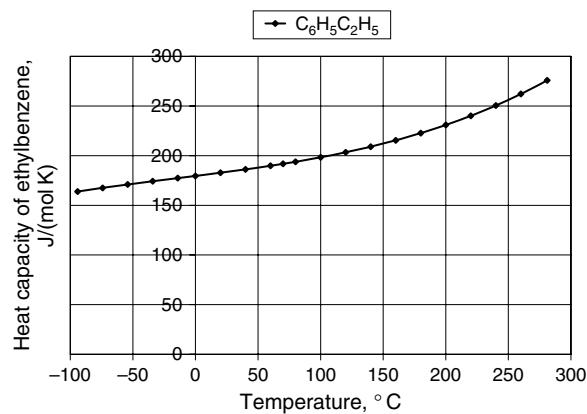


Figure 3-5 Heat capacity of ethylbenzene ( $C_6H_5C_2H_5$ ).

**EXAMPLE 3-6**

Calculate the liquid heat capacity of ethylbenzene ( $C_6H_5C_2H_5$ ) at a temperature of 298.15 K.

*Solution*

Substituting the coefficients from Table C-5 in Appendix C, and at a temperature of 298.15 K into the correlating equation for heat capacity yields

$$\begin{aligned} C_p &= 102.11 + (0.55959) (298.15) + (-1.5609 \times 10^{-3}) (298.15)^2 \\ &\quad + (2.0149 \times 10^{-6}) (298.15)^3 \\ &= 183.59 \text{ J/mol K.} \end{aligned}$$

**EXAMPLE 3-7**

Calculate the energy required to heat liquid toluene ( $C_7H_8$ ) from 350 to 600 K.

*Solution*

From thermodynamics, the change in enthalpy,  $\Delta H$  at constant pressure is

$$\Delta H = \int C_p dT = \int_{350}^{600} (A + BT + CT^2 + DT^3) dT \quad (3-9)$$

Integrating between the temperatures at 350 and 600 K gives

$$\Delta H = A\Delta T + \frac{B}{2}\Delta T^2 + \frac{C}{3}\Delta T^3 + \frac{D}{4}\Delta T^4 \quad (3-10)$$

Substituting the coefficients from Table C-5 in Appendix C, and the temperature limits into the equation yields

$$\begin{aligned} \Delta H &= (83.703) (600 - 350) + \frac{1}{2} (0.51666) (600^2 - 350^2) \\ &\quad + \frac{1}{3} (-1.491 \times 10^{-3}) (600^3 - 350^3) \\ &\quad + \frac{1}{4} (1.9725 \times 10^{-6}) (600^4 - 350^4) \end{aligned}$$

$$\Delta H = 52745.04 \text{ J/mol.}$$

### 3.6 THERMAL CONDUCTIVITY OF GAS

The thermal conductivity,  $k_G$ , of low pressure gases increases with temperature. In small temperature ranges,  $k_G$  represents some form of linear relation. However, over wide temperature ranges,  $k_G$  increases significantly more rapidly with temperature than implied by a linear function. Gas thermal conductivity data are used, for example, in the design of process equipment and unit operations involving heat transfer, such as the rating of heat exchangers and fluidized bed operations handling gases. Tsederberg [11] has compiled thermal conductivity data for pure gases; Daubert et al. [7] and Yaws [12, 13] have provided constants that estimate thermal conductivities at different temperatures. Thermal conductivity of a gas can be correlated as a function of temperature as [2]:

$$k_G = A + BT + CT^2 \quad (3-11)$$

where

$k_G$  = thermal conductivity of gas, W/(m K)  
 $A$ ,  $B$ , and  $C$  = regression coefficients for chemical compound  
 $T$  = temperature, K.

The Excel spreadsheet program from the companion website (thermal-conductivity-of-gas.xls) calculates the gas thermal conductivity for chemical compounds over the range of the minimum and the maximum temperatures denoted by  $T_{\min}$  and  $T_{\max}$ . Table C-6 in Appendix C lists the gas thermal conductivity of chemical compounds, and Figure 3-6 shows the thermal conductivity of propane ( $C_3H_8$ ) as a function of temperature.

#### EXAMPLE 3-8

Calculate the gas thermal conductivity of propane at a temperature of 500 K.

*Solution*

Substituting the correlation coefficients from Table C-6 in Appendix C and temperature of 500 K into the correlation equation gives

$$\begin{aligned} k_G &= (-0.00869) + (6.6409 \times 10^{-5}) (500) \\ &\quad + (7.8760 \times 10^{-8}) (500)^2 \\ &= 0.0442 \text{ W/(m K)}. \end{aligned}$$

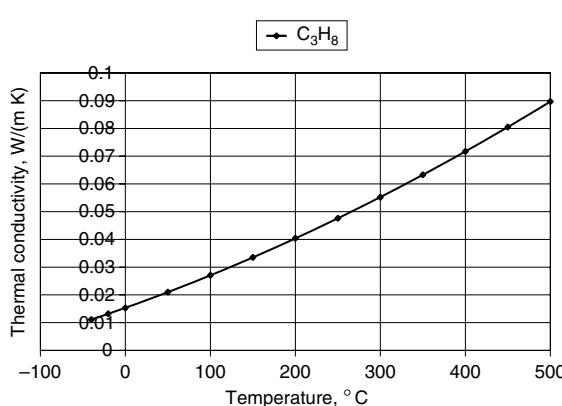


Figure 3-6 Thermal conductivity of propane ( $C_3H_8$ ).

### 3.7 THERMAL CONDUCTIVITY OF LIQUIDS AND SOLIDS

Thermal conductivity of liquids and solids is essential in many chemical and process engineering applications where heat transfer is prevalent. The liquid thermal conductivity is required to calculate the Nusselt number,  $hd/k$ , and the Prandtl number,  $c\mu/k$ , and in correlations to predict the idealized condensing film coefficient based upon laminar liquid flow over a cooled surface.

For many simple organic liquids, the thermal conductivities are between 10 and 100 times larger than those of the low pressure gases at the same temperatures, and the effect of pressure is minimal. Additionally, increasing the temperature invariably decreases the thermal conductivities, which is characteristic of those noted for liquid viscosities. Although the temperature dependence of the latter is noticeable and nearly exponential, that for thermal conductivities is weak and nearly linear [1]. Values of  $k_{\text{liq}}$  for most common organic liquid range between 0.10 and 0.17 W/(m K) at temperatures below the normal boiling point, but water, ammonia, and highly polar molecules are of values which are several times as large.

Liquid thermal conductivity data have been compiled by Jameison et al. [14] and Liley et al. [15], and constants that may be used to calculate  $k_L$  for pure liquids at different temperatures are tabulated in Daubert et al. [7], Miller et al. [16], and Yaws [13].

For inorganic compounds, the correlation for thermal conductivity of liquid and solid as a function of temperature is [2]:

$$k = A + BT + CT^2 \quad (3-12)$$

where

$k$  = thermal conductivity of liquid or solid, W/(m K)  
 $A$ ,  $B$ , and  $C$  = regression coefficients for chemical compound  
 $T$  = temperature, K.

For organic compounds, the correlation for thermal conductivity of liquid as a function of temperature is [2]:

$$\log_{10} k_{\text{liq}} = A + B \left(1 - \frac{T}{C}\right)^{2/7} \quad (3-13)$$

where

$k_{\text{liq}}$  = thermal conductivity of liquid, W/(m K)  
 $A$ ,  $B$ , and  $C$  = regression coefficients for chemical compound  
 $T$  = temperature, K.

Very limited experimental data for liquid thermal conductivities are available at temperatures in the region of the melting point. Additionally, there are very few reliable data at temperatures above a reduced temperature of  $T_r = 0.65$ . Therefore, the values in the regions of melting point and reduced temperatures above a reduced 0.65 should be considered rough approximations. The values in the intermediate region (above melting point and below reduced temperature of 0.65) are more accurate. The Excel spreadsheet program from the companion website (liquid-thermal-conductivity.xls) calculates the liquid and solid thermal conductivities of compounds between the range of the minimum and the maximum temperatures denoted by  $T_{\min}$  and  $T_{\max}$ , and at 25°C. Table C-7 in Appendix C lists the liquid

thermal conductivity of chemical compounds, and Figures 3-7a, 3-7b and 3-7c show plots for thermal conductivity of benzene ( $C_6H_6$ ), water ( $H_2O$ ), and iron (Fe) respectively as a function of temperature.

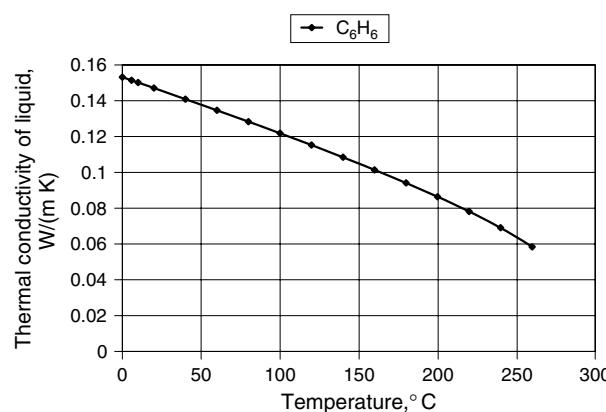


Figure 3-7a Thermal conductivity of benzene ( $C_6H_6$ ).

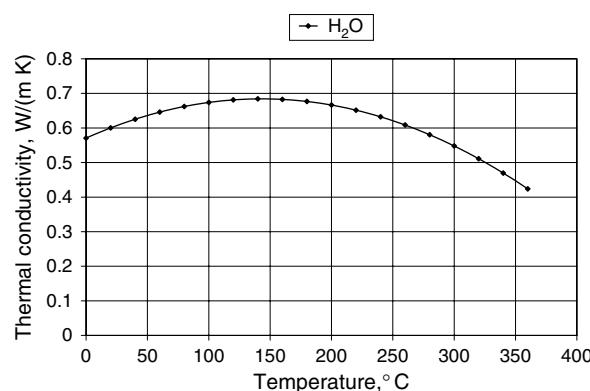


Figure 3-7b Thermal conductivity of water ( $H_2O$ ).

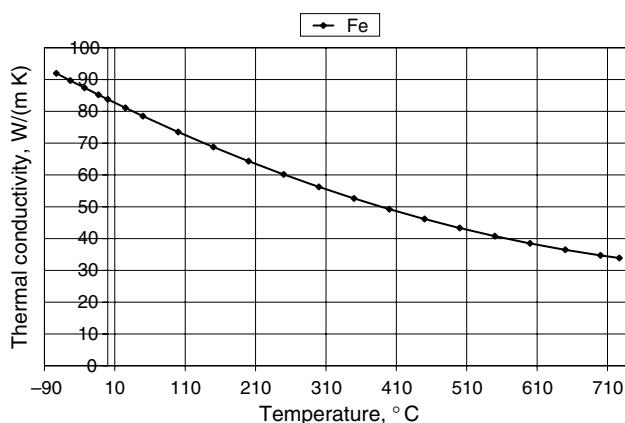


Figure 3-7c Thermal conductivity of iron (Fe).

### EXAMPLE 3-9

Determine the liquid thermal conductivity of benzene ( $C_6H_6$ ) at a temperature of 340 K.

*Solution*

Substituting the correlation coefficients from Table C-7 in Appendix C, and the temperature of 340 K into the correlation equation yields

$$\log_{10} k_{\text{liq}} = (-1.6846) + (1.052) \left( 1 - \frac{340}{562.16} \right)^{2/7}$$

$$k_{\text{liq}} = 10^{-0.87770}$$

$$= 0.1325 \text{ W}/(\text{m K})$$

### EXAMPLE 3-10

Calculate the liquid thermal conductivity of water ( $H_2O$ ) at a temperature of 320 K.

*Solution*

Substituting the correlation coefficients from Table C-7 in Appendix C and the temperature of 320 K in the correlation equation yields

$$k = (-0.2758) + (4.612 \times 10^{-3}) (320)$$

$$+ (-5.5391 \times 10^{-6}) (320^2)$$

$$= 0.6328 \text{ W}/(\text{m K}).$$

### Conversion

The units used for thermal conductivity are  $\text{W}/(\text{m K})$ . Conversion of these to Imperial or cgs units is as follows.

$$\text{W}/(\text{m K}) \times 0.5778 = \text{Btu}/(\text{h ft } ^\circ \text{R})$$

$$\text{W}/(\text{m K}) \times 0.8604 = \text{kcal}/(\text{cm h K})$$

$$\text{W}/(\text{m K}) \times 2.390 \times 10^{-3} = \text{cal}/(\text{cm s K}).$$

or

$$\text{Btu}/(\text{h ft } ^\circ \text{R}) \times 1.731 = \text{W}/(\text{m K})$$

$$\text{kcal}/(\text{cm h K}) \times 1.162 = \text{W}/(\text{m K})$$

$$\text{cal}/(\text{cm s K}) \times 418.4 = \text{W}/(\text{m K}).$$

## 3.8 SURFACE TENSION

As the temperature is raised, the surface tension of a liquid ( $\sigma$ ) in equilibrium with its own vapor decreases and becomes zero at the critical point [1]. In the reduced-temperature range of 0.45–0.65,  $\sigma$  for most organic liquids ranges from 20 to 40 dyn/cm. For low molecular weight dense liquids the  $\sigma$  range is as follows: formaldehyde has  $\sigma > 50$  dyn/cm; water,  $\sigma = 72.8$  dyn/cm at 293 K; and for liquid metals  $\sigma$  is between 300 and 600 dyn/cm. Useful techniques for estimating the surface tension of a liquid are empirical, and critical evaluations have been prepared by Jasper [17], Riddick et al. [18], Timmermans [19], and Vargaftik [20].

Surface tension data of liquids are important in many process design calculations for situations where there is a two-phase interface, for example, two-phase flow in process piping, distillation columns, absorption, and heat exchangers. Surface tension correlation as a function of temperature is based upon a modified Othmer relation, and expressed by [1]

$$\sigma = A \left(1 - \frac{T}{T_c}\right)^n \quad (3-14)$$

where

$\sigma$  = surface tension, dyn/cm  
 $A$ ,  $T_c$ , and  $n$  = regression coefficients for chemical compound  
 $T$  = temperature, K.

The Excel spreadsheet program from the companion website (surface-tension-liquid.xls) calculates the surface tension of compounds between the range of the minimum and the maximum temperatures, which is denoted by  $T_{\min}$  and  $T_{\max}$ . The surface tension has been correlated with other physical parameters such as liquid compressibility, viscosity, molar fractions, and the refractive index. Sastri and Rao [21] developed a linear relationship between the surface tension at normal boiling point ( $\log \sigma_b$ ) and the reduced boiling point ( $T_{br}$ ). Hadden [22] presented a monograph for hydrocarbons that enables rapid calculation for  $\sigma$ . For cryogenic liquids, Sprow and Prausnitz [23] introduced the equation:

$$\sigma = \sigma_o (1 - T_r)^p \quad (3-15)$$

where

$\sigma$  = surface tension, dyn/cm  
 $T_r$  = reduced temperature,  $T/T_c$   
 $\sigma_o$  and  $p$  were determined by a least square analysis of the measured data.

For water, the temperature range for which the surface tension is valid is as follows.

Temperature range 0° C–100° C       $\sigma = 71.97$

Temperature range 100° C–374.2° C       $\sigma = 58.91$ .

Table C-8 in Appendix C lists the surface tension of organic liquids, and Figures 3-8a and 3-8b show plots of surface tension of monochlorobenzene ( $C_6H_5Cl$ ) and water ( $H_2O$ ) respectively as a function of temperature.

### EXAMPLE 3-11

Determine the surface tension of methanol at a temperature of 350 K.

*Solution*

Substituting the correlation coefficients from Table C-8 in Appendix C and the temperature of 350 K into the correlation equation gives

$$\begin{aligned} \sigma &= 68.329 \left(1 - \frac{350}{512.58}\right)^{1.2222} \\ &= 16.79 \text{ dyn/cm} \end{aligned}$$

*Conversion*

$$1 \text{ dyn} = 10^{-5} \text{ N}$$

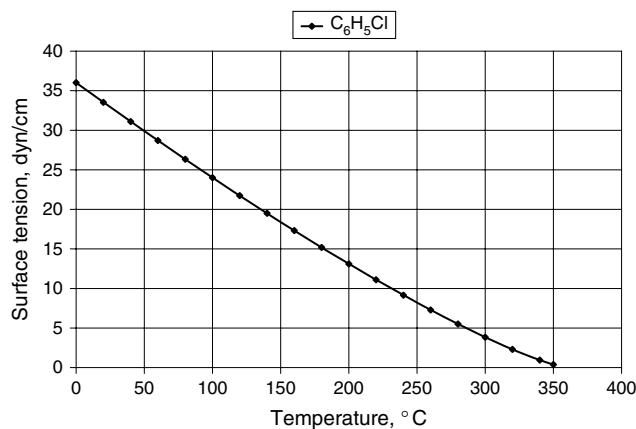


Figure 3-8a Surface tension of monochlorobenzene ( $C_6H_5Cl$ ).

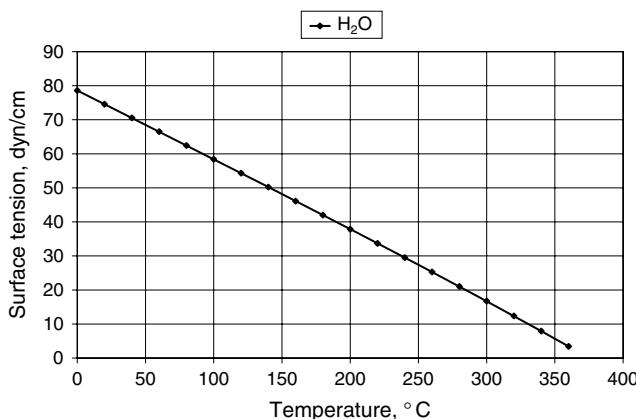


Figure 3-8b Surface tension of water ( $H_2O$ ).

### 3.9 VAPOR PRESSURE

Vapor pressure data,  $P_v$ , of pure components are important in calculations relating to vapor–liquid phase equilibria, for example, in the simplest case to predict the pressure in a closed vessel containing a specific liquid or mixture of liquids. The  $P_v$  data are required for bubble point and most dew point computations. These values are used in flash calculations involving mass-transfer operations. Clearly the design of pressure requirements for storage equipment requires knowledge of the vapor pressure of the components as does the design of appropriate pressure relief systems. The vapor pressure of the saturated liquid as a function of temperature is [2]:

$$\log_{10} P_v = A + \frac{B}{T} + C \log_{10} T + DT + ET^2 \quad (3-16)$$

where

$$\begin{aligned} P_v &= \text{vapor pressure, mm Hg} \\ A, B, C, D, \text{ and } E &= \text{regression coefficients for chemical compound} \\ T &= \text{temperature, K.} \end{aligned}$$

An equation often used to estimate the vapor pressure is the Antoine equation.

$$\ln P_{vp} = A - \frac{B}{T+C} \quad (3-17)$$

The constants  $A$ ,  $B$ , and  $C$  for many compounds are in the data bank of some simulation design packages. Alternatively, these are found in texts such as Reid et al. [6]. Equation (3-17) is only applicable for pressures ranging from 10 to 1500 mm Hg (0.01 to 2 bar). The Antoine equation should never be used outside the stated temperature limits, as extrapolation beyond these limits may lead to incorrect results. The constants  $A$ ,  $B$ , and  $C$  form a set, implying that constants are never used from different tabulations.

Vapor pressures have been measured for many substances, and when reliable measurements are available, they are preferred over results from estimation methods. Boublík et al. [24] have compiled tabulations of experimental data, and Yaws [25] has conducted literature search involving both experimental and parameter values of organic and inorganic compounds. Reid et al. [6] and Beaton and Hewitt [3] have discussed other methods in estimating vapor pressure of compounds but none appears to offer any specific advantage. The Excel spreadsheet program from the companion website (vapor-pressure.xls) calculates the vapor pressure of compounds between the range of the minimum and the maximum temperatures, denoted by  $T_{\min}$  and  $T_{\max}$ . Table C-9 in Appendix C lists the vapor pressure of compounds, and Figures 3-9a and 3-9b show plots of vapor pressure of *n*-pentane ( $C_5H_{12}$ ) and water as a function of temperature respectively.

### EXAMPLE 3-12

Calculate the vapor pressure of methylchloride ( $CH_3Cl$ ) at a temperature of 410 K.

*Solution*

Substituting the regression coefficients from Table C-9 in Appendix C and the temperature of 410 K into the correlation equation gives

$$\begin{aligned} \log_{10} P_v &= 25.7264 + \frac{(-1.7503 \times 10^3)}{410} \\ &\quad + (-6.7151)(\log_{10} 410) + (-1.2956 \times 10^{-9})(410) \\ &\quad + (4.4341 \times 10^{-6})(410^2) \end{aligned}$$

where

$$\begin{aligned} \log_{10} P_v &= 4.6576 \\ P_v &= 10^{4.6576} \\ &= 45,456.9 \text{ mm Hg.} \end{aligned}$$

## 3.10 ENTHALPY OF VAPORIZATION

The enthalpy of vaporization,  $\Delta H_v$ , is also termed the “latent heat of vaporization.” And  $\Delta H_v$  is the difference between the enthalpy of the saturated vapor and that of the saturated liquid at the same

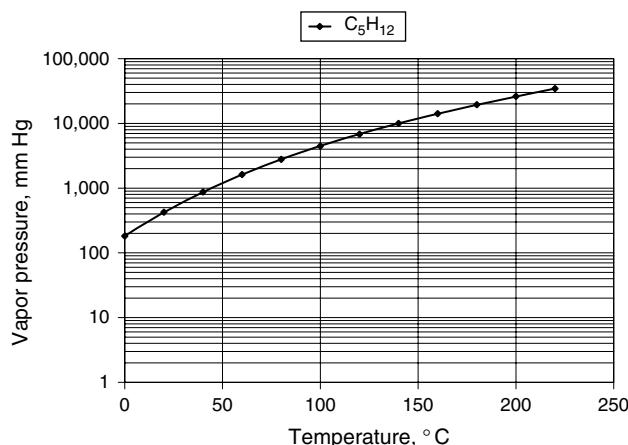


Figure 3-9a Vapor pressure of *n*-pentane ( $C_5H_{12}$ ).

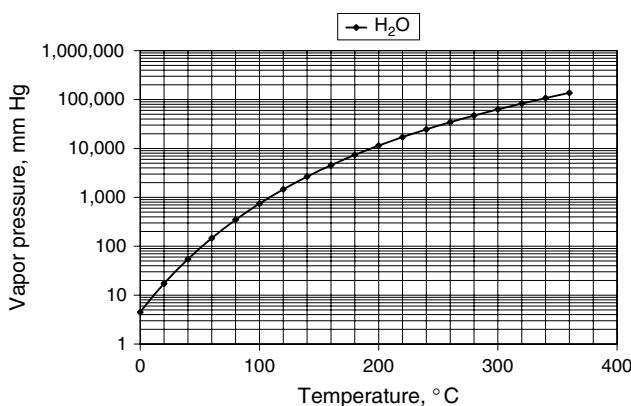


Figure 3-9b Vapor pressure of water.

temperature. The enthalpy of vaporization data is used in process calculations such as the design of relief systems involving volatile compounds. In distillation, heat of vaporization values are needed to find the heat loads for the reboiler and condenser, and knowledge of the enthalpy of vaporization is required in the design of heat exchangers for vaporizing liquids. A modified Watson equation for the enthalpy of vaporization as a function of temperature is as follows.

$$\Delta H_v = A \left( 1 - \frac{T}{T_c} \right)^n \quad (3-18)$$

where

$$\begin{aligned} \Delta H_v &= \text{enthalpy of vaporization, kJ/mol} \\ A, T_c, \text{ and } n &= \text{regression coefficients for chemical compound} \\ T &= \text{temperature, K.} \end{aligned}$$

Many experimental values of  $\Delta H_v$  have shown from calculation that  $\Delta H_v$  is related to the slope of the vapor pressure–temperature curve. Majer and Svoboda [26] have provided a comprehensive and critical compilation of experimental values of  $\Delta H_v$  measured for organic compounds. Tamir et al. [27] have given further compilations of heat of vaporization; however, it is often essential to supplement data with results, calculated or extrapolated by some

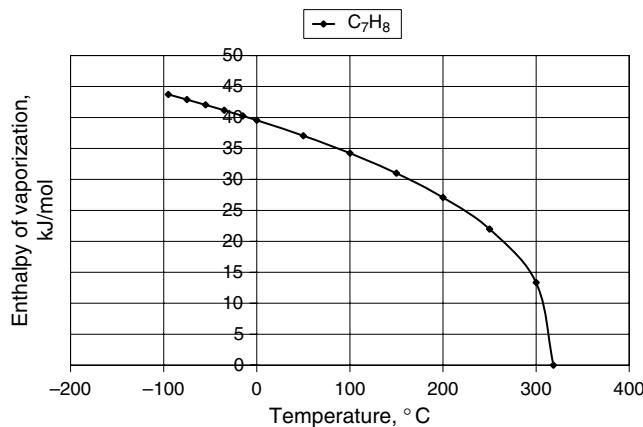


Figure 3-10 Enthalpy of vaporization of toluene ( $C_7H_8$ ).

method. Majer et al. [28] present a comprehensive description of the many methods that have been used to determine  $\Delta H_v$ .

The Excel spreadsheet program from the companion website (enthalpy-of-vaporization.xls) calculates the enthalpy of vaporization of compounds between the range of the minimum and the maximum temperatures, denoted by  $T_{\min}$  and  $T_{\max}$ . Table C-10 in Appendix C lists the enthalpy of vaporization of compounds, and Figure 3-10 shows the enthalpy of vaporization of toluene ( $C_7H_8$ ) as a function of temperature.

#### EXAMPLE 3-13

Calculate the enthalpy of vaporization of toluene ( $C_7H_8$ ) at a temperature of 400 K.

*Solution*

Substituting the regression coefficients from Table C-10 in Appendix C and the temperature at 400 K into the correlation equation yields

$$\Delta H_v = (50.139) \left(1 - \frac{400}{591.79}\right)^{0.383}$$

$$= 32.57 \text{ kJ/mol.}$$

## 3.11 ENTHALPY OF FORMATION

The enthalpy of formation ( $\Delta H_f$ ) is defined as the enthalpy change to form a species from chemical elements in their standard states by an isothermal reaction. In such a reaction scheme, the elements are assumed initially to be at the reaction temperature at 1 atm, and in their most stable phase, for example, diatomic oxygen as an ideal gas and carbon as a solid in the form of  $\beta$ -graphite. Methods of determining  $\Delta H_f$  are provided by Poling et al. [1].

The enthalpy of formation  $\Delta H_f$  for individual compounds in chemical reactions is required to determine the heat of reaction  $\Delta H_r$  and associated heating and cooling requirements:

$$\Delta H_r = \sum (n\Delta H_f)_{\text{products}} - \sum (n\Delta H_f)_{\text{reactants}} \quad (3-19)$$

If  $\Delta H_r < 0$ , then the chemical reaction is exothermic and cooling is needed to maintain the reaction temperature. Alternatively, if  $\Delta H_r > 0$ , the reaction is endothermic and heating is required to conduct the chemical reaction. The correlation of enthalpy of formation for an ideal gas is based on a series expansion temperature and is expressed by [2]:

$$\Delta H_f = A + BT + CT^2 \quad (3-20)$$

where

$\Delta H_f$  = enthalpy of formation of ideal gas, kJ/mol  
 $A$ ,  $B$ , and  $C$  = regression coefficients for chemical compound  
 $T$  = temperature, K.

The enthalpy of formation can also be found from measured heats of combustion,  $\Delta H_c$ , or from an experimental enthalpy of reaction,  $\Delta H_r$ . Domalski and Herring [29] reviewed values of  $\Delta H_{f,298\text{K}}$  and  $\Delta H_{c,298\text{K}}$  for several hundreds of organic compounds. For organic compounds, the range of application for the respective minimum and maximum temperatures are denoted by  $T_{\min}$  and  $T_{\max}$  from Table C-11 in Appendix C. The Excel spreadsheet program from the companion website (enthalpy-formation.xls) calculates the enthalpy of formation of organic compounds. Figure 3-11 shows the plot for the enthalpy of formation of ethane ( $C_2H_6$ ) as a function of temperature.

#### EXAMPLE 3-14

Determine the heat of formation of ethane ( $C_2H_6$ ) as an ideal gas at 500 K.

*Solution*

Substituting the correlation coefficients from Table C-11 in Appendix C and the temperature of 500 K into the correlation equation yields

$$\begin{aligned} \Delta H_f &= (-66.735) + (-6.9337 \times 10^{-2})(500) \\ &\quad + (3.0379 \times 10^{-5})(500)^2 \\ &= -93.81 \text{ kJ/mol.} \end{aligned}$$

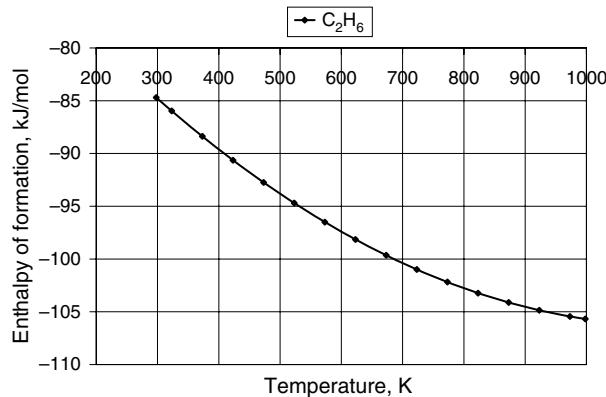


Figure 3-11 Enthalpy of formation of ethane ( $C_2H_6$ ).

**EXAMPLE 3-15**

Calculate the heat of the reaction for the dehydrogenation of ethane to ethylene at a reaction temperature 860 K.

*Solution*

The heat of the reaction may be determined from enthalpy of formation at 860 K for the products and reactants. Using regression coefficients for ethane and ethylene from Table C-11 in Appendix C, and the equation for enthalpy of formation, we obtain

$$\begin{aligned}\Delta H_{f,\text{C}_2\text{H}_6} &= (-66.735) + (-6.9337 \times 10^{-2})(860) \\ &\quad + (3.0379 \times 10^{-5})(860)^2 \\ &= -103.897 \text{ kJ/mol} \\ \Delta H_{f,\text{H}_2} &= 0 \\ \Delta H_{f,\text{C}_2\text{H}_4} &= 63.053 + (-4.1076 \times 10^{-2})(860) \\ &\quad + (1.6598 \times 10^{-5})(860)^2 \\ &= 40.0 \text{ kJ/mol.}\end{aligned}$$

Substituting  $\Delta H_f$  values at 860 K into the equation for the heat of the reaction yields

$$\begin{aligned}\Delta H_{\text{reaction}} &= 40 + 0 - (-103.897) \\ &= 143.89 \text{ kJ/mol.}\end{aligned}$$

Since  $\Delta H_{\text{reaction}} > 0$ , the reaction is endothermic, and would require heating to maintain the reaction temperature.

**3.12 GIBBS ENERGY OF FORMATION**

Gibbs energy of formation is important in the analysis of chemical reactions. Values for individual compounds (reactants and products) are required to determine the change in Gibbs energy for the reaction. This change is significant because of the associated chemical equilibrium for the reaction. The following criteria are generally true for the feasibility of chemical reaction. If the change in Gibbs energy is negative, the thermodynamics for the reaction are favorable. However, if the change in Gibbs energy is highly positive, the thermodynamics for the reaction are not favorable. The chemical equilibrium for a reaction is associated with the change in Gibbs free energy ( $\Delta G_r$ ) for the reaction:

$$\Delta G_r = \sum (n\Delta G_f)_{\text{products}} - \sum (n\Delta G_f)_{\text{reactants}} \quad (3-21)$$

The changes in Gibbs energy for a reaction may be used in preliminary work to determine if a reaction is thermodynamically favorable at a given temperature. For thermodynamic equilibrium,

the following rough criteria are useful for quick screening of chemical reactions:

$\Delta G_r < 0 \text{ kJ/mol}$	(reaction favorable)
$0 < \Delta G_r < 50 \text{ kJ/mol}$	(reaction possibly favorable)
$\Delta G_r > 50 \text{ kJ/mol}$	(reaction not favorable).

The correlation for Gibbs energy of formation is

$$\Delta G_f = A + BT + CT^2 \quad (3-22)$$

where

$$\begin{aligned}\Delta G_f &= \text{Gibbs energy of formation of ideal gas,} \\ &\quad \text{kJ/mol} \\ A, B, \text{ and } C &= \text{regression coefficients for chemical} \\ &\quad \text{compounds} \\ T &= \text{temperature, K.}\end{aligned}$$

The Excel spreadsheet program from the companion website (Gibbs-Energy-of-Formation.xls) calculates Gibbs energy of formation for a range of temperatures between minimum and maximum, as denoted by  $T_{\min}$  and  $T_{\max}$  in Table C-12 in Appendix C. Figure 3-12 shows the plot for Gibbs free energy of formation of ethanol ( $\text{C}_2\text{H}_5\text{OH}$ ).

**EXAMPLE 3-16**

Calculate the Gibbs free energy of formation of ethylene ( $\text{C}_2\text{H}_4$ ) as a low pressure gas at 500 K.

*Solution*

Correlation constants ( $A$ ,  $B$ ,  $C$ ) for ethylene from Table C-12 in Appendix C are substituted into the equation at a temperature of 500 K.

$$\begin{aligned}\Delta G_f^\circ &= 51.752 + (4.9338 \times 10^{-2})(500) \\ &\quad + (1.7284 \times 10^{-5})(500)^2 \\ &= 80.74 \text{ kJ/mol.}\end{aligned}$$

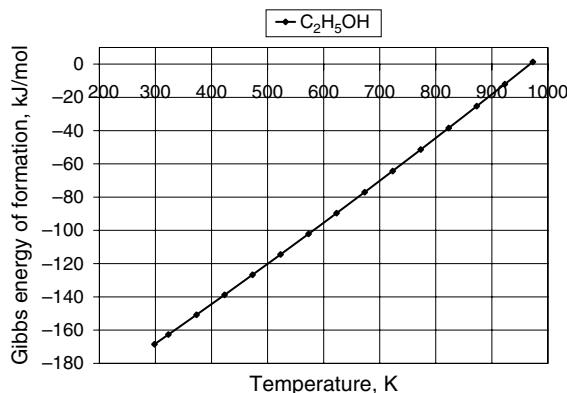
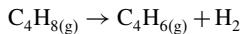


Figure 3-12 Gibbs free energy of formation of ethanol ( $\text{C}_2\text{H}_5\text{OH}$ ).

**EXAMPLE 3-17**

1,3-Butadiene is prepared by the gas phase catalytic dehydrogenation of 1-butene



Calculate the standard state Gibbs free energy of reaction at 298 K

*Solution*

The change in Gibbs free energy of reaction may be determined from Gibbs free energy of formation for the products and reactants.

$$\Delta G_{\text{reaction}} = \Delta G_{\text{f,C}_4\text{H}_6} + \Delta G_{\text{f,H}_2} - \Delta G_{\text{f,C}_4\text{H}_8}$$

Using correlation constants (*A*, *B*, and *C*) from Table C-12 in Appendix C at a temperature of 298.15 K, we obtain

$$\begin{aligned}\Delta G_{298.15,\text{C}_4\text{H}_6}^{\circ} &= 109.172 + (0.13296)(298.15) \\ &\quad + (1.9003 \times 10^{-5})(298.15)^2 \\ &= 150.50 \text{ kJ/mol.}\end{aligned}$$

$$\Delta G_{298.15,\text{H}_2}^{\circ} = 0$$

$$\begin{aligned}\Delta G_{298.15,\text{C}_4\text{H}_8}^{\circ} &= (-1.692) + (0.23442)(298.15) \\ &\quad + (3.1582 \times 10^{-5})(298.15)^2 \\ &= 71.007 \text{ kJ/mol.}\end{aligned}$$

Substituting  $\Delta G_f$  values into the equation for Gibbs free energy gives

$$\begin{aligned}\Delta G_{\text{reaction}} &= \Delta G_{\text{f,C}_4\text{H}_6} + \Delta G_{\text{f,H}_2} - \Delta G_{\text{f,C}_4\text{H}_8} \\ \Delta G_{\text{reaction}} &= 150.5 + 0 - 71.007 \\ &= 79.49 \text{ kJ/mol.}\end{aligned}$$

Since the change in Gibbs free energy for the reaction is  $\Delta G_f > 50 \text{ kJ/mol}$ , the thermodynamics for the reaction are unfavorable.

**EXAMPLE 3-18**

Calculate the Gibbs free energy for the gas phase catalytic dehydrogenation of butane to butadiene at 900 K and 1 bar.

*Solution*

Using correlation constants (*A*, *B*, and *C*) from Table C-12 in Appendix C at a temperature of 900 K yields

$$\begin{aligned}\Delta G_{298.15,\text{C}_4\text{H}_6}^{\circ} &= 109.172 + (0.13296)(900) \\ &\quad + (1.9003 \times 10^{-5})(900)^2 \\ &= 244.228 \text{ kJ/mol} \\ \Delta G_{298.15,\text{H}_2}^{\circ} &= 0\end{aligned}$$

$$\begin{aligned}\Delta G_{298.15,\text{C}_4\text{H}_8}^{\circ} &= (-1.692) + (0.23442)(900) \\ &\quad + (3.1582 \times 10^{-5})(900)^2 \\ &= 234.867 \text{ kJ/mol.}\end{aligned}$$

Substituting  $\Delta G_f$  values into the equation for Gibbs free energy gives

$$\begin{aligned}\Delta G_{\text{reaction}} &= 244.228 + 0 - 234.867 \\ &= 9.361 \text{ kJ/mol.}\end{aligned}$$

The result shows that by increasing the temperature to 900 K,  $\Delta G_f < 50 \text{ kJ/mol}$ , and the thermodynamics for the reaction are favorable.

### 3.13 SOLUBILITY IN WATER CONTAINING SALT

The solubility of hydrocarbons in water containing salt is intended for initial engineering and environmental applications, such as the distribution of a hydrocarbon spill upon its contact with sea water. The correlation is based on a series expansion in salt concentration, and is expressed by [2]

$$\log_{10} S = A + BX + CX^2 \quad (3-23)$$

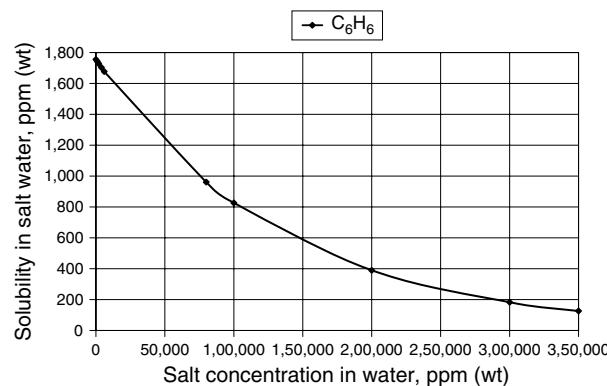
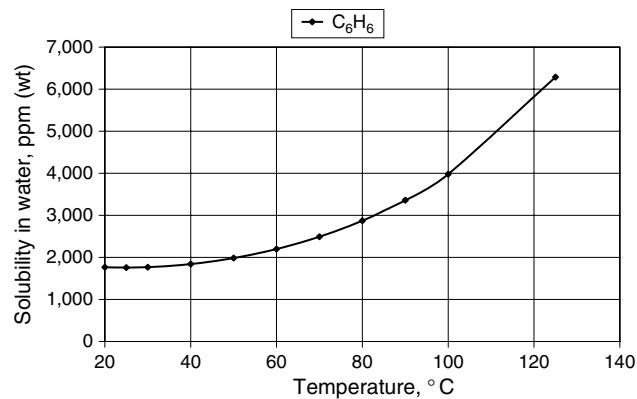
where

$S$  = solubility in salt water at 25°C, parts per million by weight, ppm (wt)

$X$  = concentration of salt (NaCl) in water, parts per million by weight, ppm (wt)

*A*, *B*, and *C* = correlation constants, and determined from regression data of water solubility.

The correlation constants from Table C-13 in Appendix C were determined from regression of the data from sources for water solubility. The tabulated values for solubility of hydrocarbons in water apply to conditions of saturation in which the hydrocarbon is in equilibrium with water. For saturation, the system pressure is approximately equal to the sum of the vapor pressures of hydrocarbon and water. The correlation is applicable to a wide variety of hydrocarbons (e.g., alkanes, naphthenes, and aromatics with no single and multiple substitutions). The Excel spreadsheet program from the companion website (solubility-in-salt-water.xls) calculates the solubility in salt water with the range of salt concentration between 0 and 35,000 ppm (wt). Figure 3-13 shows the solubility of benzene in salt water.

Figure 3-13 Solubility of benzene ( $C_6H_6$ ) in salt water.Figure 3-14 Solubility of benzene ( $C_6H_6$ ) in water.**EXAMPLE 3-19**

Calculate the solubility of pentane ( $C_5H_{12}$ ) in salt water with concentration of salt ( $NaCl$ ) in water being 100,000 ppm (wt).

*Solution*

Correlation constants ( $A$ ,  $B$ , and  $C$ ) for pentane from Table C-13 in Appendix C are substituted into the equation at salt concentration 100,000 ppm (wt):

$$\begin{aligned} \log_{10} S &= 1.5966 + (-4.5956 \times 10^{-6})(100,000) \\ &\quad + (2.2978 \times 10^{-12})(100,000)^2 \\ S &= 10^{1.160} \\ &= 14.5 \end{aligned}$$

### 3.14 SOLUBILITY IN WATER AS A FUNCTION OF TEMPERATURE

The water solubility of hydrocarbons is applicable in air- and steam-stripping operations, within the temperature range of 25–121°C. The correlation of water solubility of hydrocarbons is expressed by [2]:

$$\log_{10} S = A + \frac{B}{T} + \frac{C}{T^2} \quad (3-24)$$

where

$$\begin{aligned} S &= \text{solubility in water, ppm (wt)} \\ T &= \text{temperature, K} \end{aligned}$$

$A$ ,  $B$ , and  $C$  = correlation constants.

The Excel spreadsheet program from the companion website (solubility-in-water.xls) calculates the water solubility of hydrocarbons as a function of temperature. Table C-14 in Appendix C lists water solubility of organic compounds, and Figure 3-14 shows the plot of solubility of benzene in water as a function of temperature.

**EXAMPLE 3-20**

Determine the solubility of hexane ( $C_6H_{14}$ ) in water at a temperature of 370 K.

*Solution*

Substituting the correlation constants ( $A$ ,  $B$ , and  $C$ ) from Table C-14 in Appendix C for hexane into the equation at a temperature of 370 K gives

$$\begin{aligned} \log_{10} S &= 10.992 + \frac{(-5969.484)}{370} + \left( \frac{8.89453 \times 10^5}{370^2} \right) \\ &= 1.3553 \\ S &= 10^{1.3553} \\ &= 22.66 \text{ ppm (wt).} \end{aligned}$$

### 3.15 HENRY'S LAW CONSTANT FOR GASES IN WATER

Henry's law constant and the solubility of gases in water are important parameters that affect health, safety, and environmental considerations in the chemical process industries. Very low concentrations of gases in water can provide concentrations in air at the air–water interface that exceed the threshold limit value (TLV) for human exposure, and the lower flammability limit (LFL) for flammability [30].

#### HENRY'S LAW CONSTANT EQUATION

The correlation for Henry's law constant for gases in water as a function of temperature can be expressed by [31]

$$\log_{10} H = A + \frac{B}{T} + C \log_{10} T + DT \quad (3-25)$$

where

$$\begin{aligned} H &= \text{Henry's law constant at pressure of 1 atm} \\ &\quad (101.325 \text{ kPa), atm/mol fraction} \\ A, B, C, \text{ and } D &= \text{regression coefficients for gas} \\ T &= \text{temperature, K.} \end{aligned}$$

The Excel spreadsheet program from the companion website (Henry's-law-constant.xls) calculates the Henry's law constant for the respective minimum and maximum temperatures denoted by  $T_{\min}$  and  $T_{\max}$ . Table C-15 in Appendix C lists Henry's Law Constant for gases in water, and Figure 3-15 shows the variation of Henry's law constant with temperature of propane, ethylene, methane, and propylene. These plots demonstrate that Henry's law constant increases with temperature.

**EXAMPLE 3-21**

Calculate the Henry's Law constant of propylene ( $C_3H_6$ ) in water at a temperature of 320 K

*Solution*

Substituting the correlation constants ( $A$ ,  $B$ ,  $C$ , and  $D$ ) for propylene from Table C-15 in Appendix C into the equation at a temperature of 320 K gives

$$\begin{aligned}\log H &= (-2570.0227) + \frac{(6.86740 \times 10^4)}{320} \\ &\quad + (1.03384 \times 10^3) (\log 320) \\ &\quad + (-0.719694) (320)\end{aligned}$$

$$\begin{aligned}\log_{10} H &= 4.2057 \\ H &= 10^{4.2057} \\ &= 16058.3 \text{ atm/mol fraction.}\end{aligned}$$

**EXAMPLE 3-22**

Propylene ( $C_3H_6$ ) is in contact with water at 25°C (298.15 K) and 1 atm in an industrial process. The concentration of propylene in the liquid at the surface of the water is 15 ppm on a mol basis. Determine the concentration of propylene in the air at the surface of the water.

*Solution*

From the thermodynamics at low pressure, the partition coefficient  $K_i$  is given by

$$K_i = H_i / P_t$$

where

$P_t$  = total pressure, atm

$H_i$  = Henry's law constant, atm/mol fraction at pressure of 1 atm.

Substituting the coefficients ( $A$ ,  $B$ ,  $C$ , and  $D$ ) from Table C-15 in Appendix C and at a temperature of 298.15 K into the correlation equation gives

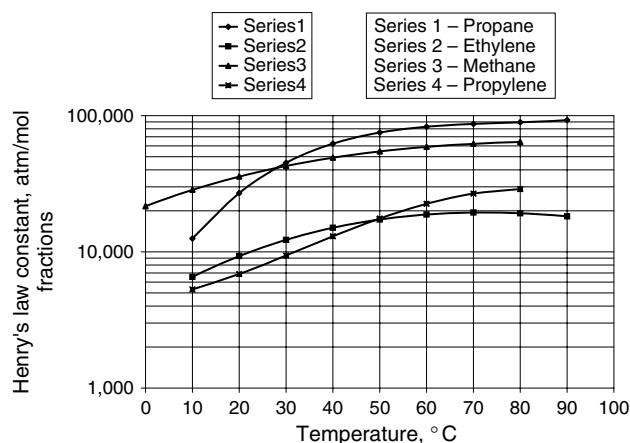


Figure 3-15 Henry's law constant of gases as a function of temperature.

### 3.16 SOLUBILITY OF GASES IN WATER

The correlation for the solubility of gases in water as a function of temperature can be expressed by [31]

$$\log_{10} xwt = A + \frac{B}{T} + C \log T + DT \quad (3-26)$$

where

$xwt$  = solubility in water at pressure of 1 atm (101.325 kPa), weight fraction

$A$ ,  $B$ ,  $C$ , and  $D$  = regression coefficients for gas  
 $T$  = temperature, K.

$$\log_{10} H = (-2570.0227) + \frac{(6.8674 \times 10^4)}{298.125}$$

$$\begin{aligned}&\quad + (1.03384 \times 10^3) (\log 298.15) \\ &\quad + (-0.71964) (298.15)\end{aligned}$$

$$H = 10^{3.903946}$$

$$= 8015.783 \text{ atm/mol fraction.}$$

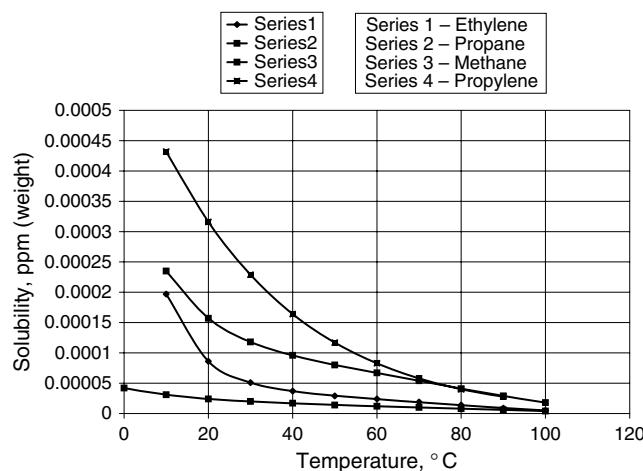
$$\text{The partition coefficient } K_i = \frac{8015.783}{1.0} = 8015.783$$

Since the vapor concentration is given by

$$y_i = K_i x_i,$$

substituting the value of  $K_i$  and the liquid concentration ( $x_i = 15 \text{ ppm}$ ) yields

$$\begin{aligned}y_i &= (8015.783) (15) \\ &= 120236.76 \text{ ppm(mol)} \\ &= 12.02 \%\end{aligned}$$



**Figure 3-16** Solubilities of selected gases in water as a function of temperature.

The Excel spreadsheet program from the companion website (*Henry's-law-constant.xls*) calculates the solubility of gases in water between the range of the minimum and the maximum temperatures, denoted by  $T_{\min}$  and  $T_{\max}$  in Table C-16 in Appendix C. Figure 3-16 shows the variation of water solubility with temperature for ethylene, propane, methane, and propylene. These plots show that solubility decreases with temperature.

### EXAMPLE 3-23

Calculate the concentration of propylene ( $C_3H_6$ ) in water at a temperature of 360 K and 1 atm.

*Solution*

Substituting the coefficients ( $A$ ,  $B$ ,  $C$ , and  $D$ ) from Table C-16 in Appendix C, and at a temperature at 360 K into the correlation equation yields

$$\begin{aligned} \log_{10} xwt &= (-39.4402) + \frac{(6.34551 \times 10^2)}{360} \\ &\quad + (1.7542 \times 10 \times \log 360) \\ &\quad + (-3.24155 \times 10^{-2}) (360) \\ xwt &= 10^{-4.504479} \\ &= 31.29 \text{ ppm (wt)}. \end{aligned}$$

### 3.17 SOLUBILITY AND HENRY'S LAW CONSTANT FOR SULFUR COMPOUNDS IN WATER

Sulfur emissions are important safety, health, and environmental concerns in the chemical process industries. As part of a comprehensive national program to reduce emissions from non-road diesel engines, the U.S. Environmental Protection Agency (EPA) has embarked on a program to further reduce the currently allowable level of sulfur in non-road diesel fuel by more than 99%. At low levels, sulfur can yield concentrations in air at the air–water interface that either exceed the TLV for human exposure, the lower explosion limit (LEL) for flammability, or both. A correlation of

water solubility of sulfur compounds based on the boiling point is expressed by [31]

$$\log_{10} S = A + BT_B + CT_B^2 + DT_B^3 \quad (3-27)$$

where

$S$  = solubility in water at 25°C, ppm (wt)

$T_B$  = boiling point of compound, K.

Estimation of solubility of hydrocarbons and organic oxygen compounds is dependent on the following regression coefficients.

$A = -6.9000$  (mercaptans)

$A = -6.8500$  (thiophenes)

$A = -6.5389$  (sulfides)

$B = +1.005 \times 10^{-1}$

$C = -2.7288 \times 10^{-4}$

$D = +1.9987 \times 10^{-7}$ .

The correlation constants  $A$ ,  $B$ ,  $C$ , and  $D$  apply to mercaptans, thiophenes, and sulfides in water, with the range for boiling point between 310 and 600 K. The regression coefficients for estimating the solubility of hydrocarbons and organic oxygen compounds are listed in Table C-17 in Appendix C. The Excel spreadsheet program from the companion website (*solubility-of-sulfur-compds-in-water.xls*) calculates the solubility of sulfur compounds in water. Figures 3-17a and 3-17b show plots of solubility of sulfur and aromatic compounds as a function of boiling point.

### EXAMPLE 3-24

Determine the solubility of ethyl mercaptans ( $C_4H_6S$ ) in water at a saturation temperature of 308.15 K.

*Solution*

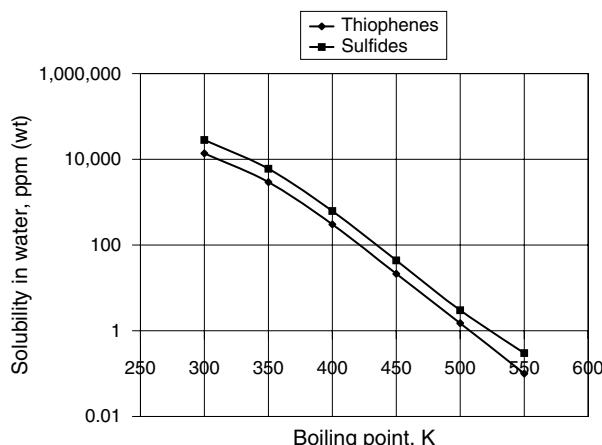
Substituting the regression coefficients ( $A$ ,  $B$ ,  $C$ , and  $D$ ) mentioned above and the boiling point of 308.15 K into the correlation equation yields

$$\begin{aligned} \log_{10} S &= (-6.9) + (0.1005 \times 308.15) \\ &\quad + (-2.7288 \times 10^{-4} \times 308.15^2) \\ &\quad + (1.9987 \times 10^{-7} \times 308.15^3) \\ S &= 10^{4.0057} \\ &= 10132 \text{ ppm (wt)}. \end{aligned}$$

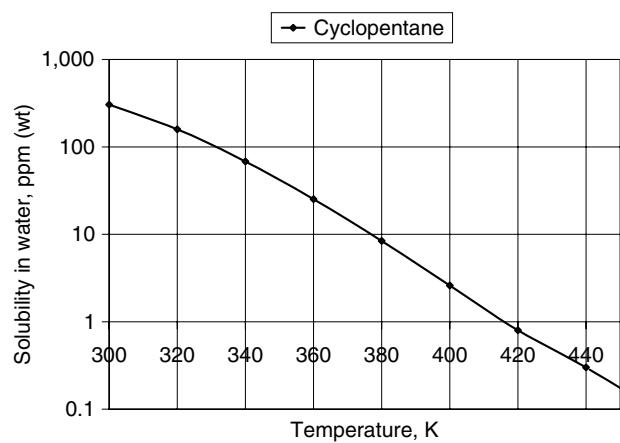
### 3.18 SOLUBILITY OF NAPHTHENES IN WATER

Naphthenes (e.g., cycloalkanes) are a major component of unrefined fuels. Like other hydrocarbons such as paraffins and aromatics, solubility in water is essential in view of the environmental hazard they present when spilled in water. The correlation of naphthenes' water solubility as a function of temperature can be expressed by Eq. 3-24 [31].

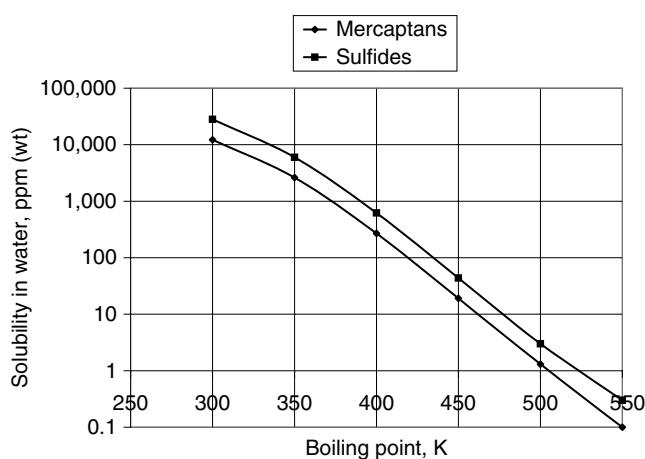
Table C-18 in Appendix C lists the correlation constants and solubilities of naphthenes at 25 and 100°C. The Excel spreadsheet program from the companion website (*solubility-of-naphthene-compds-in-water.xls*) calculates the solubility of



**Figure 3-17a** Solubility of sulfur compounds in water as a function of boiling point for thiophenes and sulfides.



**Figure 3-18a** Solubility of cyclopentane in water as a function of temperature.



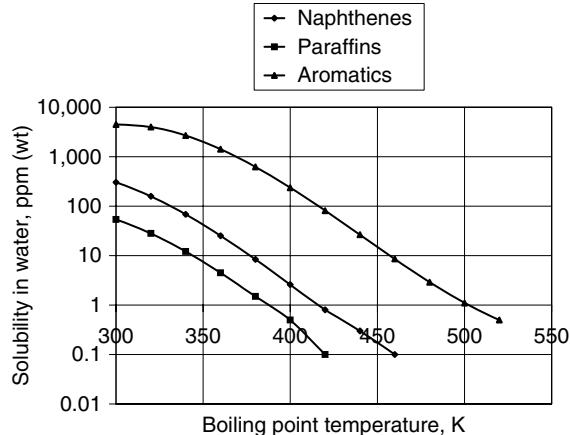
**Figure 3-17b** Solubility of sulfur compounds in water as a function of boiling point for sulfides and mercaptans.

naphthalene compounds in water as a function of temperature. Figure 3.18a shows the solubility of cyclopentane in water as a function of temperature.

### ESTIMATING EQUATION

Yaws et al. [31] developed an estimation of water solubility for naphthalenes with C<sub>5</sub> and higher carbon numbers, and this is represented by Eq. 3-27.

Table 3-1 shows the correlation constants for estimating the solubility of hydrocarbons and organic oxygen compounds in water. The correlation is applicable to substances that are liquids at ambient conditions (25°C, 1 atm). For hydrocarbons (e.g., alkanes, cycloalkanes and benzenes), the range of boiling points is 298–561 K. For organic oxygen compounds (alcohols, ketones, ethers, and aldehydes), the range for boiling points is 298–625 K. The correlation is not applicable to naphthalenes that are solids at ambient conditions, since a different solubility curve is obtained for solids. Figure 3-18b shows the solubility profiles of aromatics, naphthalenes, and alkanes as a function of boiling point.



**Figure 3-18b** Solubility of aromatics, naphthalenes, and alkanes as a function of temperature.

### EXAMPLE 3-25

A chemical spill of cyclopentane (C<sub>5</sub>H<sub>10</sub>) occurs into a body of water at ambient condition (25°C). Determine the concentration of cyclopentane in the air at the surface of the water.

*Solution*

Substituting the correlation constants A, B, and C from Table C-18 in Appendix C into Eq. 3-24, at temperature 25°C (298.15 K), gives

$$\log_{10} S = 8.665 + \frac{(-3850.808)}{298.15} + \frac{(5737.704 \times 10^2)}{298.15^2}$$

$$\log_{10} S = 2.2039$$

$$S = 10^{2.2039}$$

$$= 159.92 \text{ ppm (wt)}$$

**TABLE 3-1 Correlation Constants for Estimating the Solubility of Organic Compounds in Water**

Cyclopentane (naphthenes)				Cyclohexane (naphthenes)		
Correlation Constants	25°C	99.1°C	120°C	25°C	99.1°C	120°C
A	-16.9	-16.567	-0.033	-16.7	-16.290	0.085
B	$177.881 \times 10^{-3}$	$177.881 \times 10^{-3}$	$411.139 \times 10^{-4}$	$177.881 \times 10^{-3}$	$177.881 \times 10^{-3}$	$411.139 \times 10^{-4}$
C	$-500.907 \times 10^{-6}$	$-500.907 \times 10^{-6}$	$-136.98 \times 10^{-6}$	$-0.0907 \times 10^{-6}$	$-500.907 \times 10^{-6}$	$-136.98 \times 10^{-6}$
D	$411.124 \times 10^{-9}$	$411.124 \times 10^{-9}$	$170.019 \times 10^{-9}$	$411.124 \times 10^{-9}$	$411.124 \times 10^{-9}$	$170.019 \times 10^{-9}$

Correlation Constants	Benzene (Aromatics)	Alkanes (Paraffins)	Alcohols	Aldehydes	Ketones	Ethers
A	-24.008 (no and single substitutions) -23.650 (double and triple substitutions)	-17.652 (normal and isomers)	45.6398 (normal and isomers)	20.4898 (normal and isomers)	45.233 (normal and isomers)	7.510 (normal and isomers)
B	$221.196 \times 10^{-3}$	$177.811 \times 10^{-3}$	$-2.3859 \times 10^{-1}$	$-9.031 \times 10^{-2}$	$-2.3859 \times 10^{-1}$	$3.2057 \times 10^{-3}$
C	$-555.632 \times 10^{-6}$	$-500.907 \times 10^{-6}$	$4.8739 \times 10^{-4}$	$1.9223 \times 10^{-4}$	$4.8739 \times 10^{-4}$	$-4.0887 \times 10^{-5}$
D	$418.803 \times 10^{-9}$	$411.124 \times 10^{-9}$	$-3.716 \times 10^{-7}$	$-1.7856 \times 10^{-7}$	$-3.716 \times 10^{-7}$	$4.7284 \times 10^{-9}$

**EXAMPLE 3-26**

A chemical spill of cyclopentane ( $C_5H_{10}$ ) occurs into a body of water at ambient condition (25°C). Calculate the concentration of cyclopentane in the water at saturation.

*Solution*

Substituting the correlation constants  $A$ ,  $B$ ,  $C$ , and  $D$  from Table 3-1 and boiling point of cyclopentane ( $T_B = 322.40\text{ K}$ ) into the correlation Eq. 3-27 gives

$$\begin{aligned}\log_{10} S &= (-16.9) + (177.881 \times 10^{-3}) (322.4) \\ &\quad + (-500.907 \times 10^{-6}) (322.4)^2 \\ &\quad + (411.124 \times 10^{-9}) (322.4)^3 \\ S &= 10^{2.16078} \\ &= 144.8 \text{ ppm (wt)}.\end{aligned}$$

### 3.19 SOLUBILITY AND HENRY'S LAW CONSTANT FOR NITROGEN COMPOUNDS IN WATER

The solubility of nitrogen compounds (amines) in water is essential for health, safety, and environmental considerations. The U.S. Occupational Safety and Health Act (OSHA) has set the TLV for nitrogen compounds. For example, the TLV for triethylamine (TEA) in air is 25 ppm by volume. A concentration of 0.00001 mol fraction in water is in equilibrium at about 111.6 ppm at the air–water interface [32]. Breathing will be uncomfortable and dangerous if the TVL is greatly exceeded. Concerning safety, the LEL for TEA in air is 1.2% [33]. Therefore, a

concentration of 0.002 mol fraction of TEA in water is in equilibrium with a 2.2% concentration in air at the air–water interface. If the LEL is exceeded, it could result in an explosion. Correlation of the solubility of nitrogen compounds in water as a function of boiling point of the compound can be represented by

$$\log(\gamma^\infty) = A + BT_B \quad (3-28)$$

where

$\gamma^\infty$  = activity coefficient at infinite dilution in water

$T_B$  = boiling point of compound, K

$A = -7.07$  (primary amines)

$A = -6.88$  (secondary amines)

$A = -6.34$  (tertiary amines)

$B = 0.0233$ .

Henry's law constant is essential for engineering and environmental studies involving nitrogen compounds in water. From thermodynamics at low pressure, the partition coefficient  $K_i$  is given by

$$K_i = \frac{H_i}{P_t} \quad (3-29)$$

where

$H_i$  = Henry's law constant

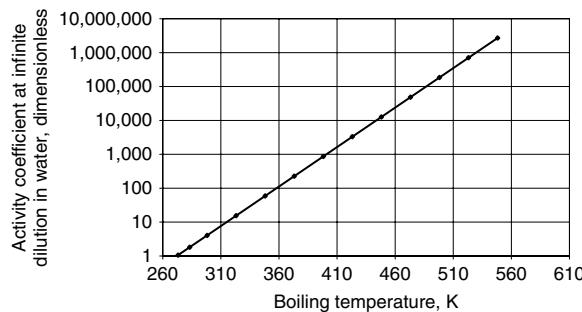
$P_t$  = Total pressure.

The vapor concentration  $y_i$  is determined by

$$y_i = K_i x_i \quad (3-30)$$

Substituting Eq. 3-29 into Eq. 3-30 gives:

$$y_i = \left( \frac{H_i}{P_t} \right) x_i \quad (3-31)$$



**Figure 3-19** Correlation of activity coefficient for tertiary amines as a function of boiling point temperature.

The coefficients  $A$  and  $B$  were determined from regression of the available data, and an extensive literature search was conducted to identify the data sources.

The correlation constants  $A$  and  $B$  apply to primary, secondary, and tertiary amines. Tables C-19 and C-20 in Appendix C list the solubility of nitrogen compounds in water and the Henry's law constant for nitrogen compounds in water at ambient condition respectively. The Excel spreadsheet program from the companion website (solubility-of-nitrogen-compds-in-water.xls) calculates the solubility of nitrogen compounds in water as a function of boiling point. Figure 3-19 shows the correlation of activity coefficient for tertiary amines as a function of boiling point.

#### EXAMPLE 3-27

Determine the concentration of triethylamine ( $C_6H_{15}N$ ) in water at a saturation temperature of 361.92 K.

*Solution*

The constants  $A$  and  $B$  and the boiling point are as follows.

$$\begin{aligned}A &= -6.34 \\B &= 0.0233 \\T_B &= 361.92 \text{ K.}\end{aligned}$$

Substituting the constants at the boiling point of 361.92 K into Eq. 3.28 gives

$$\begin{aligned}\log(\gamma^\infty) &= (-6.34) + (0.0233)(361.92) \\ \log_{10} \gamma^\infty &= 2.0927 \\ \gamma^\infty &= 123.80\end{aligned}$$

The solubility is the reciprocal of  $(S = \frac{1}{\gamma^\infty})$  of the activity coefficient  $\gamma^\infty$ .

That is,

$$\begin{aligned}S &= 1/123.8 \\&= 0.0080775 \text{ mol} \\&= 8077.5 \text{ ppm (mol).}\end{aligned}$$

#### EXAMPLE 3-28

A chemical spill of triethylamine ( $C_6H_{15}N$ ) occurs into a body of water at ambient condition. The concentration in the liquid at the surface of the water is 0.0015 mol fraction ( $x_i$ ). Calculate the concentration of triethylamine in the air at the surface of the water.

*Solution*

From thermodynamics at low pressure, the partition coefficient is given by Eq. 3-29

$$K_i = \frac{H_i}{P_t}$$

Substituting Henry's law constant from Table C-20 in Appendix C ( $H_i = 11.16 \text{ atm/mol fraction}$  for triethylamine), and the total pressure ( $P_t = 1 \text{ atm}$ ) into Eq. 3-29 gives

$$K_i = \frac{11.16}{1}$$

The vapor concentration  $y_i$  is

$$\begin{aligned}y_i &= \left( \frac{H_i}{P_t} \right) x_i \\y_i &= \left( \frac{11.16}{1} \right) (0.0015) \\&= 0.0167 \\&= 1.67\%(\text{mol}).\end{aligned}$$

## 3.20 COEFFICIENT OF THERMAL EXPANSION OF LIQUID

The thermal expansion coefficient of liquids for organic and inorganic compounds is required in the design of relief systems for process equipment. Liquids contained in process equipment will expand with an increase in temperature. In order to accommodate such expansion, relief systems are normally designed to relieve thermally expanding liquid, and to prevent pressure build-up from the expansion. Otherwise, this shows an increase in pressure from the thermally expanding liquid, and an eventual damage to the process equipment may occur if the pressure increase is excessive.

The correlation of thermal expansion coefficient of liquid as a function of temperature is expressed by [2]

$$B_{\text{liq}} = a \left( 1 - \frac{T}{T_c} \right)^m \quad (3-32)$$

where

$B_{\text{liq}}$  = thermal expansion coefficient of liquid,  $1/\text{ }^\circ\text{C}$

$a$  and  $m$  = regression coefficients for chemical compound

$T$  = temperature, K

$T_c$  = critical temperature, K.

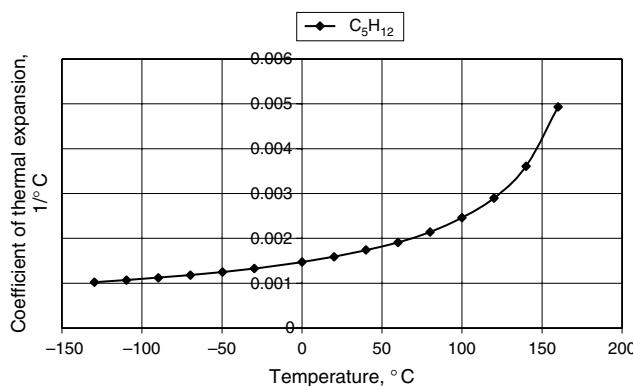


Figure 3-20 Coefficient of thermal expansion of n-pentane ( $C_5H_{12}$ ).

The Excel spreadsheet program from the companion website (thermal-expansion-of-liquid.xls) calculates the thermal expansion of compounds for respective minimum and maximum temperatures, as denoted by  $T_{\min}$  and  $T_{\max}$  in Table C-21 in Appendix C. Figure 3-20 shows the plot of coefficient of thermal expansion of *n*-pentane as a function of temperature.

#### EXAMPLE 3-29

Calculate the thermal expansion coefficient of benzene ( $C_6H_6$ ) at a temperature of 40°C.

*Solution*

Substituting the correlation constants from Table C-21 in Appendix C, and at a temperature of 40°C into the correlation equation yields

$$B_{\text{liq}} = 0.000661 \left( 1 - \frac{313.15}{562.16} \right)^{-0.7182}$$

$$= 0.001186^{\circ}C^{-1}$$

#### 3.21 VOLUMETRIC EXPANSION RATE

The volumetric expansion flow rate for a liquid contained in process equipment that undergoes thermal expansion from heat input is expressed by [2]:

$$Q_v = \frac{B_{\text{liq}}}{\rho_{\text{liq}} C_p} UA (T_{\text{ext}} - T) \quad (3-33)$$

where

$Q_v$  = volumetric expansion rate

$\rho_{\text{liq}}$  = liquid density

$C_p$  = heat capacity of liquid

$U$  = overall heat transfer coefficient

$A$  = area of heat transfer

$T_{\text{ext}}$  = external temperature

$T$  = temperature of liquid.

This equation describes the volumetric expansion rate at the beginning of the heat transfer, and is applicable for the design of relief systems. The relief systems should be sized to accommodate this volumetric flow. The Excel spreadsheet program from the companion website (thermal-expansion-of-liquid.xls) calculates the volumetric expansion rate of compounds.

#### 3.22 ADSORPTION ON ACTIVATED CARBON

Adsorption on activated carbon is an effective method for removing volatile organic compounds (VOC) from gases. The capacity data for adsorption of volatile organic compounds on activated carbon is becoming increasingly important in engineering and environmental studies, and are useful in the engineering design of carbon adsorption systems to remove trace pollutants from gases, especially in air pollution [2].

The correlation for adsorption on activated carbon is based on a logarithmic series expansion of concentration in the gas, and is expressed by

$$\log_{10} Q = A + B [\log_{10} y] + C [\log_{10} y]^2 \quad (3-34)$$

where

$Q$  = adsorption capacity at equilibrium, 1 g of compound/100 g of carbon

$y$  = concentration in gas at 25°C, and 1 atm, parts per million by volume, ppmv

$A$ ,  $B$ , and  $C$  = correlation constants.

Table C-22 in Appendix C lists the correlation constants for adsorption capacity of activated carbon of VOC. The correlations constants were determined from regression of the available data for adsorption on activated carbon. The Excel spreadsheet program from the companion website (adsorption-on-activatedcarbon.xls) calculates the adsorption capacity at equilibrium (g of compound/100 g of carbon). Figures 3-21 and 3-22 show plots of adsorption capacity of activated carbon for

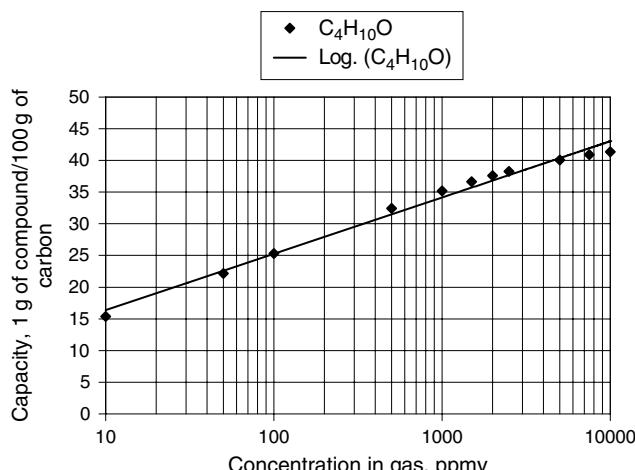


Figure 3-21 Adsorption capacity of activated carbon for removing n-Butanol ( $C_4H_{10}O$ ).

**EXAMPLE 3-30**

The tubing in a reactor contains benzene ( $C_6H_6$ ) at  $30^\circ C$  ( $86^\circ F$ ). Other data are as follows:

Heat capacity of liquid,  $C_p = 0.413 \text{ Btu/lb } ^\circ F$

Overall heat transfer coefficient,  $U = 40 \text{ Btu/h ft}^2 {}^\circ F$

Surface area of tubing,  $A = 550 \text{ ft}^2$ .

Calculate the volumetric expansion rate if the tubing is exposed to  $600^\circ F$  superheated steam.

$$\begin{aligned} T_{ex} &= 600^\circ F \\ T &= 86^\circ F \end{aligned}$$

**Solution**

Calculate the thermal expansion coefficient of liquid benzene at  $30^\circ C$ . First calculate the density of benzene at  $30^\circ C$

$$\begin{aligned} \text{density}(\rho_{liq}) &= AB^{-(1-T_r)^n} \\ &= (0.3009)(0.2677)^{-(1-\frac{303.15}{562.16})^{0.2818}} \\ &= 0.8679 \text{ g/mL} \end{aligned}$$

where

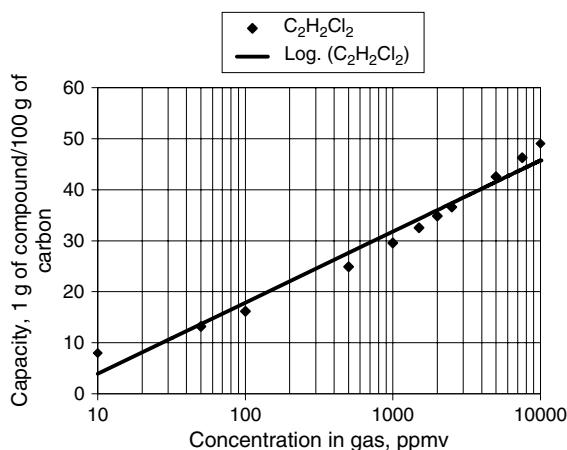
$T_r$  = Pseudo-reduced temperature

$A$ ,  $b$ , and  $n$  = regression coefficients for chemical compound.

$$\begin{aligned} B_{liq} &= a \left(1 - \frac{T}{T_c}\right)^m \\ &= (6.606 \times 10^{-4}) \left(1 - \frac{303.15}{562.16}\right)^{-0.7182} \\ &= 0.00115^\circ C^{-1} \end{aligned}$$

The volumetric expansion flow rate of benzene is expressed by

$$\begin{aligned} Q_v &= \frac{B_{liq}}{\rho_{liq} C_p} UA (T_{ext} - T) \\ Q &= \frac{(1.1525 \times 10^{-3}/1.8)}{(0.8679 \times 62.4 \times 0.413)} (40)(550)(600 - 86) \\ &= 323.71 \text{ ft}^3/\text{h} = 42.34 \text{ gal./min.} \end{aligned}$$



**Figure 3-22** Adsorption capacity of activated carbon for removing 1,2-dichloroethane ( $C_2H_2Cl_2$ ).

removing *n*-butanol and 1,2-dichloroethane respectively for conditions (concentrations in parts per million range in gas at  $25^\circ C$  and 1 atm) encountered in air pollution control. Figures 3-23, 3-24, and 3-25 show representative adsorption systems for removing organic compounds from gases. Figure 3-23 shows an adsorption system with recovery of the organic (such as a solvent) using steam for regeneration. Figure 3-24 shows an adsorption system with thermal or catalytic oxidation of the organic removed from the gas by carbon adsorption. In Figure 3-25, the organic is initially removed from wastewater by air stripping. The air leaving the stripper contains the organic, and is then sent to the adsorption system for recovery of the organic. This can be used to recover organics (e.g., benzene) from process wastewater encountered in the chemical and petroleum-refining industries.

**EXAMPLE 3-31**

The air from an industrial operation contains 10 ppmv of 1,2-dichloroethane ( $C_2H_4Cl_2$ ). Determine the adsorption capacity of activated carbon for removing the compound at  $25^\circ C$  and 1 atm.

**Solution**

Substituting the correlation coefficients from Table C-22 in Appendix C and concentration into the correlation equation yields adsorption capacity at equilibrium.

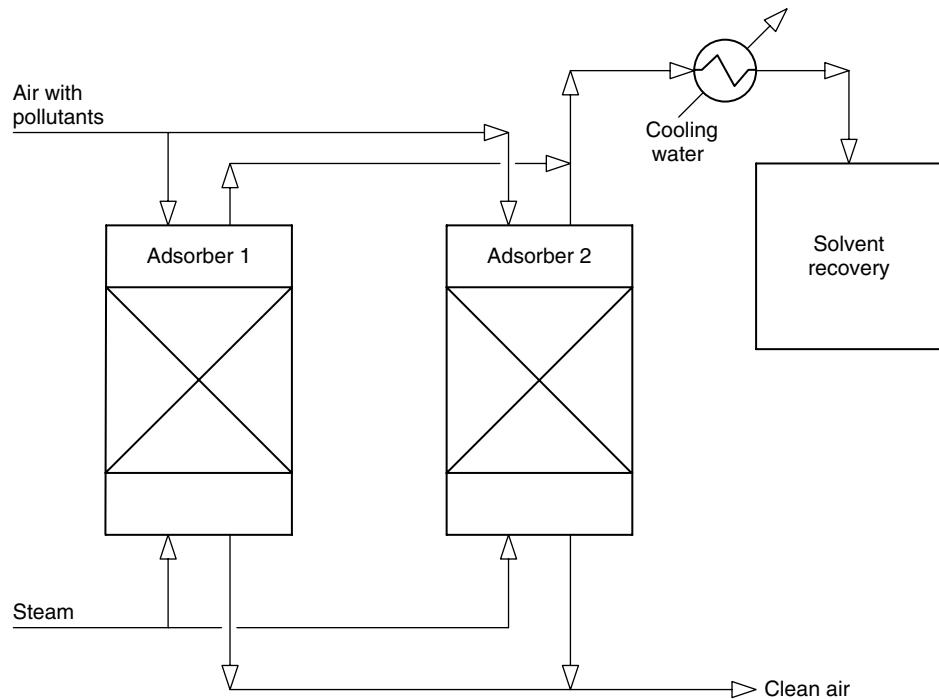
$$\begin{aligned} \log_{10} Q &= 0.5534 + (0.37072) (\log_{10} 10) \\ &\quad + (-0.02161) (\log_{10} 10)^2 \\ &= 0.902511 \\ Q &= 10^{0.902511} \\ &= 7.99 \text{ g of 1,2-dichloroethane/100 g of carbon.} \end{aligned}$$

**3.23 DIFFUSION COEFFICIENTS (DIFFUSIVITIES)**

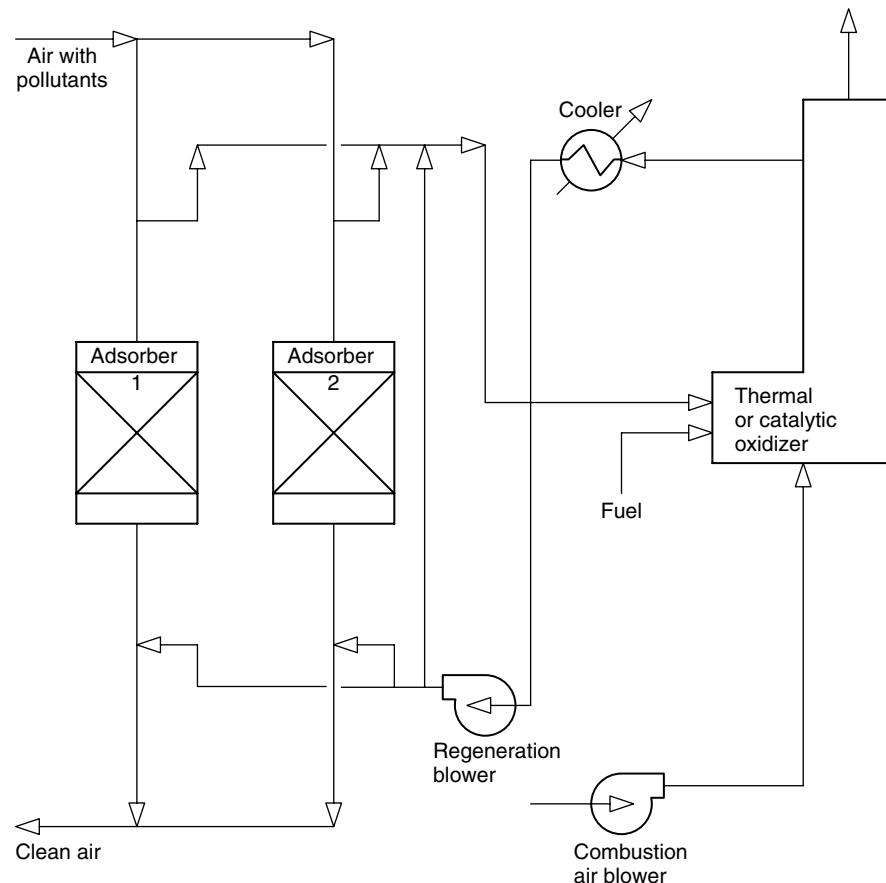
Diffusion coefficients are important in the design of mass transfer operations equipment, such as gas absorption, distillation, and liquid-liquid extraction. Experimental data for the common systems can be found in the literature, but for most design work, the values will have to be estimated.

**LIQUID-PHASE DIFFUSION COEFFICIENT**

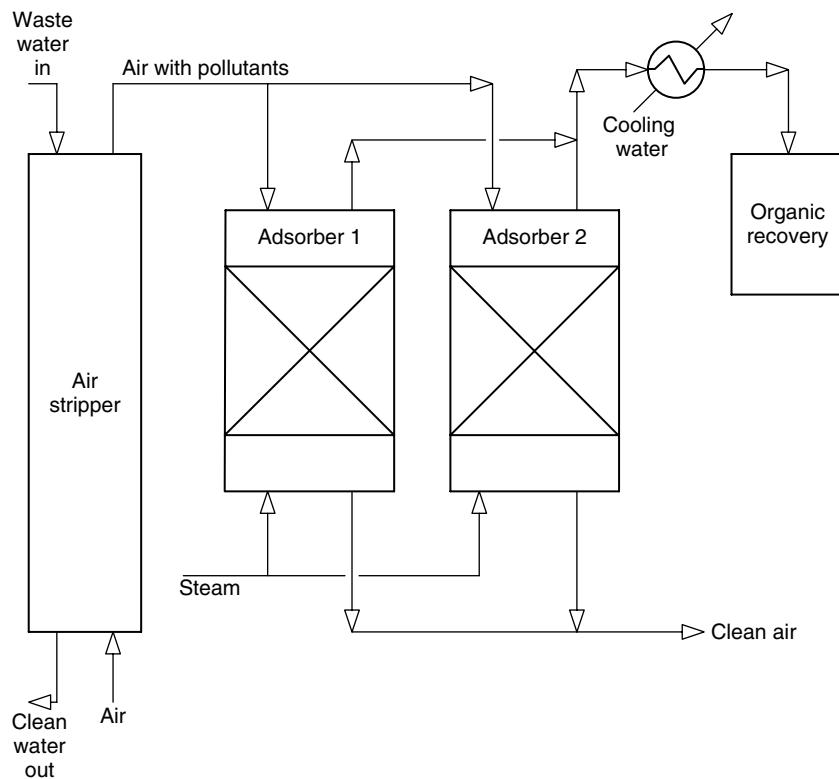
Diffusion coefficients in liquid phases depend on concentration and are valid for dilute solutions; this implies solute concentrations not greater than 10%. In addition, the lower the solute



**Figure 3-23** Adsorption system with solvent recovery. (Source: Yaws [2].)



**Figure 3-24** Adsorption system with thermal or catalytic oxidation. (Source: Yaws [2].)



**Figure 3-25** Adsorption system with air stripper. (Source: Yaws [2].)

concentration, the more accurate the calculated coefficients. For a binary mixture of solute A in solvent B, the diffusion coefficient can be represented as  $D_{AB}^{\circ}$  for concentrations of A up to 5 or 10 mol% [6]. A number of correlations have been proposed for predicting  $D_{AB}^{\circ}$  in dilute liquid solutions [32–35]. Here, the Wilke–Chang method is employed for estimating  $D_{AB}^{\circ}$  as

$$D_{AB}^{\circ} = \frac{7.4 \times 10^{-8} (\phi M_{WB})^{0.5} T}{\mu_B V_A^{0.6}} \quad (3-35)$$

where

$D_{AB}^{\circ}$  = diffusion coefficient of solute A at very low concentrations in solvent B,  $\text{cm}^2/\text{s}$

$M_{WB}$  = molecular weight of solvent B, g/mol

$T$  = absolute temperature, K

$\mu_B$  = viscosity of solvent B, cP

$V_A$  = molal volume of solute A at its normal boiling point,  $\text{cm}^3/\text{mol}$

$\phi$  = association factor of solvent B, dimensionless  
= 2.6 for water (some workers recommend 2.26)  
= 1.9 for methanol  
= 1.5 for ethanol  
= 1.0 for unassociated solvents.

The Le Bas additive volume given by Poling et al. [1] can be used for the value of  $V_A$  if no experimental data are given to obtain  $V_A$  at  $T_{ab}$ .

#### EXAMPLE 3-32

Using the Wilke–Chang relation, determine  $D_{AB}^{\circ}$  for ethylbenzene ( $C_8H_{10}$ ) into water at a temperature of 293 K. The viscosity of water at this temperature is 1.0 cP. The experimental value of  $D_{AB}^{\circ}$  is  $0.81 \times 10^{-5} \text{ cm}^2/\text{s}$ .

#### Solution

The normal boiling point of ethylbenzene is 409.36 K. At this temperature, the density is  $0.76 \text{ g/cm}^3$ .

$$\begin{aligned} \text{Molecular weight, } M_{WA} &= 106.17 \text{ g/mol} \\ V_A &= M_{WA}/\text{density} \\ &= 106.17/0.76 \\ &= 139.51 \text{ cm}^3/\text{mol} \end{aligned}$$

$$\text{Molecular weight } M_{WB} = 18.01 \text{ g/mol}$$

$$\text{Viscosity, } \mu = 1.0 \text{ cP}$$

$$\text{Temperature, } T = 293 \text{ K.}$$

$$\text{Association factor of solvent} = 2.6$$

$$\begin{aligned} D_{AB}^{\circ} &= \frac{7.4 \times 10^{-8} [(2.6)(18.0)]^{0.5} (293)}{(1.0)(139.5)^{0.6}} \\ &= 0.77 \times 10^{-5} \text{ cm}^2/\text{s} \end{aligned}$$

The percentage error is

$$\begin{aligned} \text{Error} &= \left( \frac{D_{AB,\text{expt}} - D_{AB,\text{cal}}}{D_{AB,\text{cal}}} \right) \times 100 \\ &= \left( \frac{0.81 - 0.77}{0.77} \right) \times 100 \\ &= 5.2\%. \end{aligned}$$

### GAS-PHASE DIFFUSION COEFFICIENTS

Diffusion coefficients for non-polar gases may be estimated from Chapman and Cowling [36], Wilke and Lee [37], Fuller et al. [38]. Here, the Fuller, Schettler, and Giddings' method is expressed by

$$D_{AB} = \frac{0.00143 T^{1.75}}{PM_{AB}^{0.5} \left[ (\sum v)_A^{1/3} + (\sum v)_B^{1/3} \right]^2} \quad (3-36)$$

where

$D_{AB}$  = diffusion coefficient for a binary mixture of gases A and B,  $\text{cm}^2/\text{s}$   
 $M_A, M_B$  = molecular weights of A and B, g/mol.

$$M_{AB} = 2 \left[ (1/M_A) + (1/M_B) \right]^{-1} \quad (3-37)$$

$v_A, v_B$  = atomic diffusion volumes  
 $T$  = temperature, K  
 $P$  = pressure, 1 bar.

### 3.24 COMPRESSIBILITY Z-FACTOR OF NATURAL GASES

Many petroleum engineering and process design calculations dealing with natural gases require knowledge of deviation factors or compressibility Z-factors. Experimental data from pressure-volume-temperature (P-V-T) measurements are seldom available. The Z-factors are available in charts or tables as a function of pseudo-reduced temperatures  $T_r$  and pressures  $P_r$ . However, use of these charts is often time-consuming and involve complex calculations.

Computer programs [39–41] for calculating the Z-factors have been developed solely as a function of temperature and pressure of the gas. Furthermore, numerical methods and mathematical representations of the charts have been used to estimate the Z-factors. Takacs [42] reviewed the various methods of estimating the Z-factors. Here, use is made of a modified form of the method

#### EXAMPLE 3-33

Calculate the diffusion coefficient of allyl chloride ( $\text{C}_3\text{H}_5\text{Cl}$ ) in air at 298 K and 1 bar. The experimental value is  $0.0975 \text{ cm}^2/\text{s}$ .

*Solution*

Using the correlation of Fuller–Schettler–Giddings, values of atomic diffusion volumes of  $\sum v$  from Table 3-2 are

$$\begin{aligned} \sum v(\text{AC}) &= 3(\text{C}) + 5(\text{H}) + \text{Cl} \\ &= 3(15.9) + 5(2.31) + 21.0 \\ &= 80.25 \end{aligned}$$

**TABLE 3-2 Atomic Diffusion Volumes for Use in Calculating  $D_{AB}$  by the Correlation of Fuller–Schettler–Giddings**

Atomic and Structural Diffusion-Volume Increments, $v$			
C	15.9	F	14.70
H	2.31	Cl	21.0
O	6.11	Br	21.9
N	4.54	I	29.8
Aromatic ring	-18.3	S	22.9
Heterocyclic ring	-18.3		
Diffusion Volumes for Simple Molecules, $\sum v$			
$\text{H}_2$	6.12	CO	18.0
$\text{D}_2$	6.84	$\text{CO}_2$	26.9
He	2.67	$\text{N}_2\text{O}$	35.9
$\text{N}_2$	18.5	$\text{NH}_3$	20.7
$\text{O}_2$	16.3	$\text{H}_2\text{O}$	13.1
Air	19.7	$\text{SF}_6$	71.3
Ar	16.2	$\text{Cl}_2$	38.4
Kr	24.5	$\text{Br}_2$	69.0
Xe	32.7	$\text{SO}_2$	41.8
Ne	5.98		

where

AC = allyl chloride

$$\sum v(\text{Air}) = 19.7$$

Molecular weight of air,  $M_A = 29.0$

Molecular weight of allyl chloride,  $M_B = 76.53$ .

$$\begin{aligned} M_{AB} &= 2 \left[ (1/29) + (1/76.53) \right]^{-1} \\ &= 42.06 \end{aligned}$$

$T = 298 \text{ K}$

$P = 1 \text{ bar}$ .

Substituting the parameters into the Fuller et al [38] equation yields the diffusion coefficient of the gas phase as follows.

$$\begin{aligned} D_{AB} &= \frac{(0.00143)(298)^{1.75}}{(1.0)(42.06)^{0.5} \left[ (80.25)^{1/3} + (19.7)^{1/3} \right]^2} \\ &= 0.096 \text{ cm}^2/\text{s} \end{aligned}$$

The percentage error is

$$\begin{aligned} \text{Error} &= \left( \frac{D_{AB,\text{exp}} - D_{AB,\text{cal}}}{D_{AB,\text{cal}}} \right) \times 100 \\ &= \left( \frac{0.0975 - 0.096}{0.096} \right) \times 100 \\ &= 2.08\%. \end{aligned}$$

(Source: Poling et al. [1].)

developed by Awoseyin [43]. This method gives a compressibility factor to within 5% for natural hydrocarbon gases with specific gravities between 0.5 and 0.8 and for pressures up to 5000 psia. The Z-factor can be expressed as

$$Z = F_1 \left\{ \frac{1}{1 + \frac{(A_6 P_{10} [1.785 S_g])}{T^{3.825}}} + F_2 F_3 \right\} + F_4 + F_5 \quad (3-38)$$

where

$$F_1 = P (0.251 S_g - 0.15) - 0.202 S_g + 1.106 \quad (3-39)$$

$$F_2 = 1.4 e^{-0.0054(T-460)} \quad (3-40)$$

$$F_3 = A_1 P^5 + A_2 P^4 + A_3 P^3 + A_4 P^2 + A_5 P \quad (3-41)$$

$$F_4 = (0.154 - 0.152 S_g) P^{(3.185 S_g - 1.0)} e^{(-0.5P)} - 0.02 \quad (3-42)$$

$$F_5 = 0.35 \{ (0.6 - S_g) e^{[-1.039(P-1.8)^2]} \} \quad (3-43)$$

$P$  = pressure

$T$  = temperature

The values of the constants  $A_1$ ,  $A_2$ ,  $A_3$ ,  $A_4$ ,  $A_5$ , and  $A_6$  are as follows.

$$A_1 = 0.001946$$

$$A_2 = -0.027635$$

$$A_3 = 0.136315$$

$$A_4 = -0.23849$$

$$A_5 = 0.105168$$

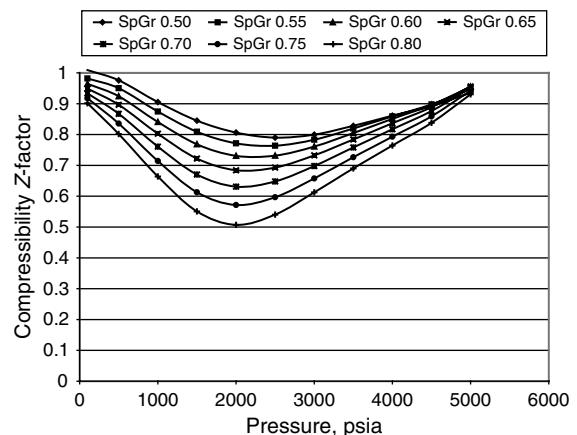
$$A_6 = 3.44 \times 10^8$$

The specific gravity  $S_g$  of natural gas can be calculated from its density or molecular weight. This is expressed as the ratio of the gas density at 60°F and 1 atm (14.7 psia),  $\rho_{\text{gas}, 60^\circ\text{F}}$ , to the density of air,  $\rho_{\text{air}, 60^\circ\text{F}}$ , under the same conditions.

$$S_g = \frac{\text{density of gas}}{\text{density of air}} \quad (3-44)$$

**TABLE 3-3 Equations of State (for 1 mol)**

van der Waals	Soave–Redlich–Kwong	Peng–Robinson	Redlich–Kwong
$(p + \frac{a}{V^2})(\hat{V} - b) = RT$	$p = \frac{RT}{\hat{V} - b} - \frac{a' \lambda}{\hat{V}(\hat{V} + b)}$	$p = \frac{RT}{\hat{V} - b} - \frac{a\alpha}{\hat{V}(\hat{V} + b) + b(\hat{V} - b)}$	$p = \frac{RT}{(\hat{V} - b)} - \frac{a}{\hat{V}(\hat{V} + b)T^{1/2}}$
$a = \left(\frac{27}{64}\right) \frac{R^2 T_c^2}{P_c}$	$a' = \frac{0.42747 R^2 T_c^2}{P_c}$	$a = 0.45724 \left(\frac{R^2 T_c^2}{P_c}\right)$	$a = 0.42747 \frac{R^2 T_c^{2.5}}{P_c}$
$b = \left(\frac{1}{8}\right) \frac{RT_c}{P_c}$	$b = \frac{0.08664 RT_c}{P_c}$	$b = 0.07780 \left(\frac{RT_c}{P_c}\right)$	$b = 0.08664 \frac{RT_c}{P_c}$
	$\lambda = [1 + m(1 - \sqrt{T_r})]^2$	$\alpha = [1 + k(1 - T_r^{1/2})]^2$	
	$m = 0.48508 + 1.55171\omega - 0.1561\omega^2$	$\kappa = 0.37464 + 1.54226\omega - 0.26992\omega^2$	
	$T_r = T/T_c$	$\omega = \text{acentric factor}$	



**Figure 3-26** Plots of compressibility  $z$ -factor of natural gases at 60°F, and at specific gravities of 0.5–0.8.

Using the molecular weight of the gas,  $S_g$  can be expressed as

$$S_g = \frac{\text{molecular weight of gas}}{\text{molecular weight of air}} = \frac{M_{W,gas}}{M_{W,air}} \quad (3-45)$$

The Excel spreadsheet program from the companion website (compressibility-Z-factor.xls) calculates the compressibility  $Z$ -factor of natural gases as a function of temperature, pressure, and specific gravity. Figure 3-26 gives plots of the compressibility  $Z$ -factor of natural gases at 60°F as a function of pressure and specific gravity.

### 3.25 GENERALIZED COMPRESSIBILITY Z-FACTOR

The two-parameter Redlich–Kwong (R–K) equation of state has an accuracy that compares well with more complicated equations that use more constants (when applied to non-polar compounds). The R–K equation is a cubic equation in the volume (or in the compressibility factor) for which analytical solutions can be found. Table 3-3 lists a few of the commonly used equations of state from among the hundreds that have been proposed that involve two or more coefficients. These equations are empirical and have proved remarkably

**TABLE 3-4 Values of the Pitzer Acentric Factor**

Compound	Acentric Factor ( $\omega$ )	Compound	Acentric Factor ( $\omega$ )
Acetone	0.309	Hydrogen sulfide	0.100
Benzene	0.212	Methane	0.008
Ammonia	0.250	Methanol	0.559
Argon	0.000	<i>n</i> -butane	0.193
Carbon dioxide	0.225	<i>n</i> -pentane	0.251
Carbon monoxide	0.049	Nitric oxide	0.607
Chlorine	0.073	Nitrogen	0.040
Ethane	0.098	Oxygen	0.021
Ethanol	0.635	Propane	0.152
Ethylene	0.085	Propylene	0.148
Freon-12	0.176	Sulfur dioxide	0.251
Hydrogen	-0.220	Water vapor	0.344

(Source: Pitzer, K.S., *J. Am. Chem. Soc.*, Vol. 77, 1954, p. 3427.)**TABLE 3-5 Constants for the van der Waals and Redlich–Kwong Equations**

	van der Waals		Redlich–Kwong	
	$a^*$ [atm $(\frac{\text{cm}^3}{\text{g mol}})^2$ ]	$b^*$ $(\frac{\text{cm}^3}{\text{g mol}})$	$a^{\dagger\dagger}$ [(atm) $(\text{K}^{1/2}) (\frac{\text{cm}^3}{\text{g mol}})^2$ ]	$b^*$ $(\frac{\text{cm}^3}{\text{g mol}})$
Air	$1.33 \times 10^6$	36.6	$15.65 \times 10^6$	25.3
Ammonia	$4.19 \times 10^6$	37.3	$85.00 \times 10^6$	25.7
Carbon dioxide	$3.60 \times 10^6$	42.8	$63.81 \times 10^6$	29.7
Ethane	$5.50 \times 10^6$	65.1	$97.42 \times 10^6$	45.1
Ethylene	$4.48 \times 10^6$	57.2	$76.92 \times 10^6$	39.9
Hydrogen	$0.246 \times 10^6$	26.6	$1.439 \times 10^6$	18.5
Methane	$2.25 \times 10^6$	42.8	$31.59 \times 10^6$	29.6
Nitrogen	$1.347 \times 10^6$	38.6	$15.34 \times 10^6$	26.8
Oxygen	$1.36 \times 10^6$	31.9	$17.12 \times 10^6$	22.1
Propane	$9.24 \times 10^6$	90.7	$180.5 \times 10^6$	62.7
Water vapor	$5.48 \times 10^6$	30.6	$140.9 \times 10^6$	21.1

(Source: Himmelblau [47].)

\* To convert to psia ( $\text{ft}^3/\text{lb mol}$ )<sup>2</sup>, multiply table value by  $3.776 \times 10^{-3}$ .† To convert to  $\text{ft}^3/\text{lb mol}$ , multiply table value by  $1.60 \times 10^{-2}$ .†† To convert to psia ( $\text{R}^{1/2}$  ( $\text{ft}^3/\text{lb mol}$ )<sup>2</sup>), multiply table value by  $5.067 \times 10^{-3}$ .

robust in describing a wide variety of systems. Tables 3-4 and 3-5 list a few values of acentric factor ( $\omega$ ) for common gases, and some of the values of the constants in the van der Waals' and Redlich–Kwong (R–K) equations respectively. Here, the R–K expression is used to illustrate the general characteristics of cubic equations of state.

## EQUATIONS

The R–K equation is expressed by [44]:

$$P = \frac{RT}{V - b} - \frac{a}{V(V + b)T^{0.5}} \quad (3-46)$$

where

$$a = 0.42747 \left( \frac{R^2 T_c^{2/5}}{P_c} \right) \quad (3-47)$$

$$b = 0.08664 \left( \frac{RT_c}{P_c} \right) \quad (3-48)$$

and

$P$  = Pressure, atm

$V$  = Molar volume, L/g-mol

$T$  = Temperature, K

$R$  = Universal gas constant,  $R = 0.08206$  (atm L/g-mol K)

$T_c$  = critical temperature, K

$P_c$  = critical pressure, atm.

Eliminating  $V$  from Eq. (3-46) and writing it as a cubic equation of the compressibility factor  $Z$  gives

$$f(Z) = Z^3 - Z^2 - qr - r = 0 \quad (3-49)$$

where

$$r = A^2 B$$

$$q = B^2 + B - A^2$$

$$A^2 = 0.42747 \left( \frac{P_R}{T_{c,5}^{2.5}} \right)$$

$$B = 0.08664 \left( \frac{P_R}{T_R} \right), \quad P_r = P/P_c \text{ and } T_r = T/T_c.$$

Equation (3-49) can be solved analytically for three roots, some of which may be complex. Considering only the real roots, the sequence of calculations involves the following steps:

$$C = \left(\frac{f}{3}\right)^3 + \left(\frac{g}{2}\right)^2 \quad (3-50)$$

where

$$f = \frac{-3q - 1}{3}$$

$$g = \frac{-27r - 9q - 2}{27}.$$

If  $C > 0$ , there is one real solution for  $Z$ , which can be expressed by

$$Z = D + E + \frac{1}{3} \quad (3-51)$$

where

$$D = \left(-\frac{g}{2} + \sqrt{C}\right)^{1/3}$$

$$E = \left(-\frac{g}{2} - \sqrt{C}\right)^{1/3}.$$

If  $C < 0$ , there are three real solutions for  $Z$ , and these are expressed by

$$Z_i = 2\sqrt{\frac{-f}{3}} \cos\left[\left(\frac{\theta}{3}\right) + \frac{2\pi(i-1)}{3}\right] + \frac{1}{3} \quad (3-52)$$

where

$$i = 1, 2, 3, \dots, n$$

$$\theta = a \cos \sqrt{\frac{g^2/4}{-f^3/27}}.$$

After computing the compressibility factor, the molar volume ( $V$ ), the isothermal enthalpy departure ( $\Delta H^*$ ), the isothermal entropy departure ( $\Delta S^*$ ), and the fugacity coefficient ( $\psi$ ) can then be calculated from the following equations as follows.

$$V = \frac{ZRT}{P} \quad (3-53)$$

$$\frac{\Delta H^*}{RT} = \frac{3a}{2bRT^{1.5}} \ln\left(1 + \frac{b}{V}\right) - (Z-1) \quad (3-54)$$

$$\frac{\Delta S^*}{R} = \frac{a}{2bRT^{1.5}} \ln\left(1 + \frac{b}{V}\right) - \ln\left(Z - \frac{Pb}{RT}\right) \quad (3-55)$$

$$\psi = \exp \left\{ Z - 1 - \ln \left[ Z \left(1 - \frac{b}{V}\right) \right] - \frac{a}{2bRT^{1.5}} \ln \left(1 + \frac{b}{V}\right) \right\}. \quad (3-56)$$

In the supercritical region, two of these solutions are negative, so the maximal  $Z_i$  is selected as the true compressibility factor. The generalized compressibility factor similar to that shown in

Figure 3-27 should not be used for strongly polar fluids, nor are they recommended for helium, hydrogen, or neon unless special, modified critical constants are used [1]. Many versions of Figure 3-27 have been published, although all differ somewhat, as each reflects the choice of experimental data and how they are smoothed. In some of the charts,  $V$  can be found directly from the lines of  $V_r = V/(RT_c/P_c)$ . Equations of state (EOS) mathematically represent graphical information as shown in Poling et al. [1], and modern computers obviate the need to manually obtain volumetric behavior such as graphs and also allow more accurate results by using equations with component-specific parameters.

### 3.26 GAS MIXTURES

For gas mixtures, Kay's method using pseudocritical values of mixtures of gases is employed on the assumption that each component in the mixture contributes to the pseudocritical value in the same proportion as the number of moles of that component. Thus, the pseudocritical values are calculated as [45].

$$\begin{aligned} \text{Pseudocritical temperature } T'_c &= y_A T_{cA} + y_B T_{cB} \\ &\quad + y_C T_{cC} + \dots \end{aligned} \quad (3-57)$$

$$\begin{aligned} \text{Pseudocritical pressure } P'_c &= y_A P_{cA} + y_B P_{cB} \\ &\quad + y_C P_{cC} + \dots \end{aligned} \quad (3-58)$$

where  $y_A, y_B, y_C, \dots$  are mole fractions of species A, B, C, ... in the mixture. Assuming that the system temperature  $T$  and pressure  $P$  are known, the pseudocritical properties can be used to estimate the pseudoreduced temperature and pressure of the mixture as

$$\text{Pseudoreduced temperature } T'_r = T/T'_c \quad (3-59)$$

$$\text{Pseudoreduced pressure } P'_r = P/P'_c. \quad (3-60)$$

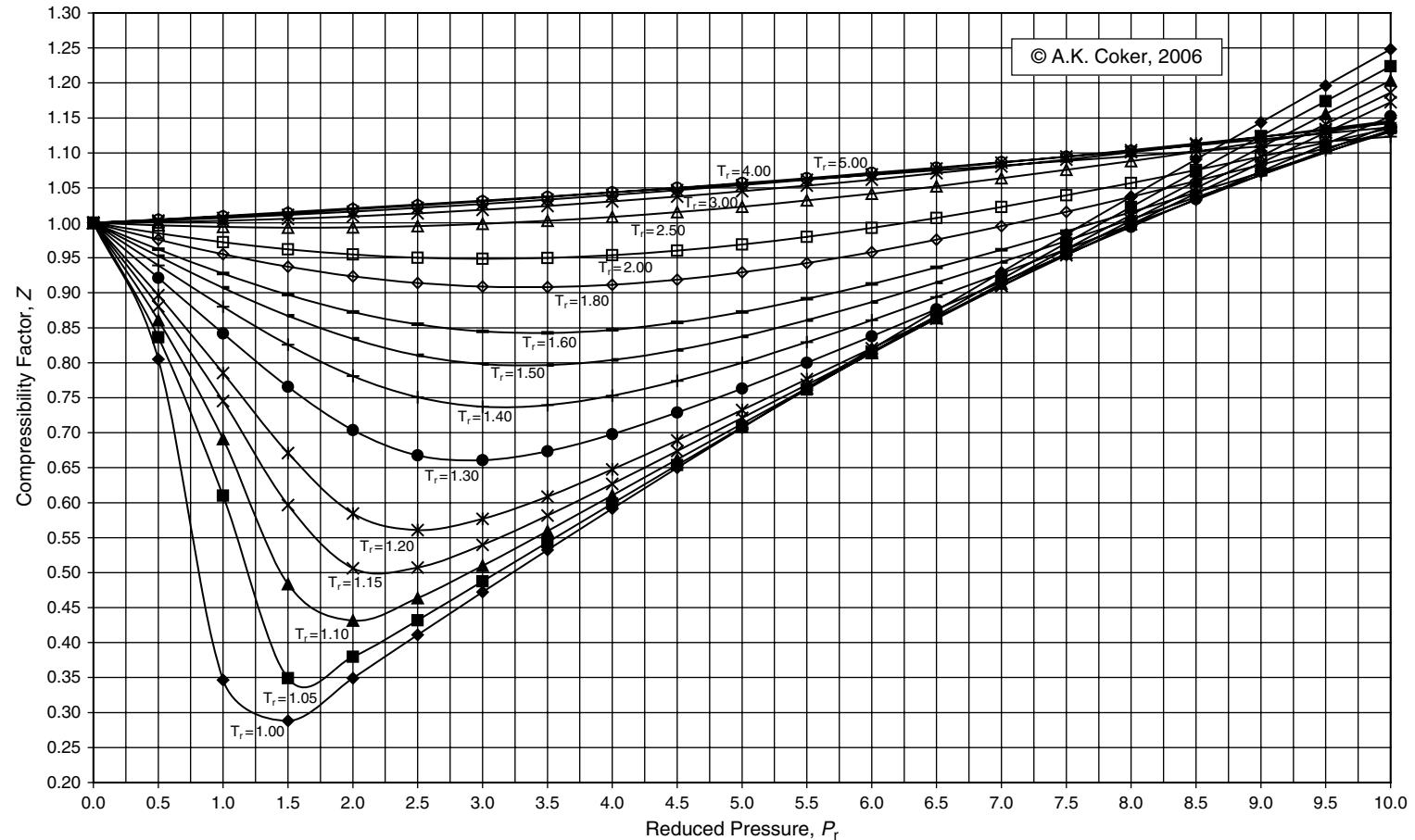
The compressibility factor for a gas mixture  $Z_m$  can then be estimated from the compressibility charts and the pseudoreduced properties, and  $\hat{V}$  for the mixture can be calculated as

$$\hat{V} = \frac{Z_m RT}{P}. \quad (3-61)$$

As with single-component gases, if  $\hat{V}$  and either  $T$  or  $P$  are known, the ideal pseudocritical volume  $\hat{V}_r^{\text{ideal}} = \hat{V}P'_c/RT'_c$  and the other known reduced property are used to determine the unknown temperature and pressure from the compressibility chart.

Kay's rule provides only approximate values of the quantities it is used to calculate, and works best when applied for mixtures of non-polar compounds whose critical temperatures and pressures are within a factor of two of one another [46].

The Excel spreadsheet program from the companion website (generalized-compressibility-Z-factor.xls) calculates the compressibility  $Z$ -factor as a function of reduced temperature and pressure, and Figure 3-27 shows a plot of the generalized compressibility  $Z$ -factor using critical temperature for water  $T_c = 647.4$  K, and critical pressure of water  $P_c = 218.3$  atm.



**Figure 3-27** Generalized compressibility factor chart.

**EXAMPLE 3-34**

A stream of propane at temperature  $T = 423\text{ K}$  and pressure  $P$  (atm) flows at a rate of  $100.0\text{ kmol/h}$ . Use the R-K equation of state to estimate the volumetric flow rate of the stream for  $P = 0.7\text{ atm}$ ,  $7\text{ atm}$ , and  $70\text{ atm}$ . In each case, calculate the percentage differences between the predictions of the R-K equation and the ideal gas equation of state [46].

*Solution*

The calculation of  $\hat{V}$  ( $\text{L/mol}$ ) proceeds as follows. The R-K equation of state is expressed in the form

$$f(\hat{V}) = P - \frac{RT}{V-b} + \frac{a}{V(V+b)T^{0.5}} = 0$$

The values of  $T_c$  and  $P_c$  are looked up;  $a$  and  $b$  are calculated from the given formulas; specified values of  $T$  and  $P$  are substituted; and the value of  $\hat{V}$  for which  $f(\hat{V}) = 0$  is found by trial and error.

$$T_c = 369.9\text{ K}, P_c = 42\text{ atm}, R = 0.08206\text{ (L-atm)/(mol K)}$$

$$\begin{aligned} a &= 0.42747 \left( \frac{R^2 T_c^{2/5}}{P_c} \right) \left( \frac{\text{L-atm}}{\text{mol K}} \right)^2 \left( \frac{\text{K}^2 \text{K}^{0.5}}{\text{atm}} \right) \\ &= \left( \frac{\text{L}^2 - \text{atm K}^{0.5}}{\text{mol}^2} \right) \\ &= 180.36 \left( \frac{\text{L}^2 - \text{atm K}^{0.5}}{\text{mol}^2} \right) \\ b &= 0.08664 \left( \frac{RT_c}{P_c} \right) \left( \frac{\text{L-atm}}{\text{mol K}} \right) \left( \frac{\text{K}}{\text{atm}} \right) = \frac{\text{L}}{\text{mol}} \\ &= 0.062\text{ (L/mol)} \end{aligned}$$

Microsoft Excel - Example 3-34

Spreadsheet of Example 3-34 by A.K. Coker

	P	V <sub>ideal</sub>	V	f(V)	Error	V
(atm)	(L/mol)	(L/mol)	(atm)	(%)	(m <sup>3</sup> /h)	
0.7	49.59	49.40	1.04808E-05	0.39	4939.8	
7	4.958769	4.77	0.000457865	4.05	476.6	
70	0.495877	0.27	9.83496E-05	64.86	26.8	

Figure 3-28 Snapshot of Microsoft Excel worksheet of Example 3-34.

(continued)

**EXAMPLE 3.34—(continued)**

The percentage difference between  $\hat{V}_{RK}$  and  $\hat{V}_{ideal} = (RT/P)$  is

$$\text{Error\%} = \frac{\hat{V}_{ideal} - \hat{V}_{RK}}{\hat{V}_{RK}} \times 100\%$$

Once  $\hat{V}$  is known for a given  $P$ , the volumetric flow rate corresponding to a molar flow rate of 100 kmol/h is

$$\begin{aligned}\hat{V} (\text{m}^3/\text{h}) &= \frac{V^3 \times (\text{L})}{\text{mol}} \times \frac{10^3 \text{ mol}}{1 \text{ kmol}} \times \frac{1 \text{ m}^3}{10^3 \text{ L}} \times \frac{100.0 \text{ kmol}}{\text{h}} \\ &= 100.0 \hat{V} (\text{L/mol})\end{aligned}$$

The Excel spreadsheet (Example 3-34.xls) calculates the volumetric flow rate of the stream for  $P = 0.7 \text{ atm}$ ,  $7 \text{ atm}$ , and  $70 \text{ atm}$ . In each case, the Excel program calculates the percentage difference between the predictions of the R-K equation and the ideal gas equation of state.

Figure 3.28 (Example 3-34.xls) shows the display contents of rows 1, 3, 4, 5, 7, and 8, which are entered exactly except for the given formulas in cells B4, D4 and B5. After the cell contents of row 9 have been entered, they are copied into rows 10 and 11, and the pressures in column A are then changed to their desired values. The entries in cells C9–C11 (the initial guesses for  $\hat{V}$ ) are the values copied from the adjacent cells in column B (the values obtained using the ideal gas equation of state). The correct values are then obtained by trial and error; for example, the value in cell C9 is varied until the value in cell D9 is close to zero using the Goal Seek tool in Add-on of the Excel tool and similarly for rows 10 and 11.

A published study provides experimental data for the PVT behavior of propane [47]. The data indicate that at  $423 \text{ K}$  and  $70 \text{ atm}$ , the value of  $\hat{V}$  is  $0.2579 \text{ L/mol}$ . The percentage error in the R-K estimate ( $\hat{V} = 0.2682 \text{ L/mol}$ ) is 3.9%, and that in the ideal gas estimate ( $\hat{V}_{ideal} = 0.4959 \text{ L/mol}$ ) is 92%.

**EXAMPLE 3.35**

A gaseous mixture has the following composition (in mole percent) at  $90 \text{ atm}$  pressure and  $100^\circ \text{C}$  as shown [47].

Component	Mole Percent (%)	$T_c$ (K)	$P_c$ (atm)
Methane ( $\text{CH}_4$ )	20	191	45.8
Ethylene ( $\text{C}_2\text{H}_4$ )	30	283	50.9
Nitrogen ( $\text{N}_2$ )	50	126	33.5
Total	100		

Compare the volume per mole as computed by the methods of

- a) The perfect gas law
- b) The pseudoreduced technique (Kay's method).

$$\text{Use } R = 82.06 \frac{(\text{cm}^3)(\text{atm})}{(\text{g mol})(\text{K})}$$

*Solution*

- (a) From the perfect gas law:

$$PV = nRT$$

Basis: 1 g mol of gas mixture

$$\begin{aligned}V &= \frac{nRT}{P} \\ &= \frac{1 (82.06) (373)}{90} \left\{ \frac{(\text{g mol})(\text{cm}^3)(\text{atm})(\text{K})}{(\text{g mol})(\text{K})} \right\} \\ &= 340.1 \text{ cm}^3 \text{ at } 90 \text{ atm and } 373 \text{ K}\end{aligned}$$

- (b) Using Kay's method, the pseudocritical values for the mixture are as follows.

$$\text{Pseudocritical temperature } T'_c = y_A T_{cA} + y_B T_{cB} + y_C T_{cC} + \dots$$

$$\text{Pseudocritical pressure } P'_c = y_A P_{cA} + y_B P_{cB} + y_C P_{cC} + \dots$$

$$\begin{aligned}T'_c &= (191)(0.2) + (283)(0.3) + (126)(0.5) \\ &= 186 \text{ K}\end{aligned}$$

$$\begin{aligned}P'_c &= (45.8)(0.2) + (50.9)(0.3) + (33.5)(0.5) \\ &= 41.2 \text{ atm.}\end{aligned}$$

$$\text{Pseudoreduced temperature } T'_r = T/T'_c$$

$$T'_r = \frac{373}{186} = 2.01$$

$$\text{Pseudoreduced pressure } P'_r = P/P'_c$$

$$P'_r = \frac{90}{41.2} = 2.18$$

Using these parameters, the compressibility Z factor is 0.965, and  $\hat{V}$  is

$$\begin{aligned}\hat{V} &= \frac{ZnRT}{P} = \frac{(0.965)(1)(82.06)(373)}{90} \\ &= 328.2 \text{ cm}^3 \text{ at } 90 \text{ atm and } 373 \text{ K}\end{aligned}$$

Error percent between the perfect gas law and the Kay's method is

$$\begin{aligned}\text{Error\%} &= \left( \frac{V - \hat{V}}{\hat{V}} \right) \times 100 \\ &= \left( \frac{340.1 - 328.2}{328.2} \right) \times 100 \\ &= 3.6\%.\end{aligned}$$

**NOMENCLATURE**

A	Area of heat transfer, ft <sup>2</sup>
A	Regression coefficient for chemical compound
B	Regression coefficient for chemical compound
B <sub>liq</sub>	Thermal expansion of liquid, 1/°C
C	Regression coefficient for chemical compound
C <sub>p</sub>	Heat capacity of ideal gas, Btu/lb °F, J/mol K
D	Regression coefficient for chemical compound
D <sub>AB</sub>	Diffusion coefficient for a binary mixture of gases A and B, cm <sup>2</sup> /s
D <sub>AB</sub> <sup>0</sup>	Diffusion coefficient of solute A at very low concentrations in solvent B, cm <sup>2</sup> /s
E	Regression coefficient for chemical compound
F	Regression coefficient for chemical compound
H	Henry's law constant at pressure of 1 atm (101.325 kPa) atm/mol
k	Thermal conductivity of liquid or solid, W/(m K)
k <sub>liq</sub>	Thermal conductivity of liquid, W/(m K)
MF <sub>i</sub>	Mole fraction of component i
MW <sub>i</sub>	Mole weight of component i
MW <sub>B</sub>	Molecular weight of solvent B, g/mol
n	Regression coefficient for chemical compound
P <sub>c</sub>	Critical pressure, atm
P' <sub>n</sub>	Pseudo-reduced pressure
P <sub>v</sub>	Vapour pressure, mmHg
P <sub>t</sub>	Total pressure, atm
Q	Adsorption capacity at equilibrium, 1 g of compound/100 g of carbon
Q <sub>v</sub>	Volumetric expansion rate, gpm
R	Universal gas constant, R = 8.314 J/mol. K (0.08206 atm L/mol g K)
S	Solubility in water, ppm (wt)
S <sub>g</sub>	Specific gravity of natural gas
T	Temperature, °F, K
T <sub>ab</sub>	Absolute temperature
T <sub>B</sub>	Boiling point temperature of compound, K
T <sub>br</sub>	Reduced boiling point temperature, K
T <sub>c</sub>	Critical temperature, K
T <sub>ext</sub>	External temperature, °F
T <sub>min</sub>	Minimum temperature, K
T <sub>max</sub>	Maximum temperature, K
T' <sub>r</sub>	Pseudo-reduced temperature
U	Overall heat transfer coefficient, Btu/hr ft <sup>2</sup> °F (W/m <sup>2</sup> °C)
V <sub>A</sub>	Molal volume of solute A at its normal boiling temperature, cm <sup>3</sup> /mol
X	Concentration of salt (NaCl) in water, ppm (wt)
x <sub>i</sub>	Liquid concentration of component i, ppm
x <sub>wt</sub>	Solubility in water at pressure of 1 atm weight fraction
y	Concentration in gas at 25°C, and 1 atm, ppm (vol)
y <sub>i</sub>	Vapour concentration of component i, ppm (mol)

**GREEK LETTERS**

$\rho_{liq}$	Liquid density
$\sigma$	Surface tension, dyn/cm
$\mu_B$	Viscosity of solvent B, cP
$\mu_G$	Viscosity of gas, $\mu$ P
$\mu_i$	Gas viscosity of component i, $\mu$ P
$\mu_L$	Viscosity of liquid, cP
$\mu_m$	Gas mixture viscosity, $\mu$ P
$\gamma^\infty$	Activity coefficient at infinite dilution in water
$\phi$	Association factor of solvent B, dimensionless

$\Delta H$	Change in enthalpy, J/mol
$\Delta H_f$	Enthalpy of formation of ideal gas, kJ/mol
$\Delta H_v$	Enthalpy of vaporization, kJ/mol
$\Delta H_r$	Enthalpy of reaction, kJ/mol
$\Delta G_f$	Gibbs energy of formation of ideal gas, kJ/mol
$\Delta G_r$	Gibbs energy of reaction, kJ/mol

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## FURTHER READING

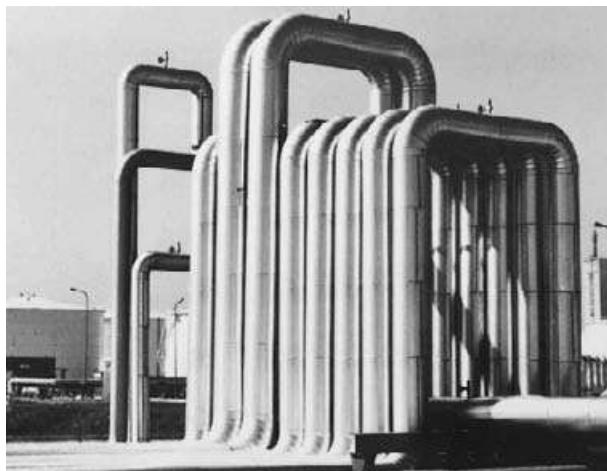
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# 4

## FLUID FLOW

### 4.1 INTRODUCTION

Transportation of fluids is important in the design of chemical process plants. In the chemical process industries (CPI), pipework and its accessories such as fitting make up 20–30% of the total design costs and 10–20% of the total plant investment. Maintenance requirements and energy usage in the form of pressure drop ( $\Delta P$ ) in the fluids being pumped add to the cost. Also, these items escalate each year in line with inflation. As a result, sound pipe-sizing practices can have a substantial influence on overall plant economics. It is the designer's responsibility to optimize the pressure drops in piping and equipment and to assess the most economic conditions of operations. Figure 4-1 illustrates piping layouts in a chemical plant.



(a)



(b)

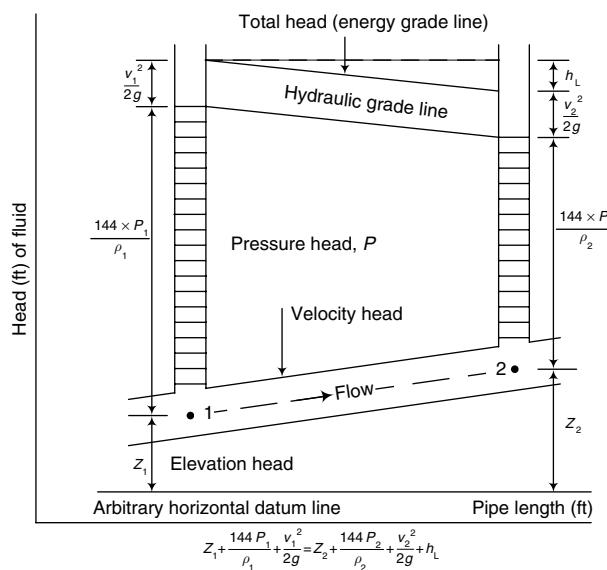
**Figure 4-1** Chemical plant piping layout. (Source: IChemE safer piping training package (Courtesy of the IChemE, UK))

The characteristics and complexity of flow pattern are such that most flows are described by a set of empirical or semi-empirical equations. These relate the pressure drop in the flow system as a function of flow rate, pipe geometry, and physical properties of the fluids. The aim in the design of fluid flow is to choose a line size and piping arrangement that achieve minimum capital and pumping costs. In addition, constraints on pressure drop and maximum allowable velocity in the process pipe should be maintained. These objectives require many trial-and-error computations which are well suited with the aid of a computer.

### 4.2 FLOW OF FLUIDS IN PIPES

Pressure drop or head loss in a piping system is caused by fluid rising in elevation, friction, shaft work (e.g., from a turbine), and turbulence due to sudden changes in direction or cross-sectional area. Figure 4-2 shows the distribution of energy between two points in a pipeline. The mechanical energy balance equation expresses the conservation of the sum of pressure, kinetic, and potential energies, the net heat transfer  $q$ , the work done by the system  $w$ , and the frictional energy  $e_f$ . The  $e_f$  term is usually positive and represents the rate of irreversible conversion of mechanical energy into thermal energy or heat, and is sometimes called head loss, friction loss, or frictional pressure drop. Ignoring this factor would imply no energy usage in piping.

$$q - w = \frac{1}{\rho} \int_1^2 dP + \alpha \int_1^2 v dv + g \int_1^2 dz + e_f \quad (4-1)$$



**Figure 4-2** Distribution of fluid energy in a pipeline.

Integrating Eq. (4-1) gives

$$q - w = \frac{1}{\rho} (P_2 - P_1) + \alpha \frac{(v_2^2 - v_1^2)}{2} + g(z_2 - z_1) + e_f \quad (4-2)$$

or

$$\begin{aligned} -\Delta P &= \frac{(P_1 - P_2)}{\rho} = \frac{1}{2} (v_2^2 - v_1^2) + g(z_2 - z_1) \\ &\quad + e_f + w - q \end{aligned} \quad (4-3)$$

where

$P$  = pressure (force/area)

$\rho$  = fluid density (mass/volume)

$g$  = acceleration due to gravity (length/time<sup>2</sup>)

$z$  = vertical height from some datum (length)

$v$  = fluid velocity (length/time)

$e_f$  = irreversible energy dissipated between points 1 and 2 (length<sup>2</sup>/time<sup>2</sup>)

$w$  = work done per unit mass of fluid [net work done by the system is (+), work done on the system is (-) (mechanical energy/time)]

$q$  = net amount of heat transferred into the system (mechanical energy/time)

$\Delta$  = difference between final and initial points

$\alpha$  = kinetic energy correction factor ( $\alpha \approx 1$  for turbulent flow,  $\alpha = 1/2$  for laminar flow).

The first three terms on the right side in Eq. (4-2) are *point* functions – they depend only on conditions at the inlet and outlet of the system, whereas the  $w$  and  $e_f$  terms are *path* functions, which depend on what is happening to the system between the inlet and the outlet points. These are rate-dependent and can be determined from an appropriate rate or transport model.

The frictional loss term  $e_f$  in Eq. (4-1) represents the loss of mechanical energy due to friction and includes losses due to flow through lengths of pipe; fittings such as elbows, valves, orifices; and pipe entrances and exits. For each frictional device a loss term of the following form is used

$$e_f = K_f \frac{v^2}{2} \quad (4-4)$$

where  $K_f$  is the excess head loss (or loss coefficient) due to the pipe or pipe fitting (dimensionless) and  $v$  is the fluid velocity (length/time).

For fluids flowing through pipes, the excess head loss term  $K_f$  is given by

$$K_f = 4f_F \frac{L}{D} \quad (4-5)$$

where

$f_F$  = Fanning friction factor

$L$  = flow path length (length)

$D$  = flow path diameter (length).

*Note:*

Conversion factor:  $g_c$  [ML/ft<sup>2</sup>],  $F = ma/g_c$

$$\begin{aligned} g_c &= 32.174 \left( \frac{\text{lb}_m \text{ ft}}{\text{lb}_f \text{ s}^2} \right) = 9.806 \left( \frac{\text{kg}_m \text{ m}}{\text{kg}_f \text{ s}^2} \right) = 980.6 \left( \frac{\text{g}_m \text{ cm}}{\text{g}_f \text{ s}^2} \right) \\ &= 1 \left( \frac{\text{lb}_m \text{ ft}}{\text{poundal s}^2} \right) = 1 \left( \frac{\text{kg}_m \text{ m}}{\text{N s}^2} \right) = 1 \left( \frac{\text{g}_m \text{ cm}}{\text{dyn s}^2} \right) \\ &= 1 \left( \frac{\text{slug ft}}{\text{lb}_f \text{ s}^2} \right) \end{aligned} \quad (4-6)$$

In general, pressure loss due to flow is the same whether the pipe is horizontal, vertical, or inclined. The change in pressure due to the difference in head must be considered in the pressure drop calculation.

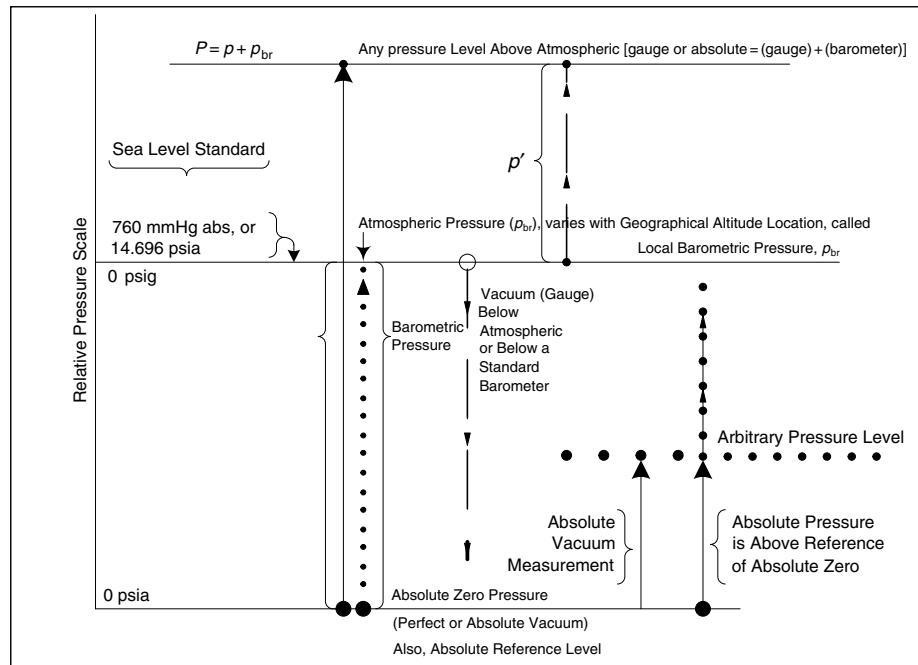
Considerable research has been done on the flow of compressible and non-compressible liquids, gases, vapors, suspensions, slurries, and many other fluid systems to allow definite evaluation of conditions for a variety of process situations for Newtonian fluids. For non-Newtonian fluids, considerable data are available. However, its correlation is not as broad in application, due to the significant influence of physical and rheological properties. This presentation reviews Newtonian systems and to some extent the non-Newtonian systems.

Primary emphasis is given to flow through circular pipes or tubes since this is the usual means of movement of gases and liquids in process plants. Flow through duct systems is treated with the fan section of compression in Volume 3.

### 4.3 SCOPE

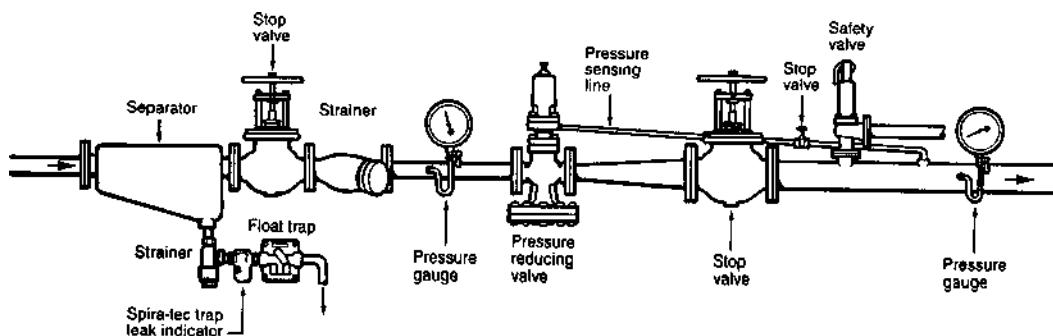
The scope of this chapter emphasizes applied design techniques for 85%  $\pm$  of the usual situations occurring in the design and evaluation of chemical and petrochemical plants for pressure and vacuum systems (Figure 4-3). Whereas computer methods have been developed to handle many of the methods described here, it is the intent of this chapter to present only design methods that may also be applied to computer programming. First, however, a thorough understanding of design methods and their fundamental variations and limitations is critical. There is a real danger in losing sight of the required results of a calculation when the computer program is “hidden” from the user and the user becomes too enamored with the fact that the calculations were made on a computer. A good designer must know the design details built into the computer program before using its results. There are many programs for process design that actually give incorrect results because the programmer was not sufficiently familiar with the design procedures and end limits/limitations of the method. Then, when such programs are purchased by others, or used in-house by others, some serious and erroneous design results can be generated. On the other hand, many design procedures that are complicated and require successive approximation (such as distillation) but are properly programmed can be extremely valuable to the design engineers.

In this book, reference to computer programs is emphasized where necessary, and important mechanical details are given to emphasize the mechanical application of the process requirements (Figure 4-4). Many of these details are essential to the proper functioning of the process in the hardware. For two fundamental aspects of fluid flow, see Figures 4-3 and 4-4 (Moody diagram). In the laminar region or the viscous flow (e.g.,  $Re < 2000$ ), the roughness and thus the relative roughness parameter do not affect the friction factor which is proportional only to the reciprocal of the Reynolds number. The slope of the relationship is such that  $f_{lam} = 64/Re$ , which is plotted in Figure 4-5. In this region, the only fluid property that influences friction loss is the viscosity (because the density cancels out). The “critical zone” is the transition from laminar to turbulent flow, which corresponds to values of  $Re$  from about 2000 to 4000. Data are not reproducible in this range, and correlations are unreliable. The transition region in Figure 4-5 is the region where the friction factor depends strongly on both the Reynolds number and relative roughness. The region in the upper right of the diagram where the lines of constant roughness are horizontal is known as “complete turbulence”, “rough pipes”, or “fully turbulent”. In this region the friction factor is independent of Reynolds number (e.g., independent of viscosity) and is a function only of the relative roughness.

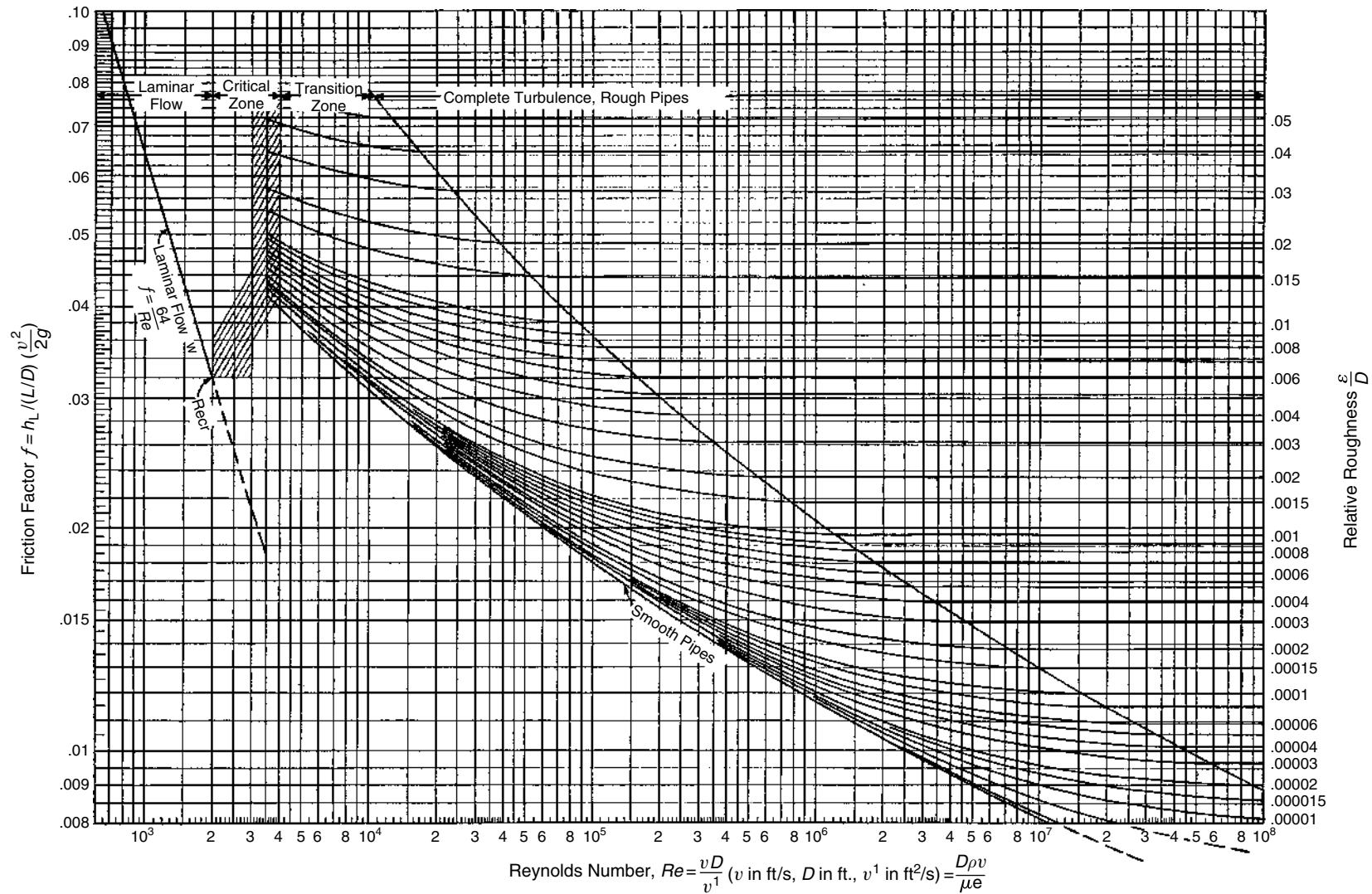


- Notes:**
1. At sea level, barometric pressure = 14.696 lb/in<sup>2</sup>, absolute, or 760 mm of mercury, referred to as "standard". This is also 0 lb/in<sup>2</sup>, gauge for that location.
  2. Absolute zero pressure is absolute vacuum. This is 0 psia, also known as 29.92 in. of mercury below atmospheric pressure, or 33.931 ft of water below atmospheric, all referenced at sea level.
  3. Important equivalents: 1 atm pressure at sea level=
    - 14.696 psia
    - 33.931 ft of water (at 60° F)
    - 29.921 in.Hg (at 32° F)
    - 760 mmHg (at 32° F)
    - 1.0332 kg/cm<sup>2</sup>
    - 10,332.27 kg/m<sup>2</sup>
  4. Barometric pressure for altitudes above "standard" sea level are given in the appendix. These correct values must be used whenever the need for the local absolute barometric pressure is involved in pressure calculations.
  5. Vacuum is expressed as either
    - Inches (or millimeters) vacuum below atmospheric or local barometric, or
    - Inches vacuum absolute, above absolute zero pressure or perfect vacuum.
    - For example, at sea level of 29.92 in.Hg abs barometer; (1) 10 in. vacuum is a gauge term, indicating 10 in. Hg below local barometric pressure; (2) 10 in. vacuum (gauge) is equivalent to 29.921 in.Hg abs – 10 in. = 19.921 in.Hg abs vacuum.

**Figure 4-3** Pressure level references. (Adapted by permission from Crane Co., Engineering Div., Technical Paper No. 410, 1957.)



**Figure 4-4** Portion of a plant piping system. (By permission from Spiral-Sarco, Inc. 1991.)



**Figure 4-5** Moody or “regular” Fanning friction factors for any kind and size of pipe. Note: The friction factor read from this chart is four times the value of the  $f$  factor read from Perry’s Handbook, 6th ed. [1] (Reprinted by permission from pipe Friction Manual, 1954 by the Hydraulic Institute. Also see *Engineering Databook*, 1st ed., The Hydraulic Institute, 1979 [2]. Data from L.F. Moody, “Friction Factors for Pipe Flow” by ASME [3].)

#### 4.4 BASIS

The basis for single-phase and some two-phase friction loss (pressure drop) for fluid flow follows the Darcy and Fanning concepts (e.g., the irreversible dissipation of energy). Pipe loss can be characterized by either the Darcy or Fanning friction factors. The exact transition from laminar or viscous flow to the turbulent condition is variously identified as between a Reynolds number of 2000 and 4000.

For an illustration of a portion of a plant piping system, see Figure 4-4.

#### 4.5 INCOMPRESSIBLE FLOW

The friction loss for laminar or turbulent flow in a pipe [4]

$$\Delta P_f = \frac{\rho f v^2 L}{144 D(2g)}, \text{ lb}_f/\text{in.}^2 \quad (4-7)$$

In SI units,

$$\Delta P_f = \frac{\rho f v^2 L}{2D}, \text{ N/m}^2. \quad (4-8)$$

In terms of feet (meters) of fluid  $h_f$  is given by

$$h_f = \frac{f L v^2}{D(2g)}, \text{ ft (m) of fluid flowing} \quad (4-9)$$

$g$  = acceleration of gravity = 32.2 ft/s<sup>2</sup> (9.81 m/s<sup>2</sup>)

See nomenclature for definition of symbols and units. The units presented are English engineering units and Metric units. The friction factor is the only experimental variable in (that must be determined by reference to) the above equations and it is represented by Figure 4-5. Note that this may sometimes be (referred to as) expressed in terms of the Fanning formula which is (and may be modified to yield a friction factor) one-fourth that of the Darcy factor (e.g.,  $f_D = 4f_F$ ). Also, it is important to note that the Figure 4-5 presented here is the Moody friction chart in terms of the Darcy friction factor, recommended and consistent with the engineering data of the Hydraulic Institute [2] (It is used mostly by Mechanical and Civil engineers, but Chemical engineers use the Fanning,  $f_F$ ).

*Note:* There is confusion in the notation for friction factors. The Darcy friction factor (used mainly by MEs and CEs) is four times the Fanning friction factor (used mainly by ChEs). Either one can be represented on a Moody diagram of  $f$  vs. Reynolds and  $\varepsilon/D$ .

#### 4.6 COMPRESSIBLE FLOW: VAPORS AND GASES [4]

Compressible fluid flow occurs between the two extremes of isothermal and adiabatic conditions. For adiabatic flow the temperature decreases (normally) for decreases in pressure, and the condition is represented by  $P'V^k = \text{constant}$ , which is usually an isentropic condition. Adiabatic flow is often assumed in short and well-insulated pipe, supporting the assumption that no heat is transferred to or from the pipe contents, except for the small heat generated by friction during flow. For isothermal conditions  $P'V = \text{constant}$  temperature, and is the mechanism usually (not always) assumed for most process piping design. This is in reality close to actual conditions for many process and utility service applications.

*Note:* Adiabatic is best for short pipes, and approaches the isothermal equations for long pipes, so adiabatic conditions are most often assumed.

The single-phase friction loss (pressure drop) for these situations in chemical and petrochemical plants is still represented by the Darcy equation with specific limitations [4]:

1. For larger pressure drops in long lines of a mile or greater in length than noted above, use methods presented with the Weymouth, Panhandle Gas formulas, or the simplified compressible flow equation. (Can integrate the compressible form of the Bernoulli equation directly for ideal gas. Must be careful to recognize when the flow is choked).
2. For isothermal non-choked conditions [4]:

$$w_s = \sqrt{\left[ \frac{144 g_c A^2 \rho_1}{\left( f \frac{L}{D} + 2 \log_e \frac{P'_1}{P'_2} \right)} \right] \left[ \frac{(P'_1)^2 - (P'_2)^2}{P'_1} \right]}, \text{ lbm/s} \quad (4-10)$$

where

$A$  = cross-sectional area of pipe or orifice, in.<sup>2</sup>

$g_c$  = dimensional constant = 32.174( $\text{lb}_m/\text{lb}_f$ )(ft/s<sup>2</sup>)

$f$  = Moody friction factor

$L$  = length of pipe, ft

$D$  = internal diameter of pipe, ft

$P'$  = Pressure, lb/in.<sup>2</sup> abs

$\rho$  = fluid density, lb/ft<sup>3</sup>

*Subscripts*

1 = inlet upstream condition

2 = outlet downstream condition.

In SI units,

$$w_s = 316.23 \sqrt{\left[ \frac{A^2 \rho_1}{\left( f \frac{L}{D} + 2 \log_e \frac{P'_1}{P'_2} \right)} \right] \left[ \frac{(P'_1)^2 - (P'_2)^2}{P'_1} \right]}, \text{ kg/s} \quad (4-11)$$

where

$A$  = cross-sectional area of pipe or orifice, m<sup>2</sup>

$f$  = Moody friction factor

$L$  = length of pipe, m

$D$  = internal diameter of pipe, m

$P'$  = Pressure, N/m<sup>2</sup> abs

$\rho$  = fluid density, kg/m<sup>3</sup>

*Subscripts*

1 = inlet upstream condition

2 = outlet downstream condition.

In terms of pipe size (diameter in in./mm),

$$w_s = 0.371 \sqrt{\left[ \frac{d^4 \rho_1}{\left( f \frac{L}{D} + 2 \log_e \frac{P'_1}{P'_2} \right)} \right] \left[ \frac{(P'_1)^2 - (P'_2)^2}{P'_1} \right]}, \text{ lb/s} \quad (4-12)$$

where  $d$  = internal pipe diameter, in.

In SI units,

$$w_s = 0.0002484 \sqrt{\left[ \frac{d^4 \rho_1}{\left( f \frac{L}{D} + 2 \log_e \frac{P'_1}{P'_2} \right)} \right] \left[ \frac{(P'_1)^2 - (P'_2)^2}{P'_1} \right]}, \text{ kg/s}$$
(4-13)

where  $d$  = internal pipe diameter, mm.

The derived equations are believed to apply to good plant design procedures with good engineering accuracy. As a matter of good practice with the exercise of proper judgment, the designer should familiarize himself/herself with the background of the methods presented in order to better select the conditions associated with a specific problem.

*Note:* These equations can be applied to piping including fittings, if the substitution  $fL/D = \sum K_{\text{fittings}}$  is made, in terms of the sum of all loss coefficients for pipe plus fittings. They apply only if the flow is choked).

Design conditions include:

1. Flow rate and pressure drop allowable (net driving force) established, determine pipe size for a fixed length.
2. Flow rate, diameter, and length known, determine (net driving force) pressure drop.

Usually either of these conditions requires a trial approach based upon assumed pipe sizes to meet the stated conditions. Some design problems may require determination of maximum flow for a fixed line size and length.

Optimum economic line size is seldom realized in the average process plant. Unknown factors such as future flow rate allowances, actual pressure drops through certain process equipment, and so on, can easily overbalance any design predicated on selecting the optimum. Certain guides as to order of magnitude of costs and sizes can be established either by one of several correlations or by conventional cost-estimating methods. The latter is usually more realistic for a given set of conditions, since generalized equations often do not fit a plant system (Darby [5]).

There are many computer programs for sizing fluid flow through pipe lines [6]. However, the designer should examine the bases and sources of such programs; otherwise, significant errors could destroy the validity of the program for its intended purpose.

#### 4.7 IMPORTANT PRESSURE LEVEL REFERENCES

Figure 4-3 presents a diagrammatic analysis of the important relationships between absolute pressure, gauge pressures, and vacuum. These are essential to the proper solution of fluid flow, fluid pumping, and compression problems. Most formulas use absolute pressures in calculations; however, there are a few isolated situations where gauge pressures are used. Care must be exercised in following the proper terminology as well as in interpreting the meaning of data and results.

#### 4.8 FACTORS OF "SAFETY" FOR DESIGN BASIS

Unless noted otherwise the methods suggested here do not contain any built-in design factors. These should be included, but only to the extent justified by the problem at hand. Although most designers place this factor on the flow rate, care must be given in analyzing the actual conditions at operating rates below this value. In some situations a large factor imposed at this point

may lead to unacceptable conditions causing erroneous decisions and serious effects on the sizing of automatic control valves internal trim.

As a general guide, factors of safety of 20–30% on the friction factor will accommodate the change in roughness conditions for steel pipe with average service of 5–10 years, but will not necessarily compensate for severe corrosive conditions. Corrosion conditions should dictate the selection of the materials of construction for the system as a part of establishing design criteria. Beyond this, the condition often remains static, but could deteriorate further. This still does not allow for increased pressure drop due to increased flow rates. Such factors are about 10–20% additional (e.g., increasing flow rate by 20% will increase friction loss by 40%). Therefore for many applications the conservative Cameron Tables [7] give good direct-reading results for long-term service (see Table 4-46).

#### 4.9 PIPE, FITTINGS, AND VALVES

To ensure proper understanding of terminology, a brief discussion of the “piping” components of most process systems is appropriate.

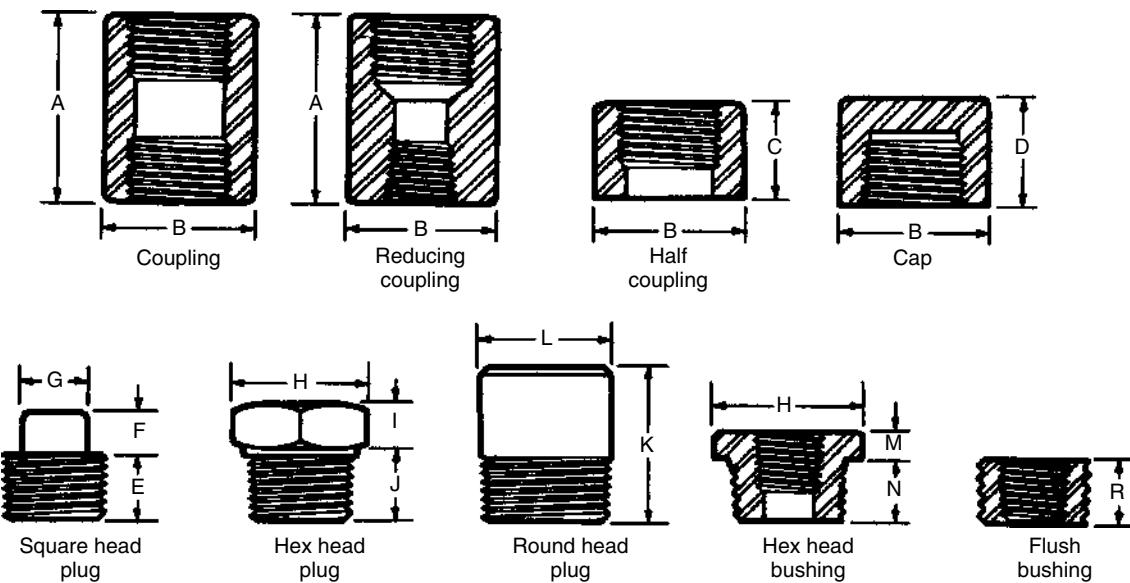
The fluids considered in this chapter consist primarily of liquids, vapors, gases, and slurries. These are transported usually under pressure through circular ducts, tubes, or pipes (except for low pressure air), and these lengths of pipe are connected by fittings (screwed or threaded, butt-welded, socket-welded, or flanged) and the flow is controlled (stopped, started, or throttled) by means of valves fixed in these line systems. The components of these systems will be briefly identified in this chapter, because the calculation methods presented are for flows through these components in a system. These flows always create some degree of friction loss (pressure drop) (or loss of pressure head) which then dictates the power required to move the fluids through the piping components (Figure 4-4). (Pump power may be required for other purposes than just overcoming friction.)

#### 4.10 PIPE

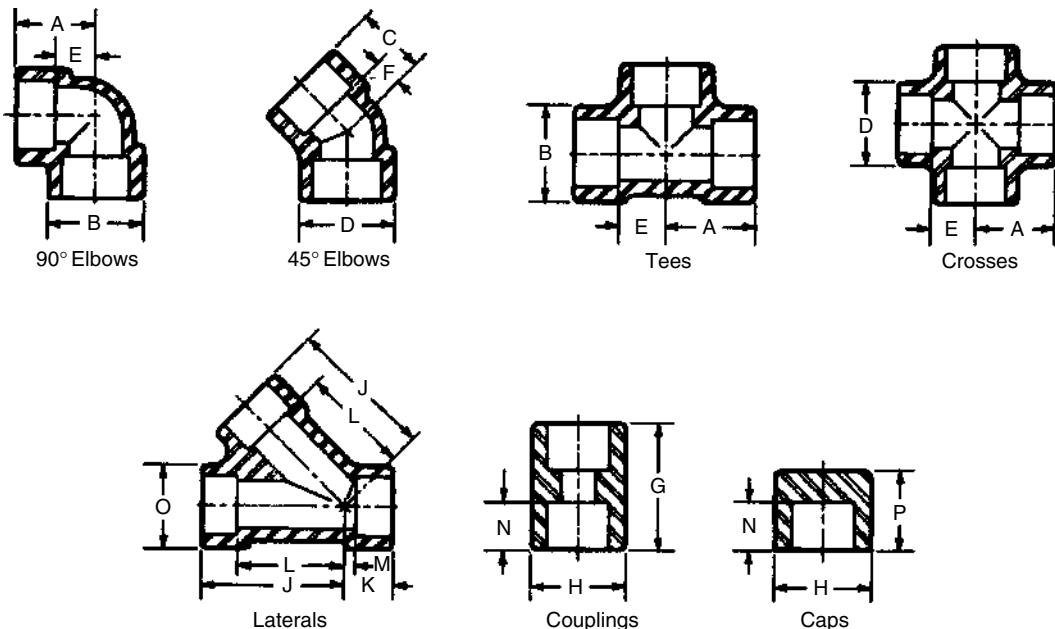
Process plants use round pipe of varying diameters (see pipe dimensions in Tables D-1, D-2, D-3 in Appendix D). Connections for smaller pipe below about 1½ in. (Figures 4-6a and b) are threaded or socket-welded, while nominal pipe sizes 2 in. and larger are generally butt- or socket-welded (Figure 4-6c) with the valves and other connections flanged into the line. Steam power plants are a notable exception. This chapter, however, does not deal with power plant design, although steam lines are included in the sizing techniques. Pipe is generally designated by nominal size, whereas calculations for flow considerations must use the actual standard inside diameter (ID) of the pipe. For example: Note that OD refers to outside diameter of pipe in the table below.

Nominal Pipe Size in Inches	OD Inches		ID Inches	
	Sch. 40	Sch. 80	Sch. 40	Sch. 80
¾	1.050	1.050	0.824	0.742
1	1.315	1.315	1.049	0.957
1½	1.900	1.900	1.610	1.500
2	2.374	2.375	2.067	1.939
3	3.500	3.500	3.068	2.900
4	4.500	4.500	4.026	3.826

See Appendix for other sizes.



**Figure 4-6a** Forged steel threaded pipe fittings, WOG (water, oil, or gas service). *Note:* The working pressures are always well above actual plant operating levels. Pressure classes 3000 psi and 6000 psi, sizes 1/8 in. through 4 in. nominal. (By permission from Ladish Co., Inc.)



**Figure 4-6b** Forged steel socket weld fittings, WOG (Water, oil, or gas service). *Note:* the working pressures are always well above actual plant operating levels and are heavy to allow for welding. Pressure classes 3000 psi and 6000 psi, sizes 1/8 in. through 4 in. nominal. Do not weld on malleable iron or cast iron fittings. (By permission from Ladish Co. Inc.)

American Standards Association piping pressure Classes are given in the table below.

#### 4.11 USUAL INDUSTRY PIPE SIZES AND CLASSES PRACTICE

ASA Pressure Class	Schedule Number of Pipe
$\leq 250 \text{ lb/in.}^2$	40
300–600	80
900	120
1500	160
2500 ( $1\frac{1}{2} \times 6$ in.)	XX (double extra strong)
2500 (8 in. and larger)	160

Certain nominal process and utility pipe sizes are not in common use and hence their availability is limited. Those not usually used are  $1\frac{1}{8}$ ,  $1\frac{1}{4}$ ,  $2\frac{1}{2}$ ,  $3\frac{1}{2}$ , 5, 22, 26, 32, and 34 (in inches).

Some of the larger sizes, 22 in. and higher are used for special situations. Also, some of the non-standard process sizes such as  $2\frac{1}{2}$ ,  $3\frac{1}{2}$ , and 5 in. are used by "packaged" equipment suppliers to connect components in their system for use in processes such as refrigeration, drying, or contacting.

90° ELBOWS Long Radius Pages 12 - 17	ECCENTRIC REDUCERS Pages 63 - 70	PIPELINE and WELDING NECK FLANGES Pages 100 - 115
90° ELBOWS Long Tangent One End Page 16	CAPS Pages 71 - 75	SLIP-ON FLANGES Pages 101 - 115
90° REDUCING ELBOWS Long Radius Pages 18 - 21	LAP JOINT STUB ENDS Pages 76 - 77	LAP JOINT FLANGES Pages 102 - 115
3R ELBOWS 45° and 90° Page 22	LATERALS Straight and Reducing Outlet Page 78	THREADED FLANGES Pages 102 - 115
90° ELBOWS Short Radius Pages 23 - 25	SHAPED NIPPLES Page 79	BLIND FLANGES Pages 102 - 115
45° ELBOWS Long Radius Pages 26 - 30	SLEEVES Page 80	SOCKET TYPE WELDING FLANGES Pages 102 - 105 108 - 109 112 - 113
180° RETURNS Long Radius Pages 31 - 35	SADDLES Page 80	REDUCING FLANGES Pages 102 - 115
180° RETURNS Short Radius Pages 37 - 39	FULL ENCIRCLEMENT SADDLES Page 81	ORIFICE FLANGES Pages 116 - 123
TEES Straight and Reducing Outlet Pages 40 - 57	WELDING RINGS Pages 82 - 83	LARGE DIAMETER FLANGES Pages 130 - 142
CROSSES Straight and Reducing Outlet Pages 58 - 62	HINGED CLOSURES Pages 84 - 87	EXPANDER FLANGES Page 143
CONCENTRIC REDUCERS Pages 63 - 70	T-BOLT CLOSURES Pages 88 - 89	VENTURI EXPANDER FLANGES Page 144

Figure 4-6c Forged steel Welded-end fittings. (By permission from Tube Turn Technologies, Inc.)

The most common Schedule in use is 40, and it is useful for a wide range of pressures defined by ANSI Std. B 36.1 (American National Standards). Lighter wall thickness pipe would be designated Schedules 10, 20, or 30; whereas, heavier wall pipe would be Schedules 60, 80, 100, 120, 140, or 160 (see Appendix Table).

Not all Schedules are in common use, because after Sch. 40, the Sch. 80 is usually sufficient to handle most pressure situations. The process engineer must check this Schedule for both pressure and corrosion to be certain there is sufficient metal wall thickness.

When using alloy pipe with greater tensile strength than carbon steel, the Schedule numbers still apply, but may vary, because it is unnecessary to install thicker-walled alloy pipe than is necessary for the strength and corrosion considerations. Schedules 10 and 20 are rather common for stainless steel pipe in low pressure applications.

For example, for 3 in. nominal carbon steel pipe, the Sch. 40 wall thickness is 0.216 in. If the pressure required in the system needs 0.200 in. wall and the corrosion rate over a 5-year life required 0.125 in. ( $\frac{1}{8}$  in.), then the 0.200 in. + 0.125 in. = 0.325 in. and the Sch. 40 pipe would not be strong enough at the end of 5 years. Often the corrosion is calculated for 10 or 15 years' life before replacement. Currently Sch. 80, 3 in. pipe has a 0.300 in. wall thickness, even this is not good enough in carbon steel. Rather than use the much heavier Sch. 160, the designer should reconsider the materials of construction as well as re-examine the corrosion data to be certain there is not unreasonable conservatism. Perhaps stainless steel pipe or a "lined" pipe would give adequate strength and corrosion resistance. For a bad corrosion condition, lined pipe using linings of PVC (polyvinyl chloride), Teflon®, or Saran® typically as shown in Figures 4-7a-d can be helpful.

While threaded pipe is joined by threaded fittings (Figure 4-6a), the joints of welded pipe are connected to each other by butt or socket welding (Figure 4-6b) and to valves by socket welds or flanges of several types (Figure 4-8) using a gasket of composition material, rubber, or metal at the joint to seal against leaks. The joint is pulled tight by bolts (Figure 4-9).

For lower pressure systems of approximately 150 psig at 400° F or 225 psig at 100° F, and where sanitary precautions (food products or chemicals used in food products) or some corrosion resistance is necessary, tubing is used. It is joined together by butt welds (Figure 4-10) or special compression or hub-type end connectors. This style of "piping" is not too common in the chemical/petrochemical industries, except for instrument lines (sensing, signal transmission) or high pressures above 2000 psig.

Figure 4-11 compares the measurement differences for tubes (outside diameter) and iron or steel pipe size (IPS), nominal inside diameter. One example of dimensional comparison for IPS pipe for Schs 5 and 10 are compared to one standard scale of tubing in Table 4-1. The tubing conforms to ANSI/ASTM A-403-78 Class CR (stainless) or MSS Manufacturers Standard Society SP-43, Sch. 5S.

### TOTAL LINE PRESSURE DROP

The total piping system friction loss (pressure drop) for a particular pipe installation is the sum of the friction loss (drop) in pipe, valves, and fittings, plus other pressure losses (drops) through control valves, plus drop through equipment in the system, plus static drop due to elevation or pressure level. For example, see Figure 4-4. This total pressure loss is not necessarily required in determining the frictional losses in the system (*Note: Pressure loss is not the same as pressure change*). It is necessary when establishing gravity flow or the pumping head requirements for a complete system.

Design practice breaks the overall problem into small component parts which allow for simple analysis and solution. This is the recommended approach for selection and sizing of process piping.

### **4.12 BACKGROUND INFORMATION (ALSO SEE CHAPTER 5)**

Gas or vapor density following perfect gas law:

$$\rho = 144 P'(T) \left( \frac{1544}{\text{MW}} \right), \text{ lb/ft}^3 \quad (4-14)$$

Gas or vapor specific gravity referred to air:

$$S_g = \frac{\text{MW of gas}}{\text{MW of air}} = \frac{\text{MW of gas}}{29} \quad (4-15)$$

(Assumes gas and air are ideal. Specific gravity for gases is usually referred to the density of air at standard conditions.)

Conversion between fluid head loss in feet (meter) and (pressure drop) friction loss in psi (bar), any fluid (with specified units for  $\rho$  and standard value of  $g$ ):

$$\text{Friction loss (pressure drop), lb/in.}^2, \Delta P_f = \frac{h_L \rho}{144} \quad (4-16)$$

$$\text{Friction loss (pressure drop, bar), } \Delta P_f = \frac{h_L \rho}{10,200} \quad (4-17)$$

$$\text{For water, psi, } \Delta P_f = \frac{h_L (\text{ft}) \text{ sp gr}}{2.31 (\text{ft/psi})} \quad (4-18)$$

Equivalent diameter and hydraulic radius for non-circular flow ducts or pipes

$$R_H = \text{hydraulic radius, ft}$$

$$R_H = \frac{\text{cross-sectional flow area for fluid flow}}{\text{wetted perimeter for fluid flow}} \quad (4-19)$$

$$D_H = \text{hydraulic diameter (equivalent diameter), ft}$$

$$D_H = 4R_H, \text{ ft} \quad (4-20)$$

$$d_H = \text{hydraulic diameter (equivalent diameter), in.}$$

$$d_H = 48R_H, \text{ in.} \quad (4-21)$$

$$d_H = \frac{4(\text{cross-sectional area for fluid flow})}{(\text{wetted perimeter for fluid flow})} \quad (4-22)$$

For the narrow shapes with small depth relative to width (length), the hydraulic radius is approximately [4]:

$$R_H = \frac{1}{2}(\text{depth of passage}) \quad (4-23)$$

For those non-standard or full circular configurations of flow, use  $d$  equivalent to actual flow area diameter, and  $D$  equivalent to  $4R_H$ .

$$d = 4 \left( \frac{\text{cross-sectional area available for fluid flow, of duct}}{\text{wetted perimeter of duct}} \right) \quad (4-24)$$

This also applies to circular pipes or ducts and oval and rectangular ducts not flowing full. The equivalent diameter is used in determining the Reynolds number for these cases.

*Minimum size of pipe* is sometimes dictated by structural considerations, that is,  $1\frac{1}{2}$ -in. Sch. 40 steel pipe is considered the smallest size to span a 15 ft–20 ft pipe rack without intermediate support.

Gravity flow lines are often set at  $1\frac{1}{4}$ –2 in. minimum, disregarding any smaller calculated size as a potential source of trouble.

Pump suction lines are designed for about a one foot/second velocity, unless a higher velocity is necessary to keep small solids or precipitates in suspension. Suction line sizes should be larger

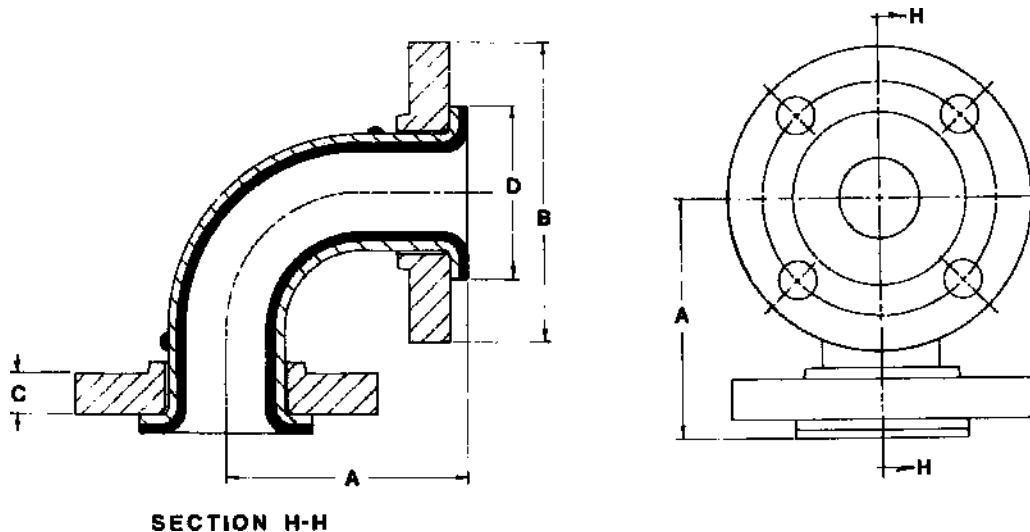


Figure 4-7a Lined-steel pipe and fittings for corrosive service. (By permission from Performance Plastics Products.)

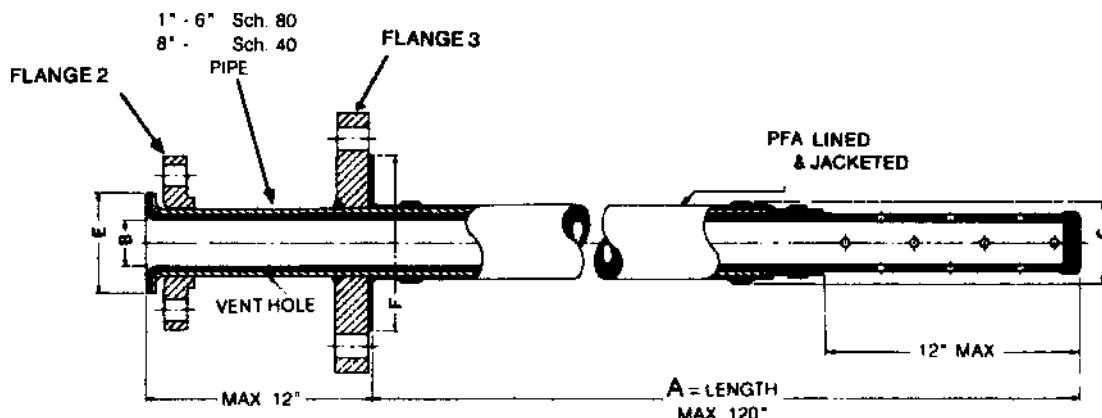
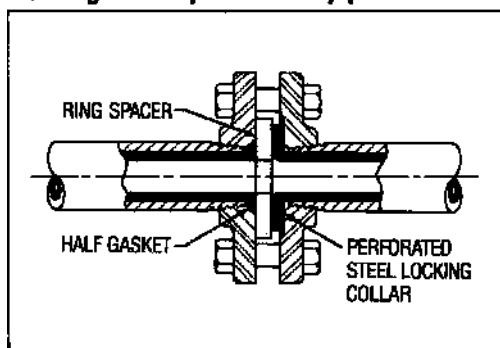
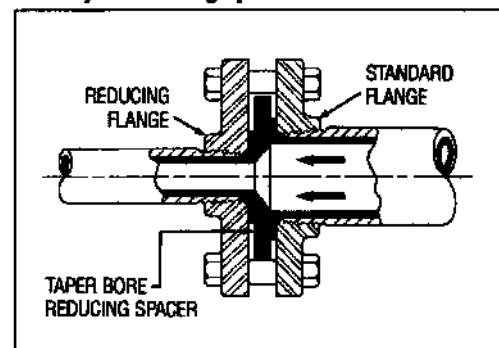


Figure 4-7b Lined-steel pipe flanged sparger for corrosive service. (By permission from Performance Plastics Products.)

**Connection of reinforced flared face to gasketed plastic-lined pipe**

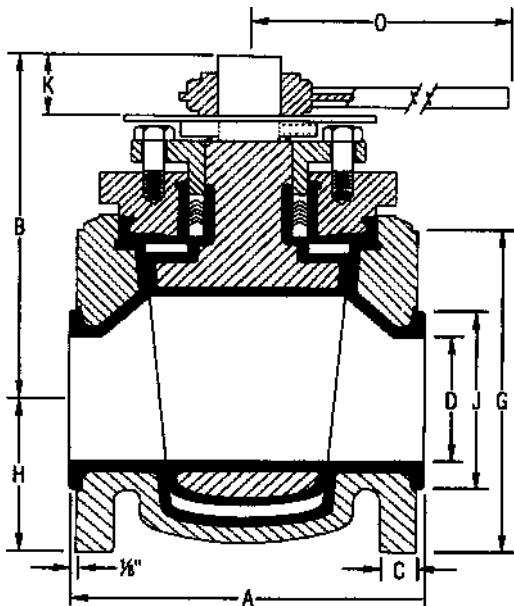


**With taper reducing spacer<sup>2</sup>**



<sup>2</sup> Only the following size reductions should be made by this technique when connecting pipe with molded raised faces:  $1\frac{1}{2} \times 1$ ,  $2 \times 1$ ,  $2 \times 1\frac{1}{4}$ ,  $2\frac{1}{2} \times 1\frac{1}{4}$ ,  $2\frac{1}{2} \times 2$ ,  $3 \times 2$ ,  $3 \times 2\frac{1}{2}$ ,  $4 \times 2\frac{1}{2}$ ,  $4 \times 3$ ,  $6 \times 4$ ,  $8 \times 6$ . All other reductions require use of reducing filler flanges or concentric reducers.

Figure 4-7c Flanged line-steel pipe fittings for corrosive service. (By permission from Dow Plastic-Lined Products, Bay City, Mich. 48707, 1-800-233-7577.)



**Figure 4-7d** Lined plug valve for corrosive service. (By permission from Dow Plastic-Lined Products, Bay City, Mich. 48707, 1-800-233-7577.)

than discharge sizes to minimize the friction loss to reach a realistic value of available Net Positive Suction Head ( $NPSH_A$ ) for the given system.

As a general guide, for pipe sizes use:

threaded pipe – up to and including  $1\frac{1}{2}$  in. or 2 in. nominal  
welded or flanged pipe – 2 in. and larger.

Situations may dictate deviations. Never use cast iron fittings or pipe in process situations unless there is only gravity pressure head (or not over 10 psig) or the fluid is non-hazardous. One exception is in some concentrated sulfuric acid applications, where extreme caution must be exercised in the design of the safety of the system area. Never use in pulsing or shock service. Never use malleable iron fittings or pipe unless the fluid is non-hazardous and the pressure not greater than 25 psig. Always use a pressure rating at least four times that of the maximum system pressure. Also, never use cast iron or malleable iron fittings or valves in pressure-pulsating systems or systems subject to physical shock.

Use forged steel fittings for process applications as long as the fluid does not create a serious corrosion problem. These fittings are attached to steel pipe and/or each other by threading, socket welding, or direct welding to steel pipe. For couplings attached by welding to pipe, Figure 4-6b, use either 2000 psi or 6000 psi rating to give adequate area for welding without distortion, even though the process system may be significantly lower (even atmospheric). Branch connections are often attached to steel pipe using Weldolets® or Threadolets® (Figure 4-12).

Note: ® = Registered Bonney Forge, Allentown, PA.

Mean pressure in a gas line [58].

$$P \text{ (mean or average)} = \frac{2}{3} \left[ (P_1 + P_2) - \frac{P_1 P_2}{P_1 + P_2} \right] \quad (4-25)$$

This applies particularly to long flow lines. The usual economic range for pressure loss due to liquid flow: (a) Suction piping  $\frac{1}{2}$  to  $1\frac{1}{4}$  psi per 100 equivalent feet of pipe; (b) Discharge piping

1 to 5 psi per 100 equivalent feet of pipe. The Appendix presents useful carbon steel and stainless steel pipe data.

#### 4.13 REYNOLDS NUMBER, $Re$ (SOMETIMES USED $N_{Re}$ )

This is the basis for establishing the condition or type of fluid flow (flow regime) in a pipe. Reynolds numbers below 2000–2100 correspond to (are usually considered to define) laminar or viscous flow; numbers from 2000 to 3000–4000 correspond to (define) a transition region of peculiar flow, and numbers above 4000 correspond to (define a state of) turbulent flow. Reference to Figure 4-5 and Figure 4-13 will identify these regions, and the friction factors associated with them [2].

$$Re = \frac{Dv\rho}{\mu_e} = 123.9 \frac{dv\rho}{\mu} = 6.31 \frac{W}{d\mu} \quad (4-26)$$

where

$d$  = pipe internal diameter, in.

$D$  = pipe internal diameter, ft

$v$  = mean fluid velocity, ft/s

$W$  = fluid flow rate, lb/h

$\rho$  = fluid density, lb/ft<sup>3</sup>

$\mu_e$  = absolute viscosity, lb<sub>m</sub>/ft s

$\mu$  = absolute viscosity, cP.

$$Re = 22,700 \frac{qp}{d\mu} = 50.6 \frac{Q\rho}{d\mu} = 0.482 \frac{q'_h S_g}{d\mu} \quad (4-27)$$

where

$q$  = fluid flow rate, ft<sup>3</sup>/s

$q'_h$  = fluid flow rate, ft<sup>3</sup>/h

$Q$  = fluid flow rate, gpm

$S_g$  = specific gravity of a gas relative to air.

In SI units,

$$Re = \frac{Dv\rho}{\mu'} = \frac{dv\rho}{\mu} = 354 \frac{W}{d\mu} \quad (4-28)$$

$d$  = pipe internal diameter, mm

$D$  = pipe internal diameter, m

$v$  = mean fluid velocity, m/s

$W$  = fluid flow rate, kg/h

$\rho$  = fluid density, kg/m<sup>3</sup>

$\mu'$  = absolute viscosity, Pa s

$\mu$  = absolute viscosity, cP

$1\text{ cP} = 10^{-3}\text{ Pa s}$

$1\text{ Pas} = 1\text{ Ns/m}^2 = 1\text{ kg/m s} = 10^3\text{ cP}$ .

$$Re = 1,273,000 \frac{qp}{d\mu} = 21.22 \frac{Q\rho}{d\mu} = 432 \frac{q'_h S_g}{d\mu} \quad (4-29)$$

where

$q$  = fluid flow rate, m<sup>3</sup>/s

$q'_h$  = fluid flow rate, m<sup>3</sup>/h

$Q$  = fluid flow rate, l/min

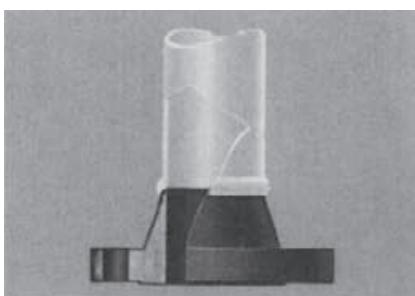
$S_g$  = specific gravity of a gas relative to air.

The mean velocity of flowing liquid can be determined by the following:

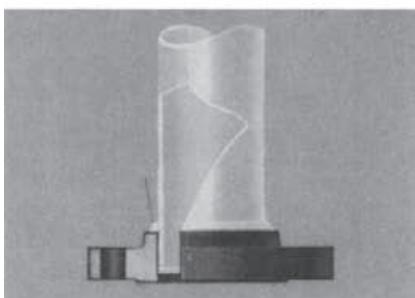
In English Engineering units,

$$v = 183.3 q/d^2 = 0.408 Q/d^2 = 0.0509 W/d^2 \rho, \text{ ft/s} \quad (4-30)$$

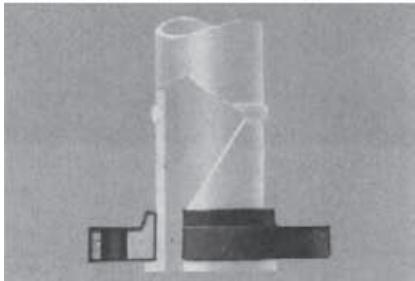
**Welding neck flanges** are distinguished from other types by their long tapered hub and gentle transition of thickness in the region of the butt weld joining them to the pipe. Thus this type of flange is preferred for every severe service condition, whether this results from high pressure or from sub-zero or elevated temperature, and whether loading conditions are substantially constant or fluctuate between wide limits.



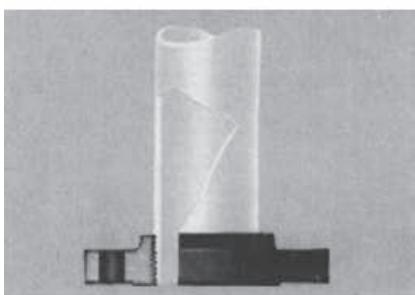
**Slip-on flanges** continue to be preferred to welding neck flanges by many users on account of their initially lower cost, the reduced accuracy required in cutting the pipe to length, and the somewhat greater ease of alignment of the assembly; however, their final installed cost is probably not much, if any, less than that of welding neck flanges. Their calculated strength under internal pressure is of the order of two-thirds that of welding neck flanges, and their life under fatigue is about one-third that of the latter.



**Lap joint flanges** are primarily employed with lap joint stubs, the combined initial cost of the two items being approximately one-third higher than that of comparable welding neck flanges. Their pressure-holding ability is little, if any, better than that of slip-on flanges and the fatigue life of the assembly is only one-tenth that of welding neck flanges. The chief use of lap joint flanges in carbon or low alloy steel piping systems is in services necessitating frequent dismantling for inspection and cleaning and where the ability to swivel flanges and to align bolt holes materially simplifies the erection of large diameter or unusually stiff piping. Their use at points where severe bending stress occurs should be avoided.



**Threaded flanges** made of steel, are confined to special applications. Their chief merit lies in the fact that they can be assembled without welding; this explains their use in extremely high pressure services, particularly at or near atmospheric temperature, where alloy steel is essential for strength and where the necessary post-weld heat treatment is impractical. Threaded flanges are unsuited for conditions involving temperature or bending stresses of any magnitude, particularly under cyclic conditions, where leakage through the threads may occur in relatively few cycles of heating or stress; seal welding is sometimes employed to overcome this, but cannot be considered as entirely satisfactory.



**Socket welding flanges** were initially developed for use on small-size high pressure piping. Their initial cost is about 10% greater than that of slip-on flanges; when provided with an internal weld as illustrated, their static strength is equal to, but their fatigue strength 50% greater than double-welded slip-on flanges. Smooth, pocketless bore conditions can readily be attained (by grinding the internal weld) without having to bevel the flange face and, after welding, to reface the flange as would be required with slip-on flanges.

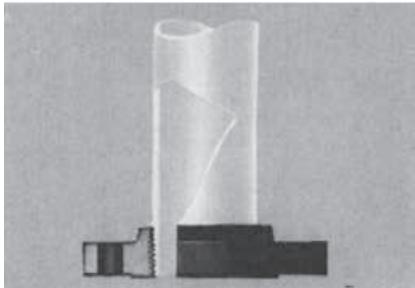
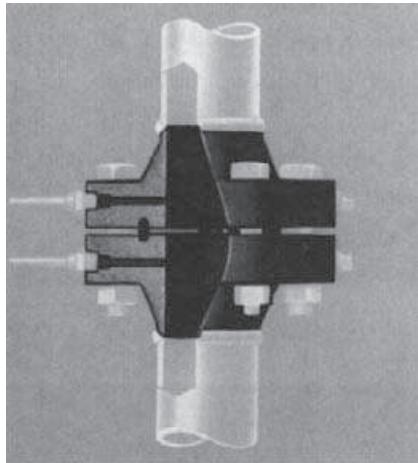
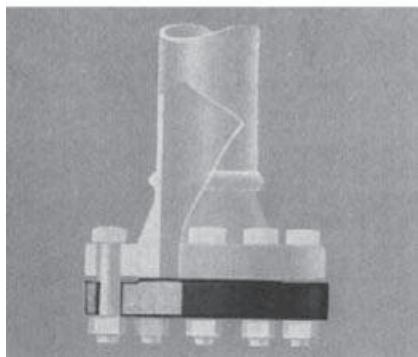


Figure 4-8 Forged steel companion flanges to be attached to steel pipe by the methods indicated. (By permission from Tube Turn Technologies, Inc.)



**Orifice flanges** are widely used in conjunction with orifice meters for measuring the rate of flow of liquids and gasses. They are basically the same as standard welding neck, slip-on and screwed flanges except for the provision of radial, tapped holes in the flange ring for meter connections and additional bolts to act as jack screws to facilitate separating the flanges for inspection or replacement of the orifice plate.



**Blind flanges** are used to blank off the ends of piping, valves and pressure vessel openings. From the standpoint of internal pressure and bolt loading, blind flanges, particularly in the larger sizes, are the most highly stressed of all American Standard flange types; however, since the maximum stresses in a blind flange are bending stresses at the center, they can safely be permitted to be higher than in other types of flanges.

1. In Tube Turns tests of all types of flanged assemblies, fatigue failure invariably occurred in the pipe or in an unusually weak weld, never in the flange proper. The type of flange, however, and particularly the method of attachment, greatly influence the number of cycles required to cause fracture.
2. ANSI B16.5-1961—Steel Pipe Flanges and Flanged Fittings.
3. ASME Boiler and Pressure Vessel Code 1966, Section I, Par. P-300.

Figure 4-8—(continued)

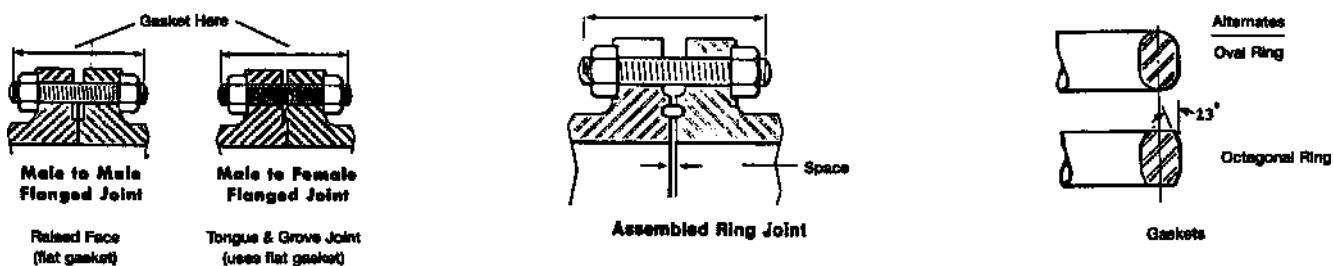


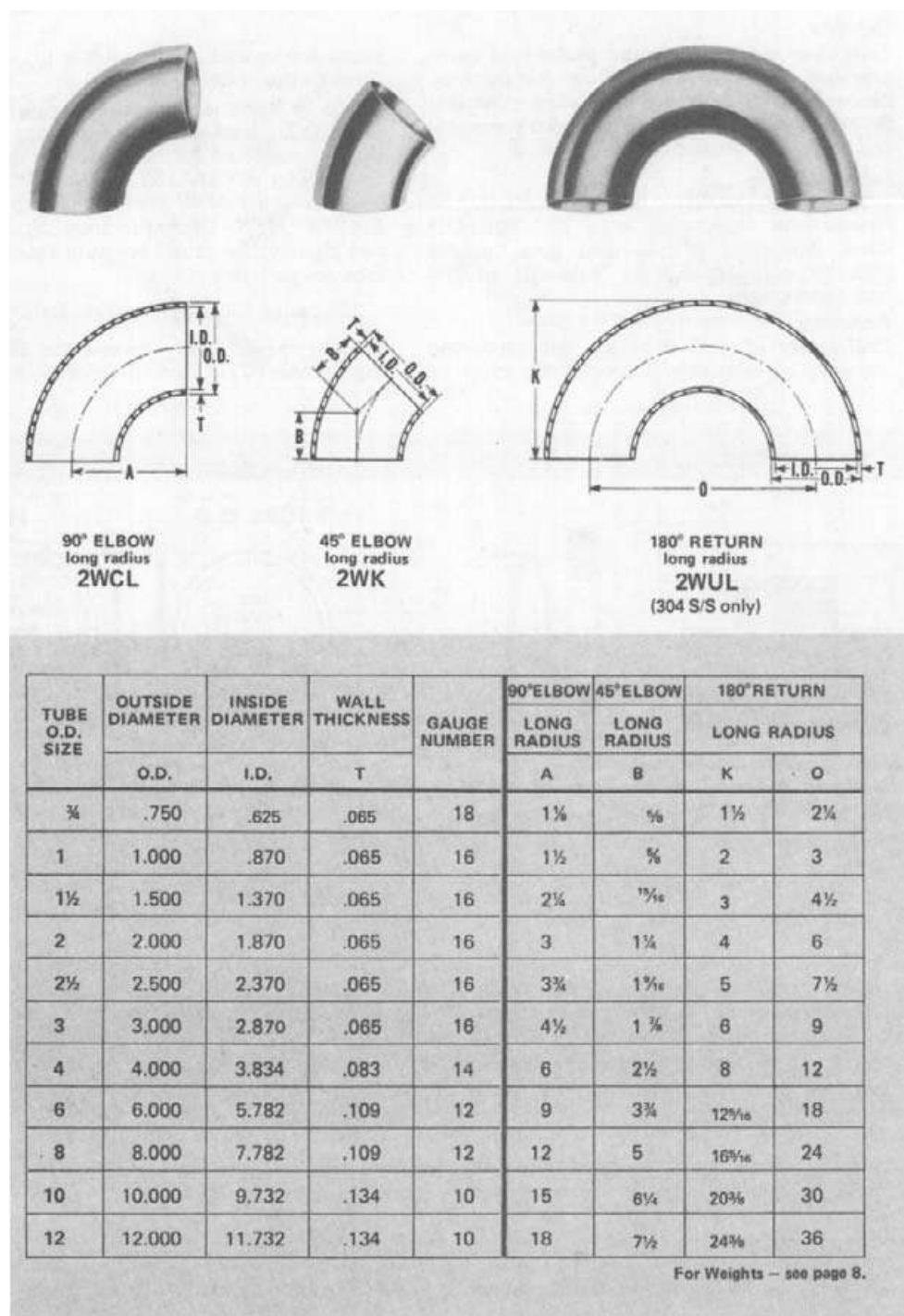
Figure 4-9 Most common flange connection joints. Cross section of a pair of flanges with bolts to draw joint tight.

In SI units,

$$\begin{aligned} v &= 1273.2 \times 10^3 q/d^2 = 21.22 Q/d^2 \\ &= 353.7 W/d^2 \rho, \text{ m/s} \end{aligned} \quad (4-31)$$

Table 4-2 gives a quick summary of various ways in which the Reynolds number can be expressed. The symbols in Table 4-2,

in the order of their appearance are  $D$  = inside diameter of pipe, ft;  $v$  = liquid velocity, ft/s;  $\rho$  = liquid density, lb/ft<sup>3</sup>;  $\mu$  = absolute viscosity of liquid, lb<sub>m</sub>/fts;  $d$  = inside diameter of pipe, in.;  $k$  =  $z/S$  liquid flow rate, lb/h;  $B$  = liquid flow rate, bbl/h;  $k$  = kinematic viscosity of the liquid, Cst;  $q$  = liquid flow rate, ft<sup>3</sup>/s;  $Q$  = liquid flow rate, ft<sup>3</sup>/min. Use Table 4-2 to find the Reynolds number of any liquid flowing through a pipe.



**Figure 4-10** Light weight stainless steel butt-weld fittings/tubing for low pressure applications. (By permission from Tri-Clover, Inc.)

#### 4.14 PIPE RELATIVE ROUGHNESS

Pipe internal roughness reflects the results of pipe manufacture or process corrosion, or both. In designing a flow system, recognition must be given to (a) the initial internal pipe condition as well as (b) the expected condition after some reasonable life period,

such as 10, 15, or 20 years in service. Usually a 10–15-year life period is a reasonable expectation. It is not wise to expect smooth internal conditions over an extended life, even for water, air, or oil flow because some actual changes can occur in the internal surface condition. Some fluids are much worse in this regard than others. New, clean steel pipe can be adjusted from

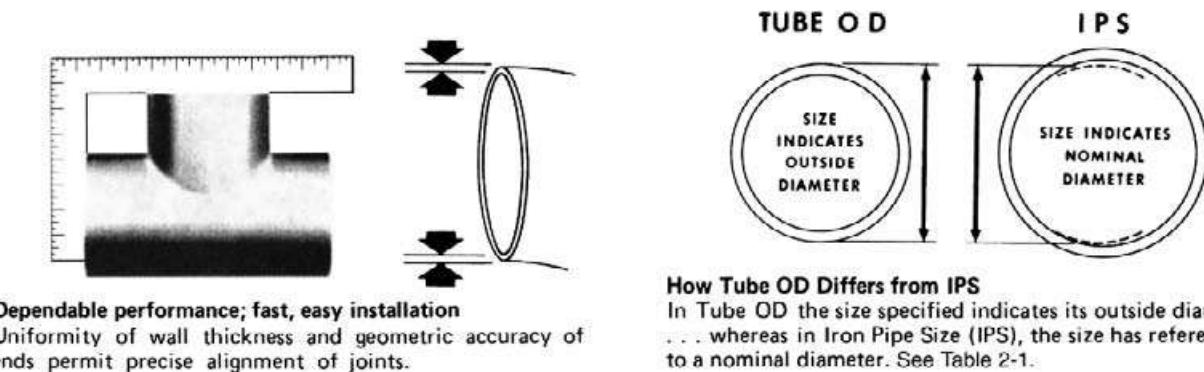


Figure 4-11 Dimension comparison of tubing and IPS (iron pipe size) steel piping. (By permission from Tri-Clover, Inc.)

**TABLE 4-1 Comparison of Dimensions and Flow Area for Tubing and Iron Pipe Size (IPS) Steel Pipe**

OD Tubing					IPS Pipe				
Od Tubing Size	Outside Diameter	Inside Diameter	Flow Area (in. <sup>2</sup> )	IPS Pipe Size	Schedule 5S		Schedule 10S		
					Outside Diameter	Inside Diameter	Flow Area (in. <sup>2</sup> )	Inside Diameter	Flow Area (in. <sup>2</sup> )
3/4	0.750	0.625	0.307	3/8	1.050	0.920	0.665	0.884	0.614
1	1.000	0.870	0.595	1	1.315	1.185	1.10	1.097	0.945
1 1/2	1.500	1.370	1.47	1 1/2	1.900	1.770	2.46	1.682	2.22
2	2.000	1.870	2.75	2	2.375	2.245	3.96	2.157	3.65
2 1/2	2.500	2.370	4.41	2 1/2	2.875	2.709	5.76	2.635	5.45
3	3.000	2.843	6.31	3	3.500	3.334	8.73	3.260	8.35
3 1/2	—	—	—	3 1/2	4.000	3.834	11.55	3.760	11.10
4	4.000	3.834	11.55	4	4.500	4.334	14.75	4.260	14.25
6	6.000	5.782	26.26	6	6.625	6.407	32.24	6.357	31.75
8	8.000	7.782	47.56	8	8.625	8.407	55.5	8.329	54.5
10	10.000	9.732	74.4	10	10.750	10.482	86.3	10.420	85.3
12	12.000	11.732	108	12	12.750	12.438	121.0	12.390	120.0

(Source: By permission from Tric-Clover, Inc.)

the initial clean condition to some situation allowing for the additional roughness. The design-roughened condition can be interpolated from Figure 4-13 to achieve a somewhat more roughened condition, with the corresponding relative roughness  $\varepsilon/D$  value. Table 4-3 shows the wall roughness of some clean, new pipe materials.

Note that the  $\varepsilon/D$  factor from Figure 4-13 is used directly in Figure 4-5. As an example that is only applicable in the range of the charts used, a 10% increase in  $\varepsilon/D$  to account for increased roughness yields from Figure 4-5 an  $f$  of only 1.2% greater than a clean, new commercial condition pipe (*Note:* This number depends on where you are on the Moody diagram. Figure 4-13 applies only for fully turbulent flow to very high Reynolds numbers). Generally the accuracy of reading the charts does not account for large fluctuations in  $f$  values. Of course,  $f$  can be calculated as discussed earlier, and a more precise number can be achieved, but this may not mean a significantly greater accuracy of the calculated pressure drop. Generally, for industrial process design, experience should be used where available in adjusting the roughness and effects on the friction factor. Some designers increase the friction factor by 10–15% over standard commercial pipe values.

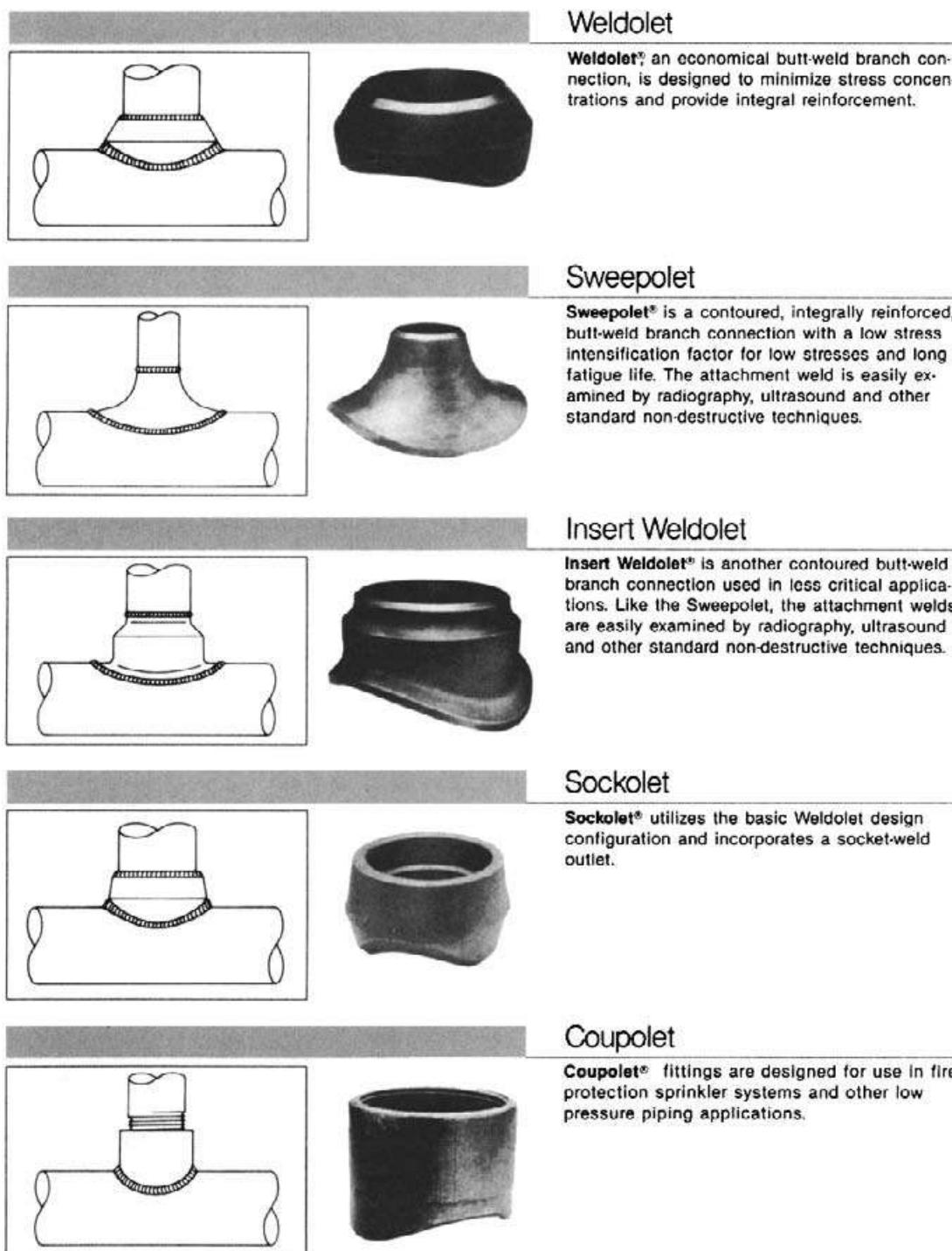
#### 4.15 DARCY FRICTION FACTOR, $F$

For laminar or viscous flow,

$$f = \frac{64}{Re} \quad \text{For } Re < 2000 \quad (4-32)$$

For transition and turbulent flow, use Figure 4-13 (the  $f$  in this figure only applies for fully turbulent flow corresponding to the flat portions of the curves in Figure 4-5) with Figure 4-5, and Figures 4-14a and 4-14b as appropriate. Friction factor in long steel pipes handling wet (saturated with water vapor) gases such as hydrogen, carbon monoxide, carbon dioxide, nitrogen, oxygen, and similar materials should be considered carefully, and often increased by a factor of 1.2–2.0 to account for corrosion.

*Important Note:* The Moody [3] friction factors ( $f_D$ ) reproduced in this text (Figure 4-5) are consistent with the published values of references [2–4], and cannot be used with the values presented in Perry's Handbook [1] (e.g., the Fanning friction factor,  $f_F$ ), as Perry's values for  $f_F$  are one-fourth times the values cited in this chapter (e.g.,  $f_F = \frac{1}{4}f_D$ ). It is essential to use  $f$  values with the corresponding formulas offered in the appropriate text.



**Figure 4-12** Branch connections for welding openings into steel pipe. See Figure 4-6c for alternate welding fittings. (By permission from Bonney Forge Corp., Allentown, PA.)

The Colebrook equation [9] is considered a reliable approach to determining the friction factor,  $f_D$  (Moody factor).

*Note:* The turbulent portion of the published Moody diagram is actually a plot of this equation, which is derived to fit “sand roughness” data in pipes [10].

$$\frac{1}{\sqrt{f_D}} = -2 \log_{10} \left\{ \frac{\varepsilon}{3.7D} + \frac{2.51}{Re\sqrt{f_D}} \right\} \quad \text{For } Re > 4000 \quad (4-33)$$

$$Re = \frac{vD}{v'}$$

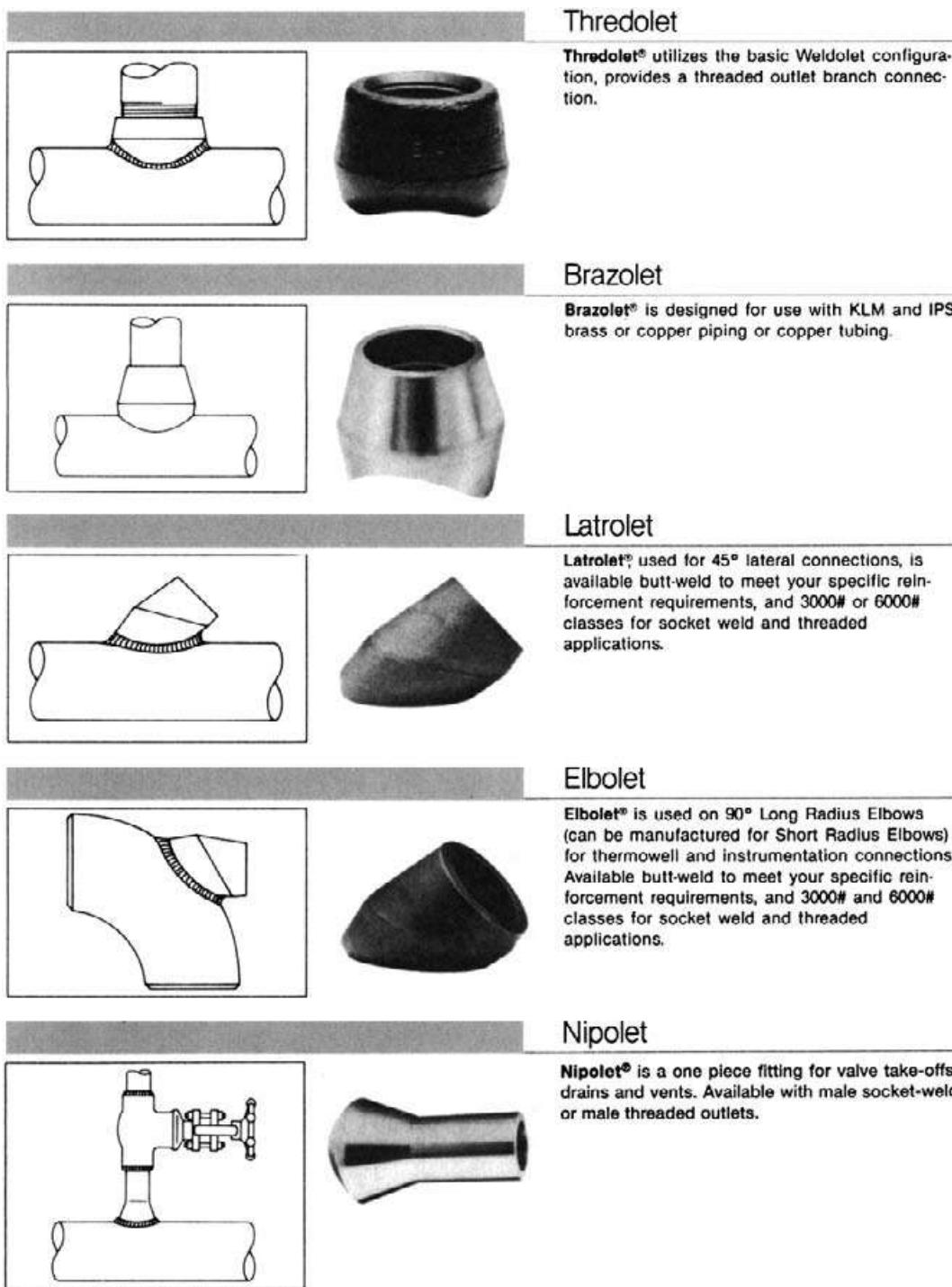


Figure 4-12—(continued)

where

$$\nu' = \text{kinematic viscosity} = \frac{\text{viscosity}}{\text{density}} = \frac{\mu}{\rho}$$

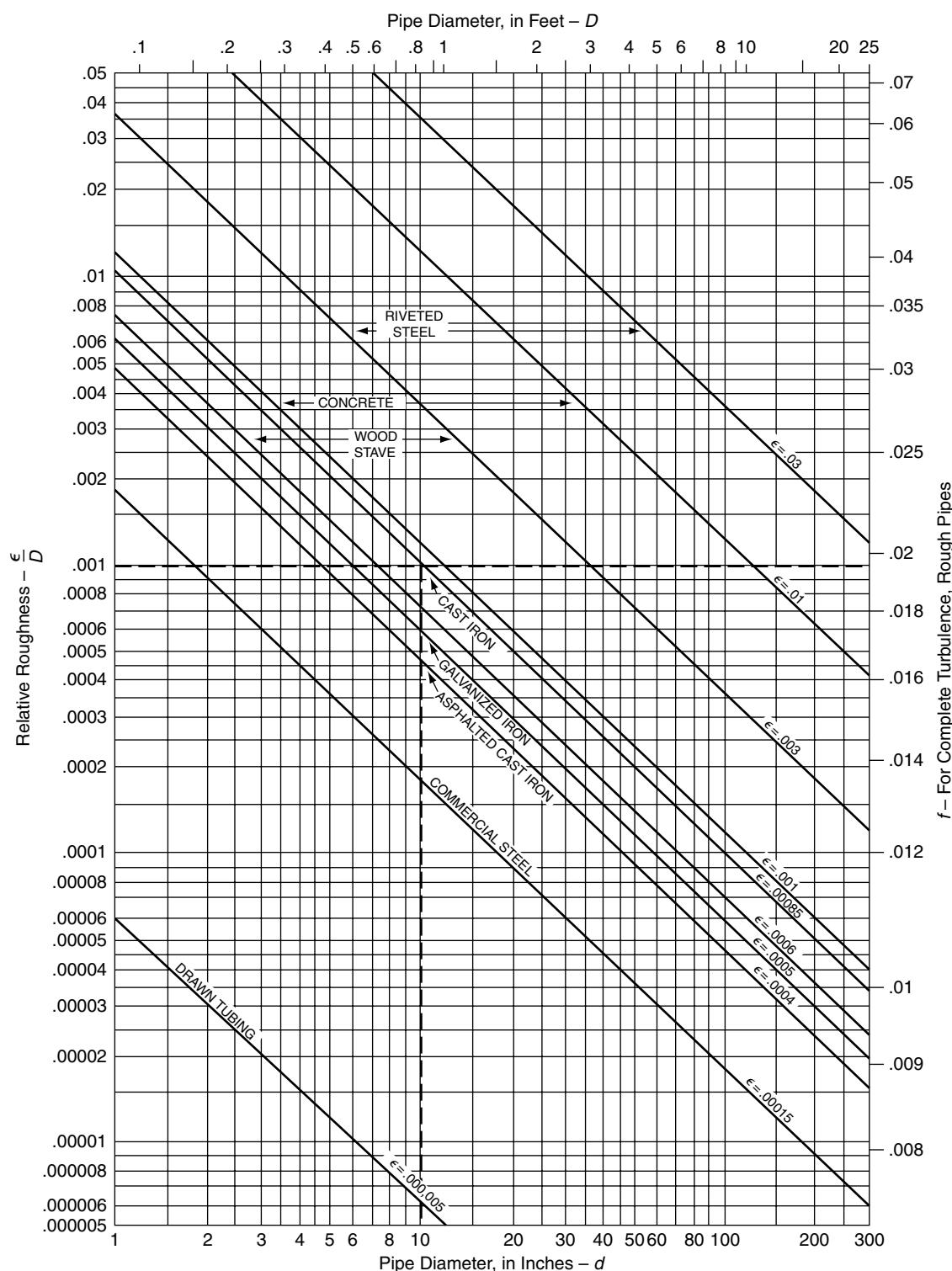
$D$  = Pipe internal diameter

$v$  = Liquid velocity.

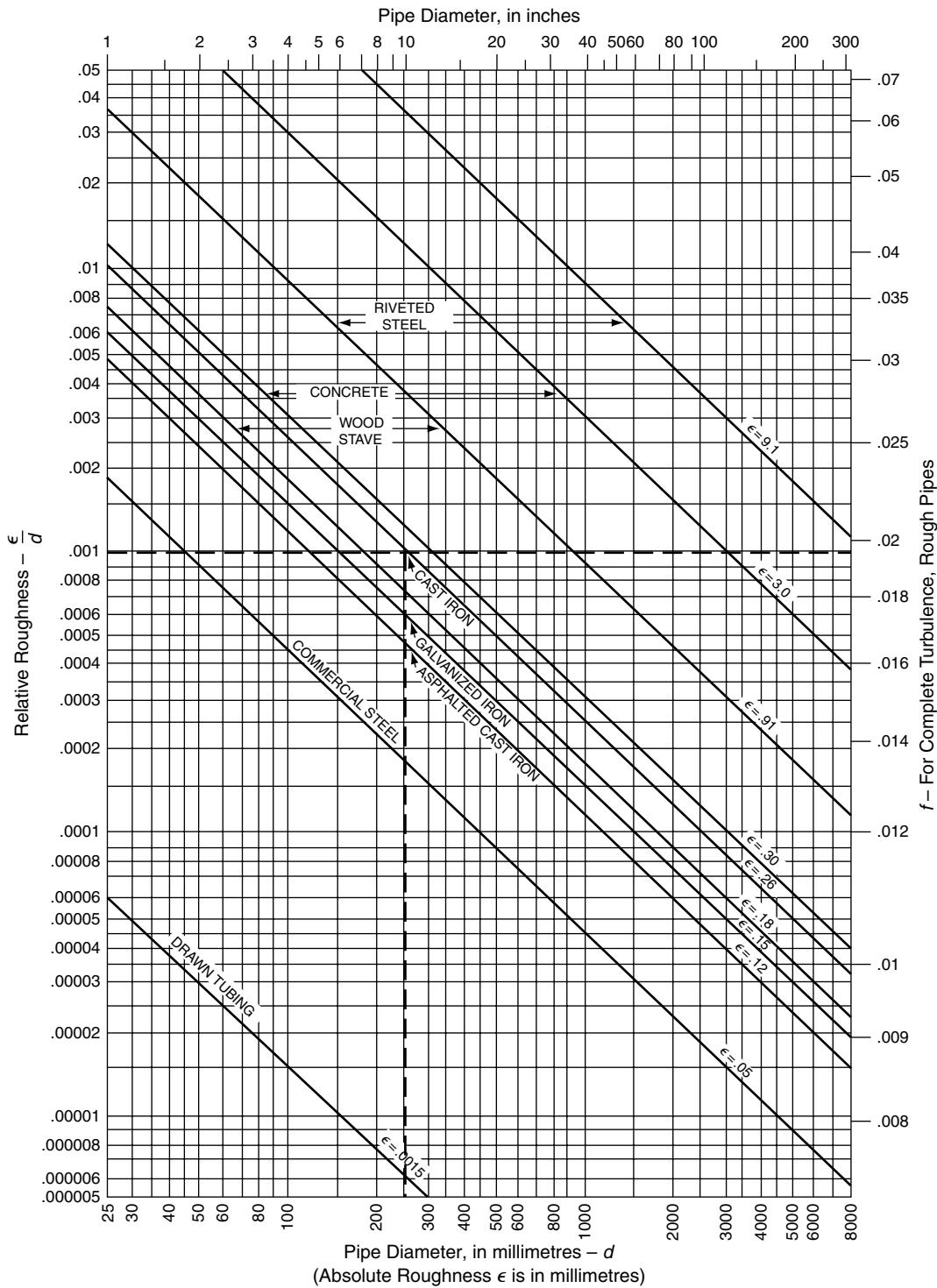
Note that the term  $\varepsilon/D$  is the relative roughness from Figure 4-13. Equation (4-33) is implicit in  $f_D$ , as it cannot be rearranged to

derive  $f_D$  directly and thus requires an iterative solution (e.g., trial and error). Colebrook [9] also proposed a direct solution equation that is reported [11] to have

$$f = 1.8 \log_{10} \left( \frac{Re}{7} \right)^{-2} \quad (4-34)$$



**Figure 4-13a** Relative roughness of pipe materials and friction factors for complete turbulence. (Reprinted/Adapted with permission from "Flow of Fluids Through Valves, Fittings, and Pipe", Technical Paper No. 410, Crane Co. All rights reserved.)



**Figure 4-13b** Relative roughness of pipe materials and friction factors for complete turbulence. (Reprinted/Adapted with permission from "Flow of Fluids Through Valves, Fittings, and Pipe", Technical Paper No. 410M, Crane Co. All rights reserved.)

**TABLE 4-2 Reynolds Number**

Reynolds Number, $Re$	Coefficient	Numerator			Denominator	
		First Symbol	Second Symbol	Third Symbol	Fourth Symbol	Fifth Symbol
$Dv\rho/\mu$	—	ft	ft/s	lb/ft <sup>3</sup>	lb mass/ft s	—
$124d\rho/\nu z$	124	in.	ft/s	lb/ft <sup>3</sup>	cP	—
$50.7G\rho/dz$	50.7	gpm	lb/ft <sup>3</sup>	—	in.	cP
$6.32W/dz$	6.32	lb/h	—	—	in.	cP
$35.5B\rho/dz$	35.5	bbl/h	lb/ft <sup>3</sup>	—	in.	cP
$7,742dv/k$	7,742	in.	ft/s	—	—	cP
$3,162G/dk$	3,162	gpm	—	—	in.	cP
$2,214B/dk$	2,214	bbl/h	—	—	in.	cP
$22,735qp/dz$	22,735	ft <sup>3</sup> /s	lb/ft <sup>3</sup>	—	in.	cP
$378.9Q\rho/dz$	378.9	ft <sup>3</sup> /min	lb/ft <sup>3</sup>	—	in.	cP

**TABLE 4-3 Equivalent Roughness of Various Surfaces**

Material	Condition	Roughness Range	Recommended
Iron	New	0.01–0.0015 mm (0.0004–0.00006 in.)	0.002 mm (0.00008 in.)
	New	0.1–0.02 mm (0.004–0.0008 in.)	0.045 mm (0.0018 in.)
	Light rust	1.0–0.15 mm (0.04–0.006 in.)	0.3 mm (0.015 in.)
	General rust	3.0–1.0 mm	2.0 mm
	Wrought, new	0.046 mm (0.002 in.)	0.046 mm (0.002 in.)
	Cast, new	1.0–0.25 mm (0.04–0.01 in.)	0.30 mm (0.025 in.)
	Galvanized	0.15–0.025 mm (0.006–0.001 in.)	0.15 mm (0.006 in.)
	Asphalt-coated	1.0–0.1 mm (0.04–0.004 in.)	0.15 mm (0.006 in.)
	Sheet metal	0.1–0.02 mm (0.004–0.0008 in.)	0.03 mm (0.0012 in.)
	Concrete	Very smooth	0.04 mm (0.0016 in.)
Wood	Wood floated, brushed	0.8–0.2 mm (0.03–0.007 in.)	0.3 mm (0.012 in.)
	Rough, visible form marks	2.5–0.8 mm (0.1–0.03 in.)	2.0 mm (0.08 in.)
	Stave, used	1.0–0.25 mm (0.035–0.01 in.)	0.5 mm (0.02 in.)
	Glass or plastic	Drawn tubing	0.01–0.0015 mm (0.0004–0.00006 in.)
Rubber	Smooth tubing	0.07–0.006 mm (0.003–0.00025 in.)	0.01 mm (0.0004 in.)
	Wire-reinforced	4.0–0.3 mm (0.15–0.01 in.)	1.0 mm (0.04 in.)

(Source: Darby [5].)

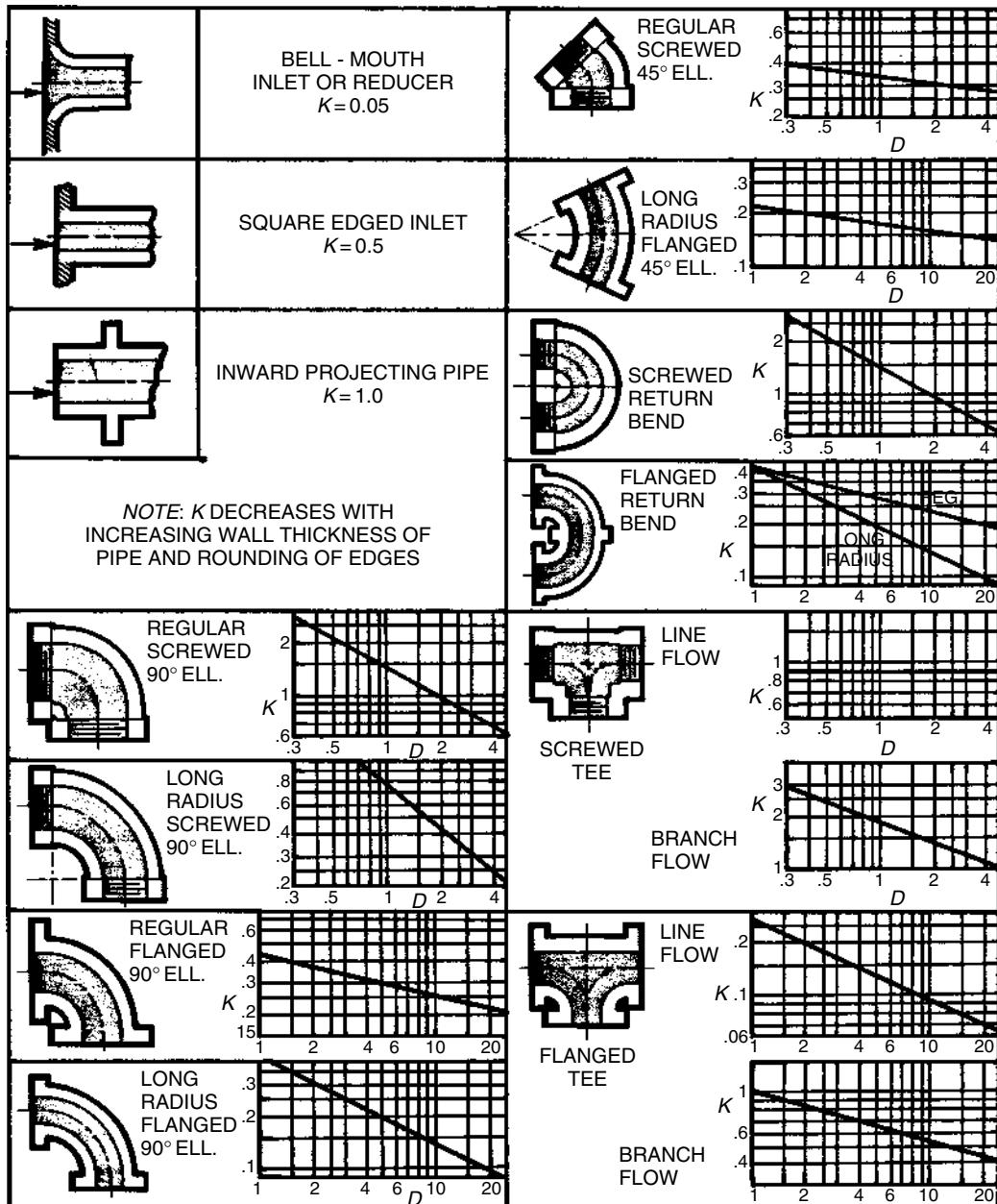
An explicit equation for calculating the friction factor ( $f_C$ ) as proposed by Chen [12] is

$$\frac{1}{\sqrt{f_C}} = -4 \log_{10} \left\{ \frac{\epsilon}{3.7D} - \frac{5.02}{Re} \log A \right\} \quad (4-35)$$

where

$$A = \frac{\epsilon/D}{3.7} + \left( \frac{6.7}{Re} \right)^{0.9}$$

 $\epsilon$  = epsilon, absolute pipe roughness, in. (mm) $D$  = pipe inside diameter, in. (mm).



$$h = K \frac{V^2}{2g} \text{ FEET OF FLUID}$$

**Figure 4-14a** Resistance coefficients for fittings. (Reprinted by permission from Hydraulic Institute, *Engineering Data Book*, 1st ed., 1979, Cleveland, OH.)

The relationship between Chen friction factor and Darcy friction factor is  $f_C = 1/4 f_D$ .

Churchill [13] developed a single expression for the friction factor in both laminar and turbulent flows as:

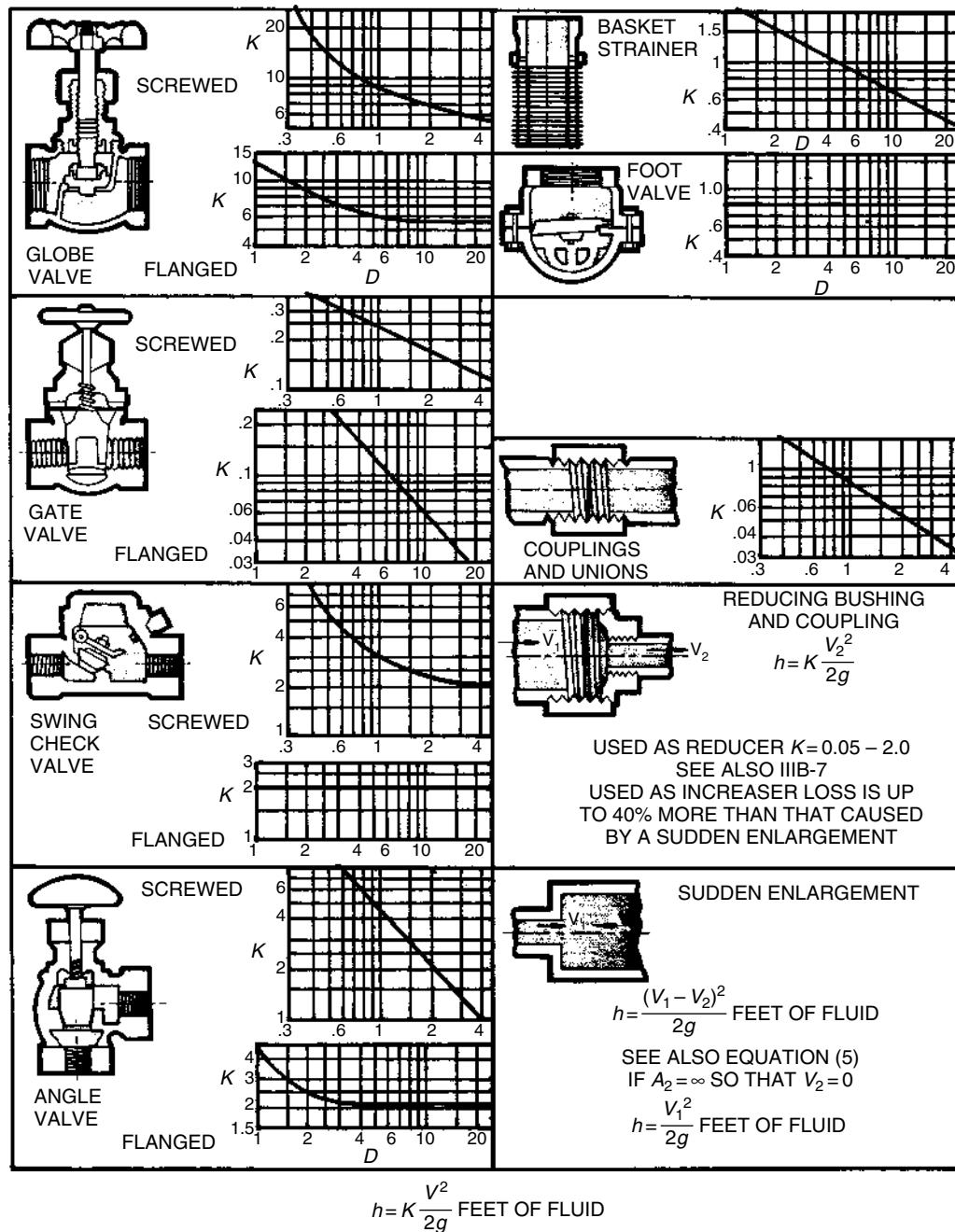
$$f = 2 \left[ \left( \frac{8}{Re} \right)^{12} + \frac{1}{(A+B)^{3/2}} \right]^{1/12} \quad (4-36)$$

where

$$A = \left[ 2.457 \ln \left( \frac{1}{(7/Re)^{0.9} + 0.27 \varepsilon/D} \right) \right]^{16}$$

$$B = \left( \frac{37,530}{Re} \right)^{16}$$

Note: Equation (4-36) gives the Fanning friction factor.



**Figure 4-14b** Resistance coefficients for valves and fittings. (Reprinted by permission from Hydraulic Institute, *Engineering Data Book*, 1st ed., 1979, Cleveland, OH.)

Equation (4-36) is an explicit equation, and adequately represents the Fanning friction factor over the entire range of Reynolds numbers within the accuracy of the data used to construct the Moody diagram, including a reasonable estimate for the intermediate or transition region between laminar and turbulent flow.

Gregory and Fogarasi [14] have provided a detailed review of other explicit equations for determining the friction factor, and concluded that Chen's friction factor equation is the most

satisfactory friction factor in comparison to others as shown in Table 4-4.

#### 4.16 FRICTION HEAD LOSS (RESISTANCE) IN PIPE, FITTINGS, AND CONNECTIONS

Friction head loss develops as fluids flow through the various pipes, elbows, tees, vessel connections, valves, and so on. These losses are expressed as loss of fluid static head in feet (meter) of

**TABLE 4-4 Explicit Equations for Calculating the Friction Factor for Rough Pipe**

Moody (1947)	Wood (1966)	Altshul (1975)	Swamee and Jain (1976)
$f = 1.375 \times 10^{-3} \times \left[ 1 + \left( 2 \times 10^4 \left[ \frac{\varepsilon}{D} \right] + \frac{10^6}{Re} \right)^{1/3} \right]$ <p>where</p> <p><math>\varepsilon</math> = absolute pipe roughness  <math>D</math> = inside pipe diameter  <math>Re</math> = Reynolds number.</p>	$f = A_1 + A_2 Re^{-A_3}$ <p>where</p> $A_1 = 0.094 \left( \frac{\varepsilon}{D} \right)^{0.225} + 0.53 \left( \frac{\varepsilon}{D} \right)$ $A_2 = 0.88 \left( \frac{\varepsilon}{D} \right)^{0.44}$ $A_3 = 1.62 \left( \frac{\varepsilon}{D} \right)^{0.134}$ <p>It is valid for <math>Re &gt; 10,000</math> and <math>10^{-5} &lt; \varepsilon/D &lt; 0.04</math></p>	$f = 0.11 \left( \frac{\varepsilon}{D} + \frac{68}{Re} \right)^{0.25}$	$\frac{1}{\sqrt{f}} = -4 \log \left[ \left( \frac{6.97}{Re} \right)^{0.9} + \frac{\varepsilon/D}{3.7} \right]$ <p>Jain (1976)</p>
<p><b>Churchill (1977)</b></p> $f = 8 \left[ \left( \frac{8}{Re} \right)^{12} + \frac{1}{(A_4 + A_5)^{3/2}} \right]^{1/12}$ <p>where</p> $A_4 = \left\{ 2.457 \ln \left( \frac{1}{\left( \left( \frac{7}{Re} \right)^{0.9} + 0.27 \frac{\varepsilon}{D} \right)} \right) \right\}^{16}$ $A_5 = \left( \frac{37,530}{Re} \right)^{16}$	<p><b>Chen (1979)</b></p> $\frac{1}{\sqrt{f}} = -2 \log \left( \frac{\varepsilon/D}{3.7065} - \frac{5.0452}{Re} \log A_6 \right)$ <p>where</p> $A_6 = \frac{(\varepsilon/D)^{1.1098}}{2.8257} + \left( \frac{7.149}{Re} \right)^{0.8981}$ <p><b>Chen (1991)</b></p> $\frac{1}{\sqrt{f}} = -4 \log \left( \frac{\varepsilon/D}{3.7} - \frac{5.02}{Re} \log A_7 \right)$ <p>where</p> $A_7 = \frac{\varepsilon/D}{3.7} + \left( \frac{6.7}{Re} \right)^{0.9}$	<p><b>Round (1980)</b></p> $\frac{1}{\sqrt{f}} = 3.6 \log \left( \frac{Re}{0.135 Re (\varepsilon/D) + 6.5} \right)$	<p><b>Zigrang and Sylvester (1982)</b></p> $\frac{1}{\sqrt{f}} = -4 \log \left( \frac{\varepsilon/D}{3.7} - \frac{5.02}{Re} \log A_8 \right)$ <p>where</p> $A_8 = \frac{\varepsilon/D}{3.7} + \frac{13}{Re}$ <p><b>Zigrang and Sylvester (1982)</b></p> $\frac{1}{\sqrt{f}} = -4 \log \left( \frac{\varepsilon/D}{3.7} - 5.02 \log A_9 \right)$ <p>where</p> $A_9 = \frac{\varepsilon/D}{3.7} - \frac{5.02}{Re} \log A_8$
<p><b>Harland (1983)</b></p> $\frac{1}{\sqrt{f}} = -3.6 \log \left[ \frac{6.9}{Re} + \left( \frac{\varepsilon/D}{3.7} \right)^{1.11} \right]$ <p>where</p> <p><math>\varepsilon</math> = absolute pipe roughness  <math>D</math> = inside pipe diameter  <math>Re</math> = Reynolds number.</p>	<p><b>Sergides (1984)</b></p> $f = \left( A_{10} - \frac{(A_{11} - A_{12})^2}{A_{12} - 2A_{11} + A_{10}} \right)^{-2}$ <p>where</p> $A_{10} = -2 \log \left( \frac{\varepsilon/D}{3.7} + \frac{12}{Re} \right)$ $A_{11} = -2 \log \left( \frac{\varepsilon/D}{3.7} + \frac{2.51 A_{10}}{Re} \right)$ $A_{12} = -2 \log \left( \frac{\varepsilon/D}{3.7} + \frac{2.51 A_{11}}{Re} \right)$		

fluid flowing. (Note: This is actually energy dissipated per unit of fluid mass.)

### PRESSURE DROP IN STRAIGHT PIPE: INCOMPRESSIBLE FLUID

The frictional resistance or pressure drop due to the flow in the fluid,  $h_f$ , is expressed by the Darcy equation:

$$h_f = f_D \left( \frac{L}{D} \right) \frac{v^2}{2g}, \text{ ft of fluid, resistance} \quad (4-37)$$

$$\begin{aligned} h_f &= 0.1863 \frac{fLv^2}{d} = 0.0311 \frac{fLQ^2}{d^5} \\ &= 0.000483 \frac{fLW^2}{\rho^2 d^5}, \text{ ft} \end{aligned} \quad (4-38)$$

where

$L$  = length of pipe, ft

$D$  = pipe internal diameter, ft

$f$  = Moody friction factor

$h_f$  = loss of static pressure head due to fluid flow, ft.

(Note: The units of  $d, v, Q, \rho$  are given above.)

In SI units,

$$\begin{aligned} h_f &= 51 \frac{fLv^2}{d} = 22,950 \frac{fLQ^2}{d^5} \\ &= 6,376,000 \frac{fLW^2}{\rho^2 d^5}, \text{ m} \end{aligned} \quad (4-39)$$

where

$L$  = length of pipe, m

$D$  = pipe internal diameter, m

$f$  = Moody friction factor.

(Note: The units of  $d, v, Q, \rho$  are given above.)

The frictional pressure loss (drop) is defined by:

$$\Delta P_f = \frac{1}{144} f_D \left( \frac{L}{D} \right) \rho \frac{v^2}{2g_c}, \text{ resistance loss, lb}_f/\text{in}^2. \quad (4-40)$$

In SI units,

$$\Delta P_f = f_D \left( \frac{L}{D} \right) \rho \frac{v^2}{2}, \text{ resistance loss, N/m}^2 \quad (4-41)$$

$$\text{For } Re < 2000, f_D = \frac{64}{Re} = \frac{64\mu}{124d\rho} \quad (4-42)$$

$$\begin{aligned} h_f &= 0.0962 \left( \frac{\mu Lv}{\rho d^2} \right) = 0.0393 \frac{\mu LQ}{\rho d^4} \\ &= 0.00490 \frac{\mu LW}{\rho^2 d^4}, \text{ ft.} \end{aligned} \quad (4-43)$$

In SI units,

$$\begin{aligned} h_f &= 3263 \left( \frac{\mu Lv}{\rho d^2} \right) = 69,220 \frac{\mu LQ}{\rho d^4} \\ &= 1,154,000 \frac{\mu LW}{\rho^2 d^4}, \text{ m} \end{aligned} \quad (4-44)$$

Substituting Eq. (4-42) into Eq. (4-40) gives

$$\begin{aligned} \Delta P &= 0.000668 \frac{\mu Lv}{d^2} = 0.000273 \frac{\mu LQ}{d^4} \\ &= 0.000034 \frac{\mu LW}{\rho d^4}, \text{ lb}_f/\text{in}^2. \end{aligned} \quad (4-45)$$

In SI units,

$$f_D = \frac{64}{Re} = \frac{64,000\mu}{dv\rho} \quad (4-46)$$

Substituting Eq. (4-46) into Eq. (4-41) gives

$$\Delta P = 0.32 \frac{\mu Lv}{d^2} = 6.79 \frac{\mu LQ}{d^4} = 113.2 \frac{\mu LW}{\rho d^4}, \text{ bar} \quad (4-47)$$

Note: These values for  $h_f$  and  $\Delta P_f$  are the losses between (differentials from) point (1) upstream to point (2) downstream, separated by a length,  $L$ . (Note that  $h_f$  is a PATH function, whereas  $\Delta P$  is a POINT function.) These are not absolute pressures, and cannot be meaningful if converted to such units (but they can be expressed as a pressure DIFFERENCE).

Feet of fluid,  $h_f$ , can be converted to pounds per square inch by:

$$h_f = \frac{\Delta P (144)}{\rho}, \text{ ft for any fluid} \quad (4-48)$$

Referenced to water, convert psi to feet of water:

$$h_f = \frac{[(1 \text{ lb/in}^2)] (144 \text{ in}^2/\text{ft}^2)}{62.3 \text{ lb}/\text{ft}^3} = 2.31 \text{ ft} \quad (4-49)$$

For conversion, 1 psi = 2.31 ft of water head.

This represents a column of water at 60° F, 2.31 ft high. The bottom pressure is one pound per square inch (psi) on a gauge. The pressure at the bottom as psi will vary with the density of the fluid.

In SI units,

$$h_f = 10,200 \frac{\Delta P}{\rho}, \text{ m} \quad (4-50)$$

where

$\Delta P$  = pressure difference (drop) in bar,

$\rho$  = fluid density, kg/m<sup>3</sup>.

For fluids other than water, the relationship is

$$1 \text{ psi} = 2.31 / (\text{sp gr related to water}), \text{ ft fluid} \quad (4-51)$$

In SI units,

$$h_f = 10,200 \frac{\Delta P}{(\text{sp gr related to water})}, \text{ m} \quad (4-52)$$

With extreme velocities of liquid in a pipe, the downstream pressure may fall to the vapor pressure of the liquid and cavitation with erosion will occur. Then the calculated flow rates or pressure drops are not accurate or reliable.

## 4.17 PRESSURE DROP IN FITTINGS, VALVES, AND CONNECTIONS

### INCOMPRESSIBLE FLUID

The resistance to flow through the various "piping" components that make up the system (except vessels, tanks, pumps – items which do not necessarily provide frictional resistance to flow) such as valves, fittings, and connections into or out of equipment (not the loss through the equipment) are established by test and presented in the published literature, but do vary depending on the investigator.

Resistance to fluid flow through pipe and piping components is brought about by dissipation of energy caused by (1) pipe component internal surface roughness along with the density and viscosity of the flowing fluid, (2) directional changes in the system through the piping components, (3) obstructions in the path to flow, and (4) changes in system component cross section and shape, whether gradual or sudden. This equation defines the loss coefficient  $K$  by

$$h_f = K(v^2/2g), \text{ ft (m) of the fluid flowing} \quad (4-53)$$

## 4.18 VELOCITY AND VELOCITY HEAD

The average or mean velocity is determined by the flow rate divided by the cross section area for flow in feet per second (meters per second),  $v$ . The loss of static pressure head due to friction of fluid flow is defined by

$$h_L = h_f = v^2/2g, \text{ termed velocity head, ft (m)} \quad (4-54)$$

Note that the static reduction (friction loss or energy dissipated) due to fluid flowing through a system component (valve, fitting, etc.) is expressed in terms of the number of corresponding velocity head, using the resistance coefficient,  $K$ , in the equation above. This  $K$  represents the loss coefficient or the energy dissipated in terms of the number of velocity heads (lost) due to flow through the respective system component. It is always associated with diameter for flow, hence velocity through the component. (*Note:*  $K$  can be defined on the basis of a specified velocity in Eq. (4-53), e.g., the velocity either into or out of a reducer, etc.).

The pipe loss coefficient is related to (from) the Darcy friction factor for straight pipe length by equation [4]:

$$K = \left( f_D \frac{L}{D} \right) \quad (4-55)$$

Head loss through a pipe,  $h_f$ , is

$$h_f = \left( f_D \frac{L}{D} \right) \frac{v^2}{2g} \quad (4-56)$$

Head loss through a valve,  $h_L$ , is (for instance)

$$h_L = K(v^2/2g) \quad (4-57)$$

## 4.19 EQUIVALENT LENGTHS OF FITTINGS

Instead of obtaining the friction loss directly and separately for each fitting at each pipe size, the equivalent length of a fitting may be found by equating Eq. (4-53) with Eq. (4-56) the Darcy friction loss formula for fluids in turbulent flow, such that

$$h_f = K \frac{v^2}{2g} = f \left( \frac{L}{D} \right) \frac{v^2}{2g} \quad (4-57a)$$

Therefore, it can be seen that  $K$ , the loss coefficient for a given fitting type divided by the Moody friction factor  $f$  is equal to the ratio of the "length" of the fitting divided by its inside diameter, or

$$\left( \frac{L}{D} \right) = \left( \frac{K}{f} \right) \quad (4-57b)$$

The  $K$  for a given fitting type is a direct function of the friction factor, so that  $L/D$  for that type of fitting is the same dimensionless constant for all pipe diameters. The right ordinate of Figure 4-13 gives the Moody friction factor for complete turbulent flow as a function of the inside pipe diameter for clean steel pipe. These factors are used with the  $K$  values found in the Hydraulic Institute or Crane data to determine  $L/D$ . The  $L/D$  value multiplied by the inside diameter\* of the pipe† gives the so-called equivalent length of pipe corresponding to the resistance induced by the fitting.

The Cameron tables show the  $L/D$  constant in addition to the  $K$  values for a wide range of fittings for common pipe sizes. The fittings are shown as pictorial representations similar to Figure 4-6 to assist the user in selecting the fitting most nearly corresponding to the actual one to be incorporated in the piping. Table 4-5 summarizes the  $L/D$  constants for the various fittings discussed in the Cameron Hydraulics Data book. The constants for valves are based on full-port openings in clean steel pipe.

## 4.20 L/D VALUES IN LAMINAR REGION

The  $L/D$  values in Table 4-5 are valid for flows in the transient and turbulent regions. However, it has been found good practice to modify  $L/D$  for use in the laminar region where  $Re < 1000$ . In this area, it is suggested that equivalent length can be calculated according to the expression

$$\left( \frac{L}{D} \right)_{\text{corr}} = \frac{Re}{1000} \left( \frac{L}{D} \right)_{\text{table}} \quad (4-58)$$

However, in no case should the equivalent length so calculated be less than the actual length of fitting. For various components'  $K$  values, see Figures 4-14a–4-18, and Tables 4-6 and 4-7.

Pressure drop through line systems containing more than one pipe size can be determined by (a) calculating the pressure drop separately for each section at assumed flows, or (b) determining the  $K$  totals for each pipe size separately using the 2 or 3- $K$  method. Flow then can be determined for a fixed head system by

$$Q = 19.65d^2 \left( \frac{h_L}{K} \right)^{1/2} = 236 d^2 \left( \frac{\Delta P}{K \rho} \right)^{1/2}, \text{ gpm} \quad (4-59)$$

In SI units,

$$Q = 0.2087d^2 \left( \frac{h_L}{K} \right)^{1/2} = 21.07 d^2 \left( \frac{\Delta P}{K \rho} \right)^{1/2}, \text{ l/min} \quad (4-60)$$

Of course, by selecting the proper equation, flows for vapors and gases can be determined in the same way, as the  $K$  value is for the fitting or valve and not for the fluid.

\*The diameter for the conversion is usually taken as feet, but if  $K$  is divided by 12, then  $L/D$  has the dimension of equivalent feet per inch of diameter.

†Use of nominal pipe size instead of actual inside diameter is usually sufficient for most engineering purposes.

**TABLE 4-5 Equivalent Length-to-Diameter Ratios for Fittings**

Fitting (Fully Open)	[L/d] (d in feet)	[L/d] d in inches	Remarks
Gate valve	8	0.67	Correct for constriction
Globe valve	340	28	Correct for constriction
Angle valve	55	4.6	Plug or Y type; correct for constriction
Angle valve	150	12.5	Globe type; correct for constriction
Ball valve	3	0.25	Correct for constriction
Butterfly valve	45	3.8	Correct for constriction
Plug valve	18	1.5	Correct for constriction
Three-way plug valve	30	2.5	Through flow; correct for constriction
Three-way plug valve	90	7.5	Branch flow; correct for constriction
Standard tee	20	1.7	Through flow
Standard tee	60	5.0	Branch flow
Standard 45° elbow	16	1.3	
Standard 90° elbow	30	2.5	r/d = 0.5
Long-radius 90° elbow	16	1.3	r/d = 1.0
90° bend	20	1.7	r/d = 1.0
90° bend	12	1.0	r/d = 2.0
90° bend	14	1.2	r/d = 4.0
90° bend	30	2.5	r/d = 10
90° bend	50	4.2	r/d = 20
30° miter bend	8	0.67	
45° miter bend	15	1.25	
60° miter bend	25	2.1	
90° miter bend	60	5.0	
close return bend	50	1.2	
Stop check valve (vertical disk rise, straight flow)	400	33	Minimum velocity = 55 / √ρ; correct for constriction
Stop check valve (vertical disk rise, right-angle flow)	200	16.7	Minimum velocity = 75 / √ρ; correct for constriction
Stop check valve (disk at 45°, right-angle flow)	350	25	Minimum velocity = 60 / √ρ; correct for constriction
Swing check valve	50	12.2	Minimum velocity = 50 / √ρ; correct for constriction

(Source: Adapted from Hydraulic Institute's *Engineering Data Book*.)

The head loss has been correlated as a function of the velocity head equation using  $K$  as the resistance (loss) coefficient in the equation.

$$h_f = K \frac{v^2}{2g} = K \frac{v^2}{64.4} = 0.00259 \frac{KQ^2}{d^4} = 0.0000403 \frac{KW^2}{\rho^2 d^4}, \text{ ft of fluid}$$

$$(4-61)$$

In SI units,

$$h_f = K \frac{v^2}{2g} = K \frac{v^2}{19.62} = 22.96 \frac{KQ^2}{d^4} = 8265 \times 10^7 \frac{K q^2}{d^4}$$

$$= 6377 \frac{KW^2}{\rho^2 d^4}, \text{ m}$$

$$(4-62)$$

#### 4.21 VALIDITY OF K VALUES

Equation (4-55) is valid for calculating the head loss due to valves and fittings for all conditions of flows: laminar, transition, and turbulent [4]. The  $K$  values are a related function of the pipe system component internal diameter and the velocity of flow for  $v^2/2g$ . The values in the standard tables are developed using standard ANSI pipe, valves, and fittings dimensions for each schedule or class [4]. The  $K$  value is for the size/type of pipe, fitting, or valve and not for the fluid, regardless of whether it is liquid or gas/vapor.

#### 4.22 LAMINAR FLOW

When the Reynolds number is below a value of 2000, the flow region is considered laminar. The pipe friction factor is defined as given in Eq. (4-42).

Between  $Re$  of 2000 and 4000, the flow is considered unsteady or unstable or transitional where laminar motion and turbulent mixing flows may alternate randomly [4].  $K$  values can still be calculated from the Reynolds number for straight length of pipe, and either the 2 or 3- $K$  method for pipe fittings.

$$K = f \left( \frac{L}{D} \right)$$

$$(4-55)$$

$$h_f = K \frac{(v_1 - v_2)^2}{2g}$$

$$(4-62a)$$

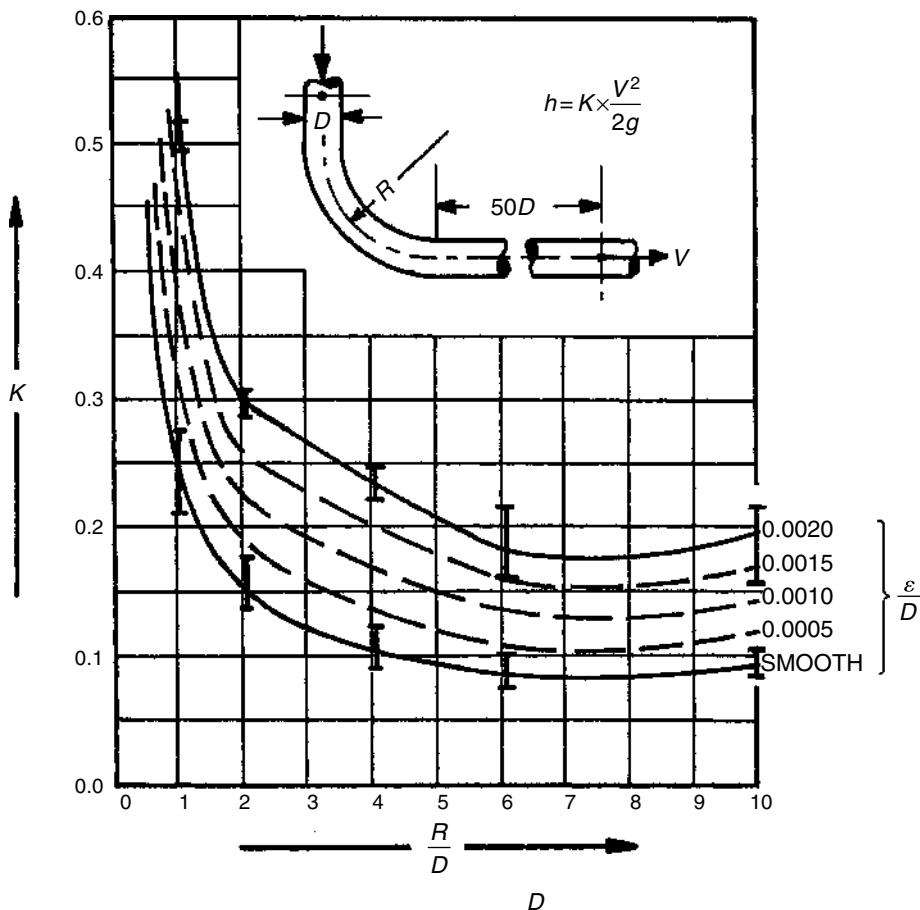
and

$$h_f = \left( f \frac{L}{D} \right) \left( \frac{v^2}{2g} \right), \text{ ft (m) fluid of pipe}$$

$$(4-37)$$

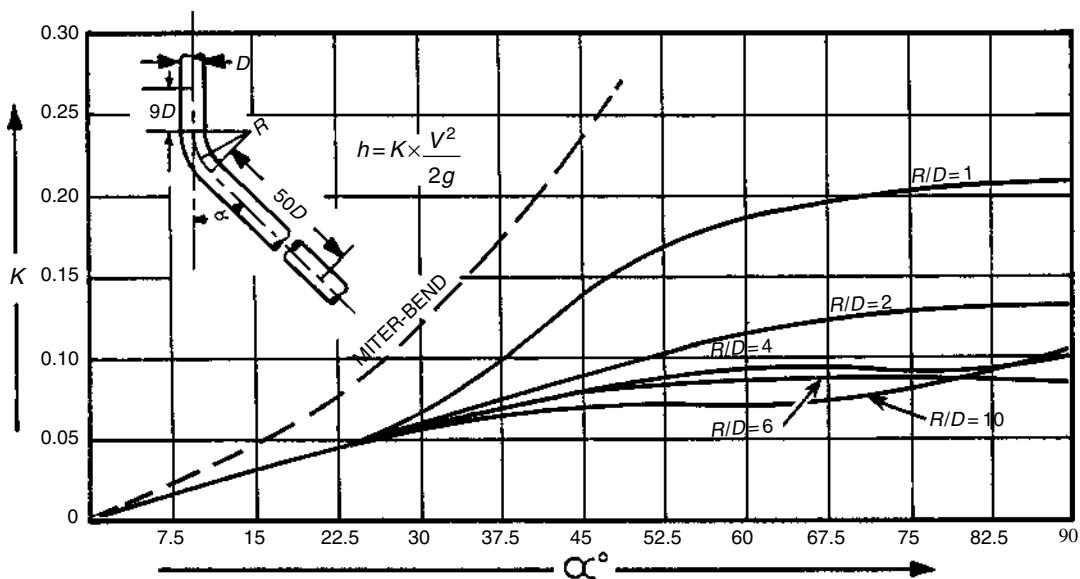
$$h_f = K \left( \frac{v^2}{2g} \right), \text{ ft (m) fluid for valves and fittings}$$

$$(4-57)$$

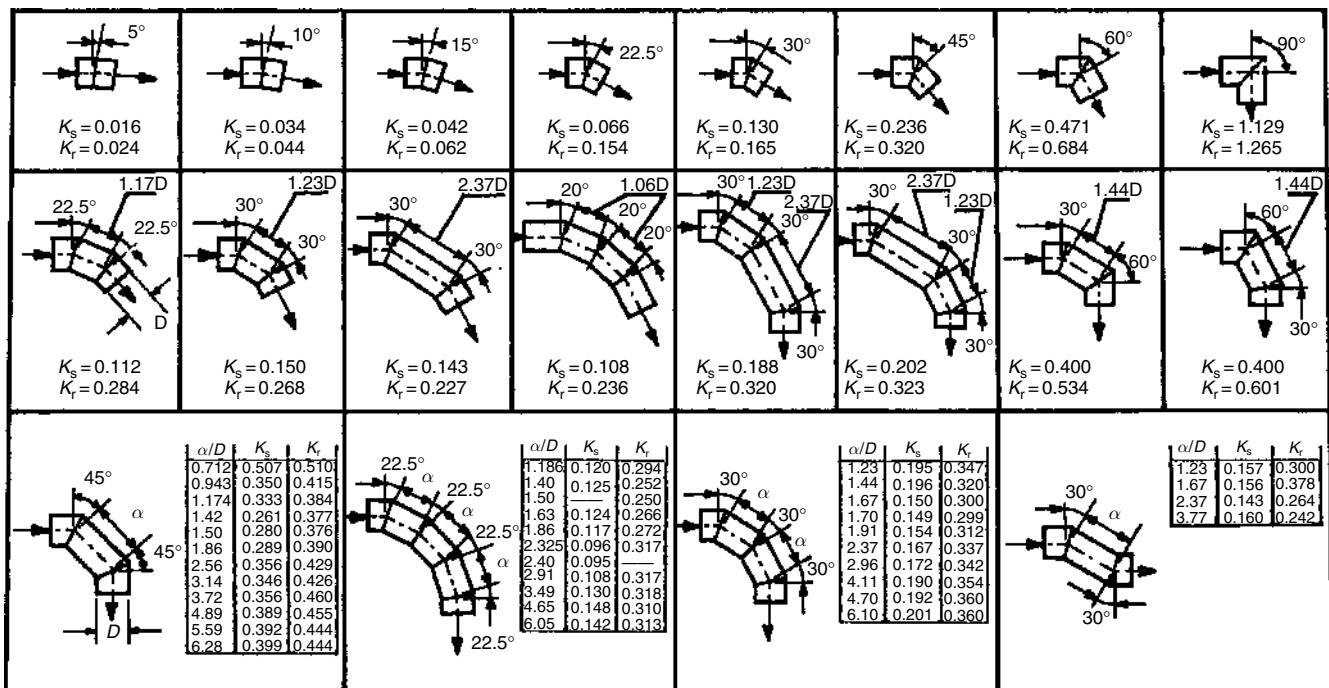


Note: 1. Use 0.00085 ft for  $\varepsilon/D$  uncoated cast iron and cast steel elbows.  
 2. Not reliable when  $R/D < 1.0$ .  
 3.  $R$  = radius of elbow, ft

**Figure 4-15a** Resistance coefficients for 90° bends of uniform diameter for water. (Reprinted by permission from Hydraulic Institute, *Engineering Data Book*, 1st ed., 1979, Cleveland, OH.)



**Figure 4-15b** Resistance coefficients for bends of uniform diameter and smooth surface at Reynolds number =  $2.25 \times 10^5$ . (Reprinted by permission from Hydraulic Institute, *Engineering Data Book*, 1st ed., 1979, Cleveland, OH.)



Note:

$K_s$  = RESISTANCE COEFFICIENT FOR SMOOTH SURFACE

$K_r$  = RESISTANCE COEFFICIENT FOR ROUGH SURFACE,  $\frac{\varepsilon}{D} \approx 0.0022$

\* OPTIMUM VALUE OF  $\alpha$  INTERPOLATED

Figure 4-16 Resistance coefficients for miter bends at Reynolds number =  $2.25 \times 10^5$  for water. (Reprinted by permission from Hydraulic Institute, *Engineering Data Book*, 1st ed., 1979, Cleveland, OH.)

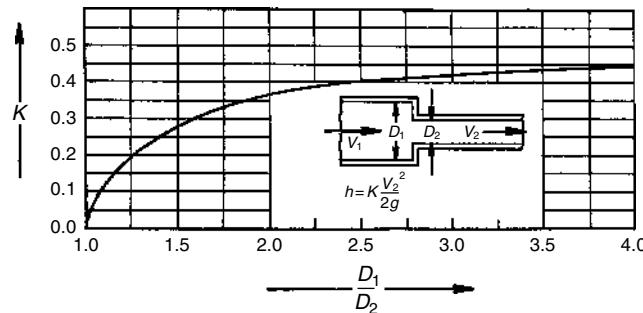


Figure 4-17 Resistance coefficients for reducers for water. (Reprinted by permission from Hydraulic Institute, *Engineering Data Book*, 1st ed., 1979, Cleveland, OH.)

For  $Re < 2000$   
English Engineering units

$$\Delta P/100 \text{ eq.ft}^* = 0.0668 \left( \frac{\mu v}{d^2} \right) = 0.0273 \left( \frac{\mu Q}{d^4} \right), \text{ psi}/100 \text{ eq.ft}$$

$$= 0.0034 \left( \frac{\mu W}{d^4 \rho} \right) \quad (4-63)$$

$$\Delta P = 0.0001078 K \rho v^2 = 0.00001799 \frac{K \rho Q^2}{d^4}$$

$$= 0.00000028 \frac{K W^2}{\rho d^4} \quad (4-63a)$$

In SI units,

$$\Delta P/100 \text{ eq.m}^* = 32 \left( \frac{\mu v}{d^2} \right) = 679 \left( \frac{\mu Q}{d^4} \right), \text{ bar}/100 \text{ eq.m}$$

$$= 11320 \left( \frac{\mu W}{d^4 \rho} \right) \quad (4-64)$$

Turbulent Flow ( $Re > 2000$ )  
English Engineering units

$$\Delta P/100 \text{ eq.ft}^* = 0.1294 \frac{f \rho v^2}{d} = 0.0216 \frac{f \rho Q^2}{d^5}, \text{ psi}/100 \text{ eq.ft}$$

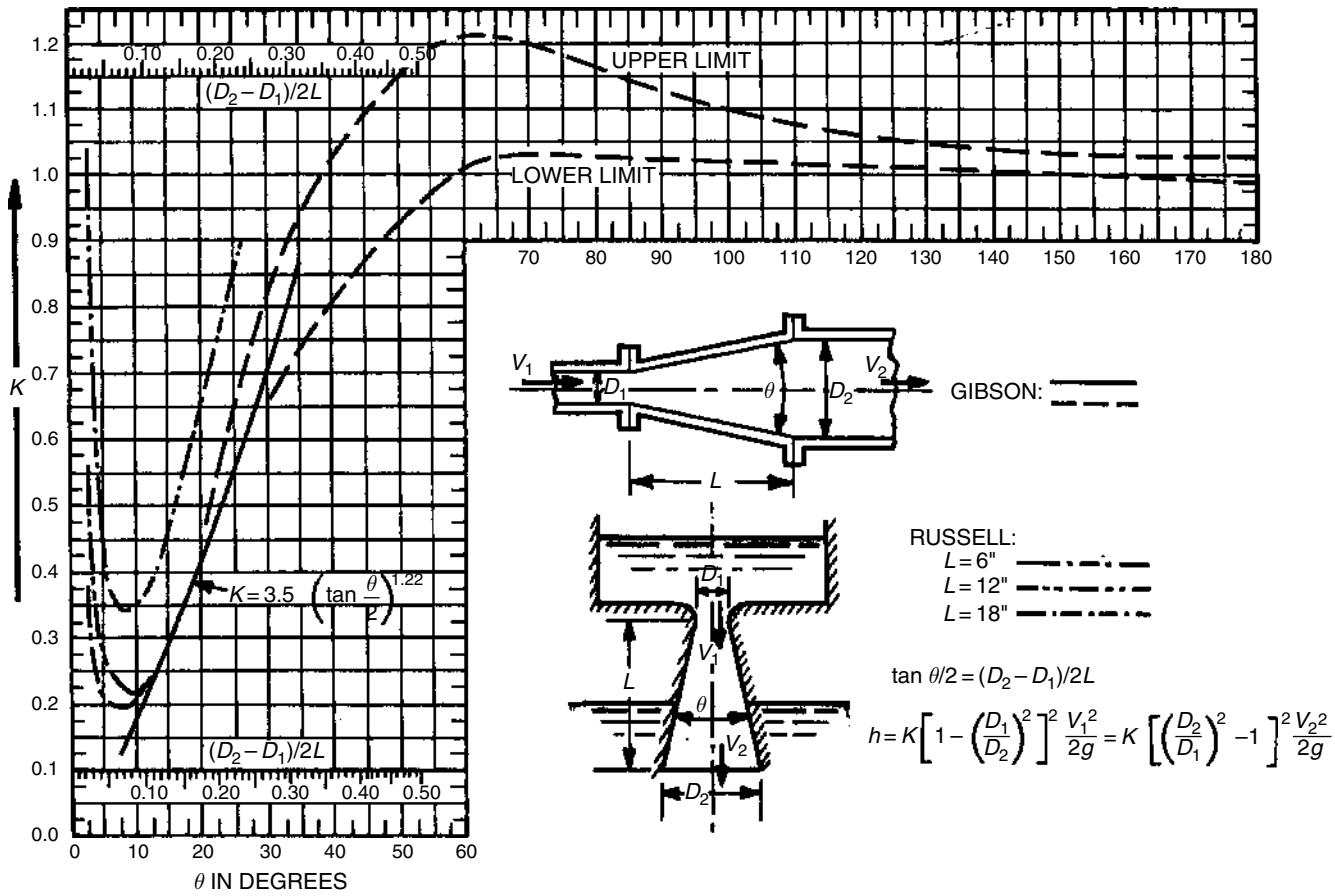
$$= 0.000336 \frac{f W^2}{\rho d^5} \quad (4-65)$$

In SI units,

$$\Delta P/100 \text{ eq.m}^* = 0.5 \frac{f \rho v^2}{d} = 225 \frac{f \rho Q^2}{d^5}, \text{ bar}/100 \text{ eq.m}$$

$$= 62,530 \frac{f W^2}{\rho d^5} \quad (4-66)$$

\* Equivalent feet (m) of straight pipe; that is, straight pipe plus equivalents for valves, fittings, other system components (except vessels, etc.). Therefore,  $\Delta P/100 \text{ eq. ft (m)} = \text{pressure drop (friction) per 100 equivalent feet (m) of straight pipe.}$



**Figure 4-18** Resistance coefficients for increases and diffusers for water. (Reprinted by permission from Hydraulic Institute, *Engineering Data Book*, 1st ed., 1979, Cleveland, OH.)

$$\begin{aligned}\Delta P &= 5 \times 10^{-6} K \rho v^2 = 2.25 \times 10^{-3} \frac{K \rho Q^2}{d^4}, \text{ bar} \\ &= 0.6253 \frac{K W^2}{\rho d^4}\end{aligned}\quad (4-67)$$

$$\Delta P = \left( \frac{\Delta P}{100} \right) (L_{eq}), \text{ psi (bar)} \quad (4-68)$$

The total pressure drop equation including the elevation change can be expressed as:

English Engineering units

$$\begin{aligned}-\Delta P = (P_1 - P_2) &= \frac{\rho}{144} \frac{\Delta v^2}{2g_c} + \frac{\rho \Delta z}{144} \frac{g}{g_c} \\ &+ \frac{1}{144} \left( f_D \frac{L}{D} + \sum K \right) \rho \frac{v^2}{2g_c}, \text{ psi}\end{aligned}\quad (4-69)$$

In SI units,

$$\begin{aligned}-\Delta P = (P_1 - P_2) &= \frac{\rho \Delta v^2}{2} + \rho g \Delta z \\ &+ \left( f_D \frac{L}{D} + \sum K \right) \rho \frac{v^2}{2}, \text{ N/m}^2\end{aligned}\quad (4-69a)$$

and the head loss equation can be expressed as:

$$h_f = \frac{e_f}{g} = \left( f_D \frac{L}{D} + \sum K \right) \frac{v^2}{2g} \quad (4-70)$$

where

$D$  = internal diameter of pipe, ft (m)

$f_D$  = Darcy friction factor

$K$  = loss coefficient

$L$  = pipe length, ft (m)

$v$  = fluid velocity, ft/s (m/s)

$\Delta P$  = pressure drop, psi ( $\text{N}/\text{m}^2$ )

$h_f$  = head loss, ft (m)

$\rho$  = fluid density,  $\text{lb}/\text{ft}^3$  ( $\text{kg}/\text{m}^3$ )

$e_f$  = energy dissipated by friction,  $\frac{\text{ft} - \text{lbf}}{\text{lbfm}}$  ( $\text{Nm}/\text{kg}$ ).

## 4.23 LOSS COEFFICIENT

$K$  is a dimensionless factor defined as the loss coefficient in a pipe fitting, and expressed in velocity heads. The velocity head is the amount of kinetic energy contained in a stream or the amount of potential energy required to accelerate a fluid to its flowing velocity. Most published  $K$  values apply to fully turbulent flow, because at high Reynolds number,  $K$  is found to be independent of Reynolds

**TABLE 4-6 "K" Factor Table: Representative Resistance Coefficients ( $K$ ) for Valves and Fittings. Pipe Friction Data for Clean Commercial Steel Pipe with Flow in Zone of Complete Turbulence**

Nominal size (in.)	1/2	3/3	1	1 1/4	1 1/2	2	2 1/2, 3	4	5	6	8-10	12-16	18-24
Friction factor, $f_T$	0.027	0.025	0.023	0.022	0.021	0.019	0.018	0.017	0.016	0.015	0.014	0.013	0.012

Formulas for Calculating "K" Factors for Valves and Fittings with Reduced Port.

**Formula 1**

$$K_1 = \frac{0.8 \sin \frac{\theta}{2} (1 - \beta^4)}{\beta^4}$$

**Formula 2**

$$K_2 = \frac{0.5 (1 - \beta^4) \sqrt{\sin \frac{\theta}{2}}}{\beta^4}$$

**Formula 3**

$$K_3 = \frac{2.6 \sin \frac{\theta}{2} (1 - \beta^4)^2}{\beta^4}$$

**Formula 4**

$$K_4 = \frac{(1 - \beta^4)^2}{\beta^4}$$

**Formula 5**

$$K_5 = \frac{K_1}{\beta^4} + \text{Formula 1} + \text{Formula 3}$$

$$K_5 = \frac{K_1 + \sin \frac{\theta}{2} [0.8 (1 - \beta^4) + 2.6 (1 - \beta^4)^2]}{\beta^4}$$

**Formula 6**

$$K_6 = \frac{K_1}{\beta^4} + \text{Formula 2} + \text{Formula 4}$$

$$K_6 = \frac{K_1 + 0.5 \sqrt{\sin \frac{\theta}{2} (1 - \beta^4) + (1 - \beta^4)^2}}{\beta^4}$$

**Formula 7**

$$K_7 = \frac{K_1}{\beta^4} + \beta (\text{Formula 2} + \text{Formula 4}) \text{ when } \theta = 180^\circ$$

$$K_7 = \frac{K_1 + \beta [0.5 (1 - \beta^4) + (1 - \beta^4)^2]}{\beta^4}$$

$$\beta = \frac{d_1}{d_2}$$

$$\beta^4 = \left( \frac{d_1}{d_2} \right)^4 = \frac{a_1}{a_2}$$

Subscript 1 defines dimensions and coefficients with reference to the smaller diameter.

Subscript 2 refers to the larger diameter.

### Sudden and gradual contraction



If:  $\theta \approx 45^\circ \dots \dots K_1 = \text{Formula 1}$   
 $\theta > 45^\circ \approx 180^\circ \dots K_1 = \text{Formula 2}$

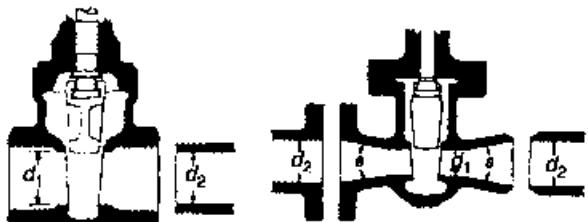
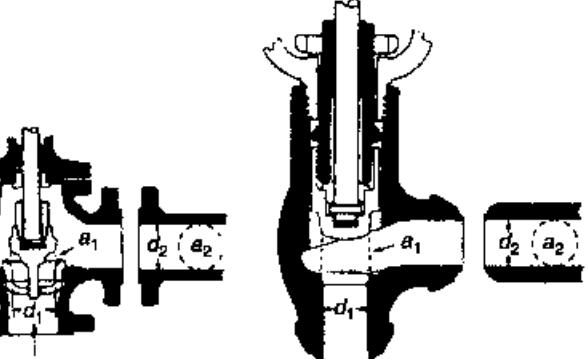
### Sudden and gradual enlargement



If:  $\theta \approx 45^\circ \dots \dots K_1 = \text{Formula 3}$   
 $\theta > 45^\circ \approx 180^\circ \dots K_1 = \text{Formula 4}$

(Continued)

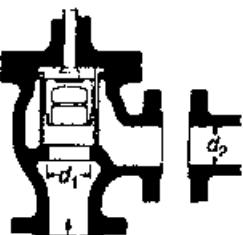
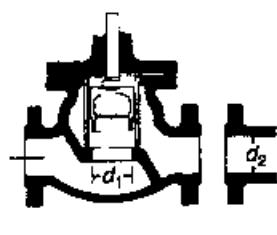
TABLE 4-6—(continued)

Gate values wedge disc, double disc, or plug type	Swing check values										
 <p>If: <math>\beta = 1, \theta = 0 \dots K_1 = 8 f_T</math>  <math>\beta &lt; 1</math> and <math>\theta \leq 45^\circ \dots K_2 = \text{Formula 5}</math>  <math>\beta &lt; 1</math> and <math>\theta &gt; 45^\circ \geq 180^\circ \dots K_3 = \text{Formula 6}</math></p>	 <p><math>K = 100 f_T</math>      <math>K = 50 f_T</math></p> <p>Minimum pipe velocity (fps) for full disc lift  <math>= 35 \sqrt{V}</math></p> <p>Minimum pipe velocity (fps) for full disc lift  <math>= 48 \sqrt{V}</math></p>										
<b>Globe and angle values</b>  <p>If: <math>\beta = 1 \dots K_1 = 340 f_T</math></p>	<b>Lift check values</b>  <p>If: <math>\beta = 1 \dots K_1 = 600 f_T</math>  <math>\beta &lt; 1 \dots K_2 = \text{Formula 7}</math></p> <p>Minimum pipe velocity (fps) for full disc lift  <math>= 40 \beta^2 \sqrt{V}</math></p>										
 <p>If: <math>\beta = 1 \dots K_1 = 150 f_T</math>      If: <math>\beta = 1 \dots K_1 = 55 f_T</math></p> <p>All globe and angle valves, whether reduced seat or throttled,  If: <math>\beta &lt; 1 \dots K_2 = \text{Formula 7}</math></p>	 <p>If: <math>\beta = 1 \dots K_1 = 55 f_T</math>  <math>\beta &lt; 1 \dots K_2 = \text{Formula 7}</math></p> <p>Minimum pipe velocity (fps) for full disc lift  <math>= 140 \beta^2 \sqrt{V}</math></p>										
	<b>Tilting disc check values</b> <table border="1" style="width: 100%; text-align: center;"> <thead> <tr> <th style="width: 50%;"><math>\alpha = 5^\circ</math></th> <th style="width: 50%;"><math>\alpha = 15^\circ</math></th> </tr> </thead> <tbody> <tr> <td>Sizes 2 to 8" ... <math>K = 40 f_T</math></td> <td><math>120 f_T</math></td> </tr> <tr> <td>Sizes 10 to 14" ... <math>K = 30 f_T</math></td> <td><math>90 f_T</math></td> </tr> <tr> <td>Sizes 16 to 48" ... <math>K = 20 f_T</math></td> <td><math>60 f_T</math></td> </tr> <tr> <td>Minimum pipe velocity (fps) for full disc lift  <math>= 80 \sqrt{V}</math></td> <td><math>30 \sqrt{V}</math></td> </tr> </tbody> </table>	$\alpha = 5^\circ$	$\alpha = 15^\circ$	Sizes 2 to 8" ... $K = 40 f_T$	$120 f_T$	Sizes 10 to 14" ... $K = 30 f_T$	$90 f_T$	Sizes 16 to 48" ... $K = 20 f_T$	$60 f_T$	Minimum pipe velocity (fps) for full disc lift $= 80 \sqrt{V}$	$30 \sqrt{V}$
$\alpha = 5^\circ$	$\alpha = 15^\circ$										
Sizes 2 to 8" ... $K = 40 f_T$	$120 f_T$										
Sizes 10 to 14" ... $K = 30 f_T$	$90 f_T$										
Sizes 16 to 48" ... $K = 20 f_T$	$60 f_T$										
Minimum pipe velocity (fps) for full disc lift $= 80 \sqrt{V}$	$30 \sqrt{V}$										

(continued)

TABLE 4-6—(continued)

**Stop-check values  
(Globe and angle types)**

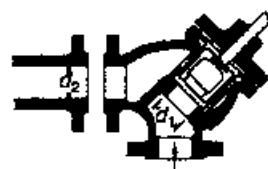
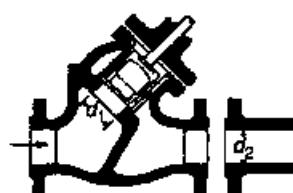


If:  
 $\beta = 1 \dots K_1 = 400 f_T$   
 $\beta < 1 \dots K_2 = \text{Formula 7}$

If:  
 $\beta = 1 \dots K_1 = 200 f_T$   
 $\beta < 1 \dots K_2 = \text{Formula 7}$

Minimum pipe velocity  
for full disc lift  
 $= 55 \beta^2 \sqrt{V}$

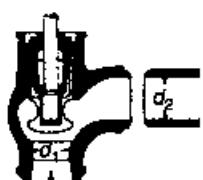
Minimum pipe velocity  
for full disc lift  
 $= 75 \beta^2 \sqrt{V}$



If:  
 $\beta = 1 \dots K_1 = 350 f_T$   
 $\beta < 1 \dots K_2 = \text{Formula 7}$

If:  
 $\beta = 1 \dots K_1 = 300 f_T$   
 $\beta < 1 \dots K_2 = \text{Formula 7}$

Minimum pipe velocity (fps) for full disc lift  
 $= 60 \beta^2 \sqrt{V}$



If:  
 $\beta = 1 \dots K_1 = 55 f_T$   
 $\beta < 1 \dots K_2 = \text{Formula 7}$

If:  
 $\beta = 1 \dots K_1 = 55 f_T$   
 $\beta < 1 \dots K_2 = \text{Formula 7}$

Minimum pipe velocity (fps) for full disc lift  
 $= 140 \beta^2 \sqrt{V}$

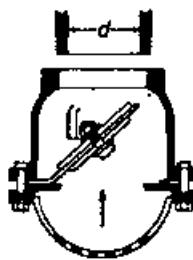
**Foot values with strainer**

**Poppet disk**



$K = 420 f_T$

**Hinged disk**

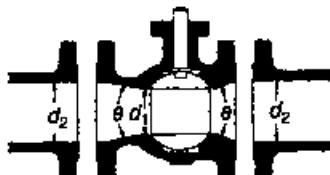


$K = 75 f_T$

Minimum pipe velocity (fps) for full disc lift  
 $= 15 \sqrt{V}$

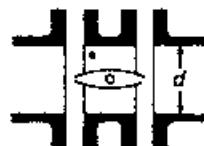
Minimum pipe velocity (fps) for full disc lift  
 $= 35 \sqrt{V}$

**Ball values**



If:  
 $\theta = 0 \dots K_1 = 3 f_T$   
 $\theta < 1 \text{ and } \theta \leq 45^\circ \dots K_2 = \text{Formula 5}$   
 $\theta < 1 \text{ and } \theta > 45^\circ \geq 180^\circ \dots K_2 = \text{Formula 6}$

**Butterfly values**



Sizes 2 to 8" ...  $K = 45 f_T$   
 Sizes 10 to 14" ...  $K = 35 f_T$   
 Sizes 16 to 24" ...  $K = 25 f_T$

(continued)

TABLE 4-6—(continued)

Plug valves and cocks			Standard elbows																													
Straight-way	3-way		90°	45°																												
If: $\beta = 1$ , $K_1 = 18 f_T$	If: $\beta = 1$ , $K_1 = 30 f_T$	If: $\beta = 1$ , $K_1 = 90 f_T$	$K = 30 f_T$	$K = 16 f_T$																												
If: $\beta < 1 \dots K_1 = \text{Formula 6}$																																
Mitre bends																																
	<table border="1"> <thead> <tr> <th><math>\alpha</math></th> <th><math>K</math></th> </tr> </thead> <tbody> <tr><td>0°</td><td>2 ft</td></tr> <tr><td>15°</td><td>4 ft</td></tr> <tr><td>30°</td><td>8 ft</td></tr> <tr><td>45°</td><td>15 ft</td></tr> <tr><td>60°</td><td>25 ft</td></tr> <tr><td>75°</td><td>40 ft</td></tr> <tr><td>90°</td><td>60 ft</td></tr> </tbody> </table>	$\alpha$	$K$	0°	2 ft	15°	4 ft	30°	8 ft	45°	15 ft	60°	25 ft	75°	40 ft	90°	60 ft															
$\alpha$	$K$																															
0°	2 ft																															
15°	4 ft																															
30°	8 ft																															
45°	15 ft																															
60°	25 ft																															
75°	40 ft																															
90°	60 ft																															
			Flow thru run ..... $K = 20 f_T$																													
			Flow thru branch ..... $K = 60 f_T$																													
90° Pipe bends and flanged or butt-welding 90° elbows																																
	<table border="1"> <thead> <tr> <th><math>r/d</math></th> <th><math>K</math></th> <th><math>r/d</math></th> <th><math>K</math></th> </tr> </thead> <tbody> <tr><td>1</td><td>20 ft</td><td>10</td><td>30 ft</td></tr> <tr><td>2</td><td>12 ft</td><td>12</td><td>34 ft</td></tr> <tr><td>3</td><td>12 ft</td><td>14</td><td>38 ft</td></tr> <tr><td>4</td><td>14 ft</td><td>16</td><td>42 ft</td></tr> <tr><td>6</td><td>17 ft</td><td>18</td><td>46 ft</td></tr> <tr><td>8</td><td>24 ft</td><td>20</td><td>50 ft</td></tr> </tbody> </table>	$r/d$	$K$	$r/d$	$K$	1	20 ft	10	30 ft	2	12 ft	12	34 ft	3	12 ft	14	38 ft	4	14 ft	16	42 ft	6	17 ft	18	46 ft	8	24 ft	20	50 ft			
$r/d$	$K$	$r/d$	$K$																													
1	20 ft	10	30 ft																													
2	12 ft	12	34 ft																													
3	12 ft	14	38 ft																													
4	14 ft	16	42 ft																													
6	17 ft	18	46 ft																													
8	24 ft	20	50 ft																													
The resistance coefficient, $K_B$ , for pipe bends other than 90° may be determined as follows:																																
$K_B = (n - 1) \left( 0.25 + f_T \frac{r}{d} + 0.5 K \right) + K$																																
$n$ = number of 90° bends																																
$K$ = resistance coefficient for one 90° bend (per table)																																
Close pattern return bends																																
			$K = 50 f_T$																													
Pipe entrance																																
Inward projecting			<table border="1"> <thead> <tr> <th><math>r/d</math></th> <th><math>K</math></th> </tr> </thead> <tbody> <tr><td>0.00*</td><td>0.5</td></tr> <tr><td>0.02</td><td>0.28</td></tr> <tr><td>0.04</td><td>0.24</td></tr> <tr><td>0.06</td><td>0.18</td></tr> <tr><td>0.10</td><td>0.09</td></tr> <tr><td>0.15 &amp; up</td><td>0.04</td></tr> </tbody> </table>	$r/d$	$K$	0.00*	0.5	0.02	0.28	0.04	0.24	0.06	0.18	0.10	0.09	0.15 & up	0.04	Flush														
$r/d$	$K$																															
0.00*	0.5																															
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$K = 0.78$			*Sharp-edged																													
For $K$ , see table																																
Pipe exit																																
Projecting			Sharp-edged	rounded																												
$K = 1.0$																																
$K = 1.0$																																
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**TABLE 4-7 Resistance Coefficients for Valves and Fittings**

Approximate Range of Variations for $K$		
	Fitting	Range of Variation
90° elbow	Regular screwed	±20% above 2-in. size
	Regular screwed	±40% below 2-in. size
	Long radius, screwed	±25%
	Regular flanged	±35%
	Long radius, flanged	±30%
45° elbow	Regular screwed	±10%
	Long radius, flanged	±10%
180° bend	Regular screwed	±25%
	Regular flanged	±35%
	Long radius, flanged	±30%
Tee	Screwed, line or branch flow	±25%
	Flanged, line or branch flow	±35%
Globe valve	Screwed	±25%
	Flanged	±25%
Gate valve	Screwed	±25%
	Flanged	±50%
Check valve	Screwed	±30%
	Flanged	{+200% -80%}
Sleeve check valve		Multiply flanged values by 0.2–0.5
Tilting check valve		Multiply flanged values by 0.13–0.19
Drainage gate check		Multiply flanged values by 0.03–0.07
Angle valve	Screwed	±20%
	Flanged	±50%
Basket strainer		±50%
Foot valve		±50%
Couplings		±50%
Unions		±50%
Reducers		±50%

(Source: Reprinted by permission from Hydraulic Institute, *Engineering Data Handbook*, 1st ed., 1979, Cleveland, OH.)

*Notes on the use of Figures 4-12a and b, and Table 4-7*

1. The values of  $D$  given in the charts is nominal IPS (Iron Pipe Size).
2. For velocities below 15 ft/s, check valves and foot valves will be only partially open and will exhibit higher values of  $K$  than that shown in the charts.

number. However, the 2- $K$  technique includes a correction factor for low Reynolds number. Hooper [15, 16] gives a detailed analysis of his method compared to others and has shown that the 2- $K$  method is the most suitable for any pipe size. In general, the 2- $K$  method is independent of the roughness of the fittings, but it is a function of Reynolds number and of the exact geometry of the fitting. The method can be expressed as:

$$K_f = \frac{K_1}{Re} + K_\infty \left( 1 + \frac{1}{ID, \text{in.}} \right) \quad (4-71)$$

In SI units,

$$K_f = \frac{K_1}{Re} + K_\infty \left( 1 + \frac{25.4}{ID, \text{mm}} \right) \quad (4-72)$$

where

$$K_1 = K \text{ for the fitting at } Re = 1$$

$$K_\infty = K \text{ for a large fitting at } Re = \infty$$

$$ID = \text{internal diameter of attached pipe, in. or mm.}$$

The Hooper's method is valid over a much wider range of Reynolds numbers than the other methods. However, the effect of the pipe size (e.g.,  $1/ID$ ) in Eq. (4-71) does not accurately reflect data over a wide range of sizes for valves and fittings as reported in sources [1, 4 and 7]. Hooper's method and that given in Crane tend to under-predict the friction loss for pipes of larger diameters. The disadvantage of the 2- $K$  method is that it is limited to the number of values of  $K_1$  and  $K_\infty$  available as shown in Table 4-8. For other fittings, approximations must be made from data in Table 4-8.

Darby [5, 17] recently developed a 3- $K$  equation which represents various valves, tees, and elbows and is expressed by

$$K_f = \frac{K_1}{Re} + K_i \left( 1 + \frac{K_d}{D_{n,in.}^{0.3}} \right) \quad (4-73)$$

**TABLE 4-8 2-K Constants for Loss Coefficients for Valves and Fittings**

		Fitting Type	$K = \frac{K_1}{Re} + K_\infty \left( 1 + \frac{1}{ID_{inch}} \right)$	$K_1$	$K_\infty$
Elbows	90°	Standard ( $R/D = 1$ )	Screwed	800	0.40
		Long-radius ( $R/D = 1.5$ )	Flanged/welded	800	0.25
		Mitered ( $R/D = 1.5$ )	All types	800	0.20
			1 weld (90° angle)	1000	1.15
			2 weld (45° angle)	800	0.35
	45°		3 weld (30° angle)	800	0.30
		Standard ( $R/D = 1$ )	4 weld (22.5° angle)	800	0.27
		Long-radius ( $R/D = 1.5$ )	5 weld (18° angle)	800	0.25
		Mitered ( $R/D = 1.5$ )	All types	500	0.20
			All types	500	0.15
Used as Elbow	180°	Standard ( $R/D = 1$ )	1 weld (45° angle)	500	0.25
		Standard ( $R/D = 1$ )	2 weld (22.5° angle)	500	0.15
	Used as Elbow	Long-radius ( $R/D = 1.5$ )	Screwed	1000	0.70
		Standard	Flanged/welded	1000	0.35
		Long-radius	All types	1000	0.30
Tees	Run Through Tee	Standard	Screwed	500	0.70
		Stub-in type branch	Screwed	800	0.40
		Stub-in type branch	Flanged/welded	800	0.80
	Tees	Stub-in type branch		1000	1.00
Valves	Gate Ball Plug Globe	Full line size	$\beta = 1.0$	300	0.10
		Reduced trim	$\beta = 0.9$	500	0.15
		Reduced trim	$\beta = 0.8$	1000	0.25
		Standard		1500	4.00
	Diaphragm Butterfly Check	Angle or Y-type		1000	2.00
		Dam type		1000	2.00
		Lift		800	0.25
		Swing		2000	10.00
		Tilting-disc		1500	1.50
				1000	0.50

(Source: Hooper, *Chem. Eng.*, Aug 24, 1981, pp. 96–100.)

Note: For pipe bends of angle 45°–180° with  $R/D = 5$ , use  $K$  values for  $R/D = 1.5$ . For flow through crosses, use the appropriate tee values. Inlet, flush,  $K = 160/Re + 0.5$ ; Inlet, intruding,  $K = 160/Re = 1.0$ ; Exit,  $K = 1.0$ .  $K = K_1/Re + K_\infty (1 + 1/ID)$ , with ID in inches.

where

 $D_n$  = nominal pipe diameter in (mm)

In SI units,

$$K_f = \frac{K_1}{Re} + K_i \left( 1 + K_d \left( \frac{25.4}{D_{n, \text{mm}}} \right)^{0.3} \right) \quad (4-74)$$

where  $K_1$  are mostly those of the Hooper 2- $K$  method, and  $K_i$  were mostly determined from the Crane data. The values of  $K_d$  are all very close to 4.0, and this can be used to scale known values of  $K_f$  for a given pipe size to apply to other sizes. Darby's 3- $K$  method is the most accurate method for all Reynolds numbers and fitting sizes, since it scales with pipe size more accurately than the 2- $K$  method.

The conversion between equivalent pipe length and the resistance coefficient,  $K_f$ , can be expressed as

$$K_f = f_D \frac{L_{\text{eq}}}{D} \quad (4-55)$$

Table 4-9 lists values of the 3- $K$  method, and Table 4-10 shows a comparison between the methods used to calculate the loss coefficient,  $K_f$ , for pipe fittings and valves.

#### 4.24 SUDDEN ENLARGEMENT OR CONTRACTION [2]

For sudden enlargements in a pipe system when there is an abrupt change from a smaller pipe flowing into a larger pipe, the resistance coefficient (or loss coefficient)  $K$  is given by:

For sudden enlargement:

$$K_1 = (1 - D_1^2/D_2^2)^2 = (1 - \beta^2)^2 \quad (4-75)$$

where subscripts 1 and 2 refer to the smaller (upstream) and larger pipes respectively [4] or using Eq. (4-57),  $h_f$  is given by

$$h_f = K_1 \left[ 1 - \left( \frac{d_1^2}{d_2^2} \right)^2 \right]^2 \left( \frac{v_1^2}{2g} \right), \text{ ft (m) of fluid} \quad (4-76)$$

$$K_1 = (1 - d_1^2/d_2^2)^2 \quad (4-77)$$

(Note: This applies for fully turbulent flow.)

**TABLE 4-9 3-K Constants for Loss Coefficients for Valves and Fittings**

	Fitting	$K = \frac{K_m}{Re} + K_i \left(1 + \frac{K_d}{D_n^{0.3}}\right)^*$	(L/D) <sub>eq</sub>	$K_m$	$K_i$	$K_d$	
<i>Elbows - 90°</i>	Threaded, standard	$r/D = 1$	30	800	0.14	4.0	
	Threaded, long radius	$r/D = 1.5$	16	800	0.071	4.2	
	Flanged, welded, bends	$r/D = 1$	20	800	0.091	4.0	
		$r/D = 2$	12	800	0.056	3.9	
		$r/D = 4$	14	800	0.066	3.9	
		$r/D = 6$	17	800	0.075	4.2	
		Mitered	1 weld, 90°	60	1000	0.27	4.0
			2 welds, 45°	15	800	0.068	4.1
			3 welds, 30°	8	800	0.035	4.2
<i>Elbows - 45°</i>	Threaded, standard	$r/D = 1$	16	500	0.071	4.2	
	Long radius	$r/D = 1.5$		500	0.052	4.0	
	Mitered, 1 weld	45°	15	500	0.086	4.0	
	Mitered, 2 welds	22.5°	6	500	0.052	4.0	
<i>Elbows - 180°</i>	Threaded,						
	Close return bend	$r/D = 1$	50	1000	0.23	4.0	
	Flanged	$r/D = 1$		1000	0.12	4.0	
	All	$r/D = 1.5$		1000	0.10	4.0	
<i>Tees</i>	Through-branch (as elbow)						
	Threaded	$r/D = 1$	60	500	0.274	4.0	
		$r/D = 1.5$		800	0.14	4.0	
	Flanged	$r/D = 1$	20	800	0.28	4.0	
	Stub-in branch			1000	0.34	4.0	
	Run through threaded	$r/D = 1$	20	200	0.091	4.0	
<i>Valves</i>	Flanged	$r/D = 1$		150	0.017	4.0	
	Stub-in branch			100	0	0	
	Angle valve - 45°	Full line size, $\beta = 1$	55	950	0.25	4.0	
	Angle valve - 90°	Full line size, $\beta = 1$	150	1000	0.69	4.0	
	Globe valve	Standard, $\beta = 1$	340	1500	1.70	3.6	
	Plug valve	Branch flow	90	500	0.41	4.0	
	Plug valve	Straight through	18	300	0.084	3.9	
	Plug valve	Three-way (flow through)	30	300	0.14	4.0	
	Gate valve	Standard, $\beta = 1$	8	300	0.037	3.9	
	Ball valve	Standard, $\beta = 1$	3	300	0.017	4.0	
<i>Diaphragm</i>	Diaphragm	Dam-type		1000	0.69	4.9	
	Swing check <sup>†</sup>	$V_{min} = 35 \rho^{-1/2}$	100	1500	0.46	4.0	
	Lift check <sup>†</sup>	$V_{min} = 40 \rho^{-1/2}$	600	2000	2.85	3.8	

(Source: Darby, *Chem. Eng.*, July, 1999, pp. 101–104.)<sup>\*</sup> See Equation<sup>†</sup> Units of  $\rho$  are  $\text{lb}_m/\text{ft}^3$  $D_n$  = nominal pipe size, in.

For sudden pipe system contraction as represented in Figures 4-14a–4-18, the values of the resistance or loss coefficient,  $K$ , can be read from the charts. For more details for various angles of enlargements and contractions, see [2, 4].

For sudden contractions:

$$K_1 = 0.5 \left(1 - d_1^2/d_2^2\right) = 0.5 \left(1 - \beta^2\right) \quad (4-78)$$

This applies for fully turbulent flow, where subscripts 1 and 2 indicate small and large pipes respectively.

Then

$$h_f = K_1 \left(\frac{v_1^2}{2g}\right), \text{ ft (m)} \quad (4-79)$$

Table 4-11 shows how  $K$  varies with changes in pipe size.

#### 4.25 PIPING SYSTEMS

The  $K$  coefficient values for each of the items of pipe, bends, valves, fittings, contractions, enlargements, entrance/exits into/from

vessels are additive as long as they are on the same size (velocity) basis (see Table 4-6 and Figures 4-14a–4-18). Thus the resistance equation is applicable to calculate the head or pressure loss through the specific system when the combined  $K$  value is used.

$$h_f = K \left(\frac{v^2}{2g}\right) \quad (4-57)$$

or

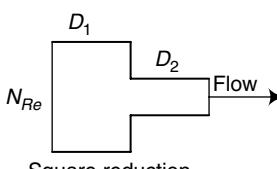
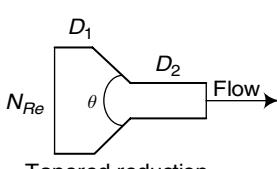
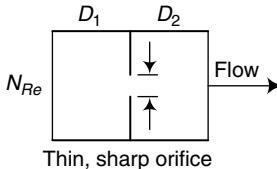
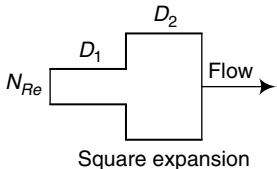
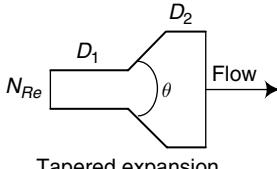
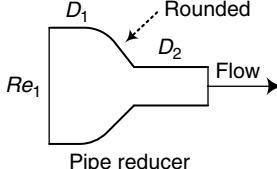
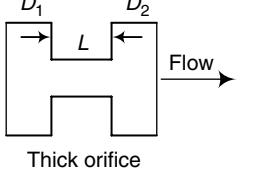
$$h_f = f \left(\frac{L}{D}\right) \left(\frac{v^2}{2g}\right) \quad (4-37)$$

where  $K$  = summation of all  $K$  values in a specific system, when all are on the same size (internal flow) basis. See discussion in “Common Denominator” section. (Note: The frictional energy loss, or head loss, is additive even if the velocities change).

**TABLE 4-10 Comparison Between the Loss Coefficients (velocity heads),  $K_f$ , for Pipe Fittings and Valves**

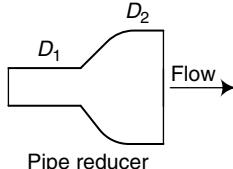
The Equivalent length ( $L/D$ ) Method	K-factor Method	New Crane Method	2-K (Hooper's) method	3-K (Darby's) method
<p>The equivalent length adds some hypothetical length of pipe to the actual length of the fitting. However, the drawback is that the equivalent length for a given fitting is not constant, but depends on Reynolds number, pipe roughness, pipe size, and geometry.</p> $K_f = f \left( \frac{L_{eq}}{D} \right)$ $L_{eq} = K_f \cdot \frac{D}{f}$ <p>Every equivalent length has a specific friction factor. The method assumes that</p> <p>(1) Sizes of fittings of a given type can be scaled by the corresponding pipe diameter.          (2) The influence of Reynolds number on the friction loss on the fitting is the same as the pipe loss.</p> <p>However, neither of the above assumption is accurate.</p> <p>Furthermore, the nature of the laminar or turbulent flow field within a valve or a fitting is generally quite different from that in a straight pipe.</p> <p>Therefore, there is an uncertainty when determining the effect of Reynolds number on the loss coefficients. This method does not properly account for the lack of exact scaling for valves and fittings.</p>	<p>The excess loss in a fitting is normally expressed in a dimensionless "K" factor. The excess head loss (<math>\Delta H</math>) is less than the total by the amount of frictional loss that would be experienced by straight pipe of the same physical length.</p> <p>The loss coefficient <math>K_f</math>, depends on the Reynolds number of the flow. The values of <math>K_f</math> at low <math>Re</math> can be significantly greater than those at high <math>Re</math>.</p> <p>Additionally, valves and fittings do not scale exactly, e.g., the loss coefficient for a 1/4 in. valve is not the same as that for a 4 in. valve.</p>	<p>The latest version of the equivalent length method given in Crane Technical Paper 410 (Crane Co. 1991) requires the use of two friction factors. The first is the actual friction factor for flow in the straight pipe (<math>f</math>), and the second is a standard friction factor for the particular fitting (<math>f_T</math>) given in Table 4-.</p> $h_L = K_f \frac{V^2}{2g}$ <p>where</p> $K_f = 4f \left( \frac{L_{eq}}{D} \right)$ <p>The value of <math>f</math> is determined from the Colebrook equation</p> $f = \frac{0.0625}{[\log(3.7D/\varepsilon)]^2}$ <p>where</p> <p><math>\varepsilon</math> is the pipe roughness (0.0018 in., 0.045 mm) for commercial steel pipe.</p> <p>The Crane paper gives <math>f_T</math> for a wide variety of fittings, valves, etc. This method gives satisfactory results for high turbulence levels (e.g., high Reynolds number) but is less accurate at low Reynolds number.</p> <p>This method provides a better estimate for the effect of geometry, but does not reflect any Reynolds number dependence.</p>	<p><math>K</math> is a dimensionless factor defined as the excess head loss in pipe fitting, expressed in velocity heads. <math>K</math> does not depend on the roughness of the fitting (or the attached pipe) or the size of the system, but it is a function of the Reynolds number and the exact geometry of the fitting. The 2-K method accounts for these dependencies by the following equation.</p>	<p>Darby's 3-K method represents improved features over the widest range of Reynolds number and fitting size, and is expressed by</p> $K_f = \frac{K_1}{Re} + K_i \left( 1 + \frac{K_d}{D_{n,in.}^{0.3}} \right)$ <p>Or in SI units</p> $K_f = \frac{K_1}{Re} + K_i \left( 1 + K_d \left[ \frac{25.4}{D_{n,mm}} \right]^{0.3} \right)$ <p>Where</p> <p><math>D_n</math> = nominal pipe diameter.</p> <p>The values of <math>K</math> are shown in Table 4-10 for various valves and fittings. These values were determined from combinations of literature values from references, and follow the scaling law given in the 3-K method equation.</p> <p>The values of <math>K_1</math> are mostly those of the Hooper 2-K method, and the values of <math>K_i</math> were determined from the Crane data. <math>K_d</math> values are very close to 4.0.</p> <p>The 3-K method is highly recommended because it accounts directly for the effect of both Reynolds number and fitting size on the loss coefficient. Also, it more accurately reflects the scale effect of fitting size than the 2-K method.</p>

TABLE 4-11 Excess Head Loss  $K$  Correlation for Changes in Pipe Size

Type	Fitting	Inlet $N_{Re}$	$K$ based on inlet velocity head
1	 Square reduction	$N_{Re} \leq 2500$	$K = \left[ 1.2 + \frac{160}{N_{Re}} \right] \left[ \left( \frac{D_1}{D_2} \right)^4 - 1 \right]$
		$N_{Re} > 2500$	$K = [0.6 + 0.48f_D] \left( \frac{D_1}{D_2} \right) \left[ \left( \frac{D_1}{D_2} \right)^2 - 1 \right]$
2	 Tapered reduction	All	Multiply $K$ from Type 1 by $\sqrt{\sin\left(\frac{\theta}{2}\right)}$ for $45^\circ < \theta < 180^\circ$
			Or $\left[ 1.6 \sin\left(\frac{\theta}{2}\right) \right]$ for $0^\circ < \theta < 45^\circ$
3	 Thin, sharp orifice	$N_{Re} \leq 2500$	$K = \left[ 2.7 + \left( \frac{D_1}{D_2} \right)^2 \left( \frac{120}{N_{Re}} - 1 \right) \right] \left[ 1 - \left( \frac{D_1}{D_2} \right)^2 \right] \left[ \left( \frac{D_1}{D_2} \right)^4 - 1 \right]$
		$N_{Re} > 2500$	$K = \left[ 2.7 + \left( \frac{D_1}{D_2} \right)^2 \left( \frac{4000}{N_{Re}} \right) \right] \left[ 1 - \left( \frac{D_1}{D_2} \right)^2 \right] \left[ \left( \frac{D_1}{D_2} \right)^4 - 1 \right]$
4	 Square expansion	$N_{Re} \leq 4000$	$K = 2 \left[ 1 - \left( \frac{D_1}{D_2} \right)^4 \right]$
		$N_{Re} > 4000$	$K = [1 + 0.8f_D] \left\{ \left[ 1 - \left( \frac{D_1}{D_2} \right)^2 \right]^2 \right\}$
5	 Tapered expansion	All	If $\theta > 45^\circ$ , use $K$ from Type 4, otherwise multiply $K$ from Type 4 by $\left[ 2.6 \sin\left(\frac{\theta}{2}\right) \right]$
6	 Pipe reducer	All	$K = \left[ 0.1 + \frac{50}{Re_1} \right] \left[ \left( \frac{D_1}{D_2} \right)^4 - 1 \right]$
7	 Thick orifice	All	If $L/D_2 > 5$ , use Case A and Case F; Otherwise multiply $K$ from Case D by $\left\{ 0.584 + \left[ \frac{0.0936}{(L/D)^{1.5} + 0.225} \right] \right\}$

(continued)

**TABLE 4-11—(continued)**

Type	Fitting	Inlet $N_{Re}$	K based on inlet velocity head
8	 Pipe reducer	All	Use the K for Case F

(Source: Hooper, W.B. *Chem. Eng.*, Nov 7, 1988, pp. 89–92.)

## 4.26 RESISTANCE OF VALVES

Figure 4-14b and Table 4-6 present several typical valves and connections, screwed and flanged, for a variety of sizes or internal diameters. These do not apply for mixtures of suspended solids in liquids; rather specific data for this situation are required (see [2]). Reference [4] presents data for specific valves.

Valves such as globes and angles generally are designed with changes in flow direction internally and, thereby, exhibit relatively high flow resistances. These same types of valves exhibit even greater resistances when they are throttled down from the “wide open” position for control of flow to a smaller internal flow path. For design purposes, it is usually best to assume a  $1/2$  or  $1/4$  open position, rather than wide open. Estimated  $K$  values can be determined [4] by reference to Figures 4-14a–4-18 and Tables 4-6 and 4-7.

where

$K_1$  = refers to coefficient for smaller diameter

$K_2$  = refers to coefficient for larger diameter

$\beta$  = ratio of diameters of smaller to larger pipe size

$\theta$  = angles of convergence or divergence in enlargements or contractions in pipe systems, degrees.

From [4],  $K$  values for straight-through valves, such as gate and ball (wide open), can also be calculated. These types of valves are not normally used to throttle flow, but are either open or closed.

For sudden and gradual (Note: Subscript 1 = smaller pipe; Subscript 2 = larger pipe)

$$K_2 = K_1 / \beta^4 \quad (4-80)$$

For  $\theta \leq 45^\circ$ , as enlargements:

$$K_2 = 2.6 \left[ (\sin \theta/2) (1 - \beta^2)^2 \right] / \beta^4 \quad (4-81)$$

For  $\theta \leq 45^\circ$ , as contractions:

$$K_2 = [0.8 (\sin \theta/2) (1 - \beta^2)] / \beta^4 \quad (4-82)$$

For higher resistance valves, such as globes and angles, the losses are less than sudden enlargements or contractions situations. For these reduced seat valves the resistance or loss coefficient  $K$  can be calculated as [4]:

At  $\theta \leq 180^\circ$ , for sudden and gradual enlargements:

$$K_2 = [(1 - \beta^2)^2] / \beta^4 \quad (4-83)$$

At  $\theta \leq 180^\circ$ , for gradual contraction:

$$K_2 = \left[ \left\{ 0.5 (\sin \theta/2)^{1/2} \right\} (1 - \beta^2) \right] / \beta^4 \quad (4-84)$$

The use of these equations requires some assumptions or judgment regarding the degree of opening for fluid flow. Even so, this is better than assuming a wide open or full flow condition which would result in too low a resistance to flow for the design situation.

## 4.27 FLOW COEFFICIENTS FOR VALVES, $C_v$

Flow coefficients (not resistance) for valves are generally available from the manufacturer. The  $C_v$  coefficient of a valve is defined as the flow of water at  $60^\circ\text{F}$ , in gallons per minute, at a pressure drop of one pound per square inch across the valve [4], regardless of whether the valve ultimately will be flowing liquid or gases/vapors in the plant process. (Manufacturers give values of  $C_g$  or  $C_1$ , the coefficient for gas flow, for valves flowing gas or vapor.) It is expressed:

$$C_v = 29.9 d^2 / (K)^{1/2} \quad (4-85)$$

$$C_v = Q [\rho / (\Delta P_c) (62.4)]^{1/2} \quad (4-86)$$

$$Q = C_v [\Delta P_c (62.4 / \rho)]^{1/2} \quad (4-87)$$

$$= 7.90 C_v [\Delta P_c / \rho]^{1/2} \quad (4-87a)$$

$$\Delta P = [Q / C_v]^2 [\rho / 62.4] \quad (4-88)$$

where

$d$  = internal pipe diameter, in.

$C_v$  = flow coefficient for valves; expresses flow rate in gallons per minute of  $60^\circ\text{F}$  water with 1.0 psi pressure drop across valve

$K$  = resistance (loss) coefficient

$Q$  = flow rate, gpm

$\Delta P$  = pressure drop across the control valve, psi

$\rho$  = fluid density, lb/ft<sup>3</sup>.

#### 4.28 NOZZLES AND ORIFICES [4]

The piping items shown in Figures 4-19 and 4-20 are important pressure drop or head loss items in a system and must be accounted for to obtain the total system pressure loss. For liquids (Note: The  $\Delta P$  in these equations is NOT a "loss" pressure).

$$q = C' A \sqrt{2g(144)(\Delta P)/\rho} = C' A (2gh_L)^{1/2} \quad (4-89)$$

In SI units,

$$q = C' A \sqrt{\frac{2\Delta P}{\rho}} = C' A (2gh_L)^{1/2} \quad (4-90)$$

where

$q$  =  $\text{ft}^3/\text{s}$  ( $\text{m}^3/\text{s}$ ) of fluid at flowing conditions

$C'$  = flow coefficient for nozzles and orifices.

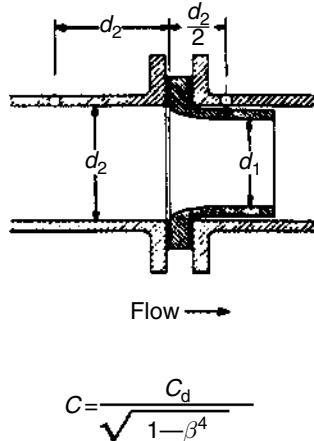
$$C' = C_d / \sqrt{1 - \beta^4}, \text{ corrected for velocity of approach} \quad (4-91)$$

Note:

$C' = C_d$  for Figures 4-19 and 4-20, corrected for velocity of approach.

$C_d$  = discharge coefficient for nozzles and orifices

$h_L$  = differential static head or pressure loss across flange taps when  $C$  or  $C'$  values come from Figures 4-19 and 4-20, ft of fluid. Taps are located one diameter upstream and 0.5 diameter down from the device.



**Example:** The flow coefficient  $C$  for a diameter ratio  $\beta$  of 0.60 at a Reynolds number of 20,000 ( $2 \times 10^4$ ) equals 1.03.

$A$  = cross-sectional area of orifice, nozzle, or pipe,  $\text{ft}^2(\text{m}^2)$

$h$  = static head loss, ft (m) of fluid flowing

$\Delta P$  = differential static loss,  $\text{lb/in.}^2(\text{N/m}^2)$  of fluid flowing, under conditions of  $h_L$  above

$\beta$  = ratio of small to large diameter orifices and nozzles and contractions or enlargements in pipes.

For discharging incompressible fluids to atmosphere, take  $C_d$  values from Figure 4-19 or 4-20 if  $h_L$  or  $\Delta P$  is taken as upstream head or gauge pressure.

For flow of compressible fluids use the net expansion factor  $Y$  (see later discussion) [4]:

$$q = YC' A [2g(144)(\Delta P)/\rho]^{1/2} \quad (4-92)$$

In SI units,

$$q = YC' A \sqrt{\frac{2\Delta P}{\rho}} \quad (4-93)$$

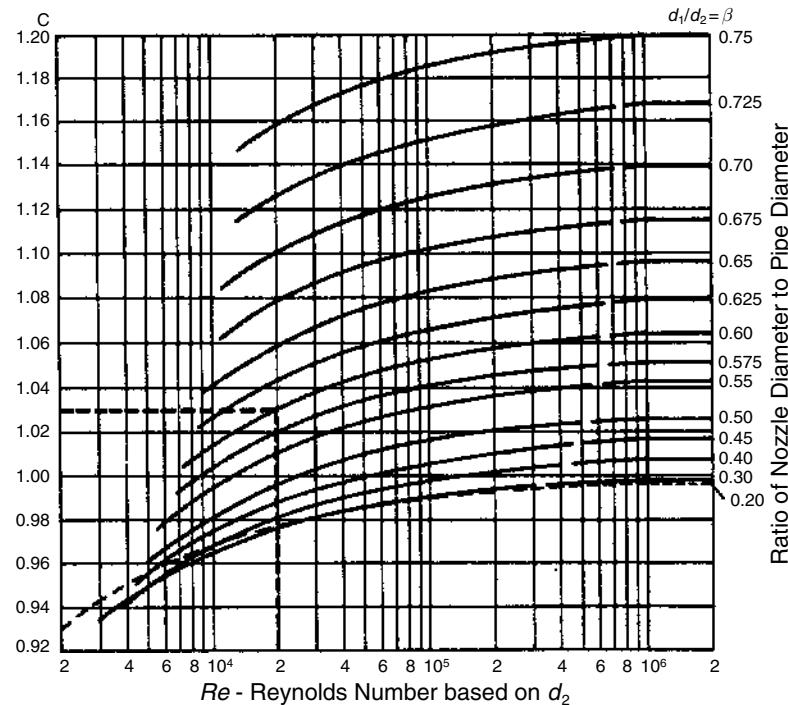
where  $Y$  = net expansion factor for compressible flow through orifices, nozzles, and pipe.

The expansion factor  $Y$  is a function of:

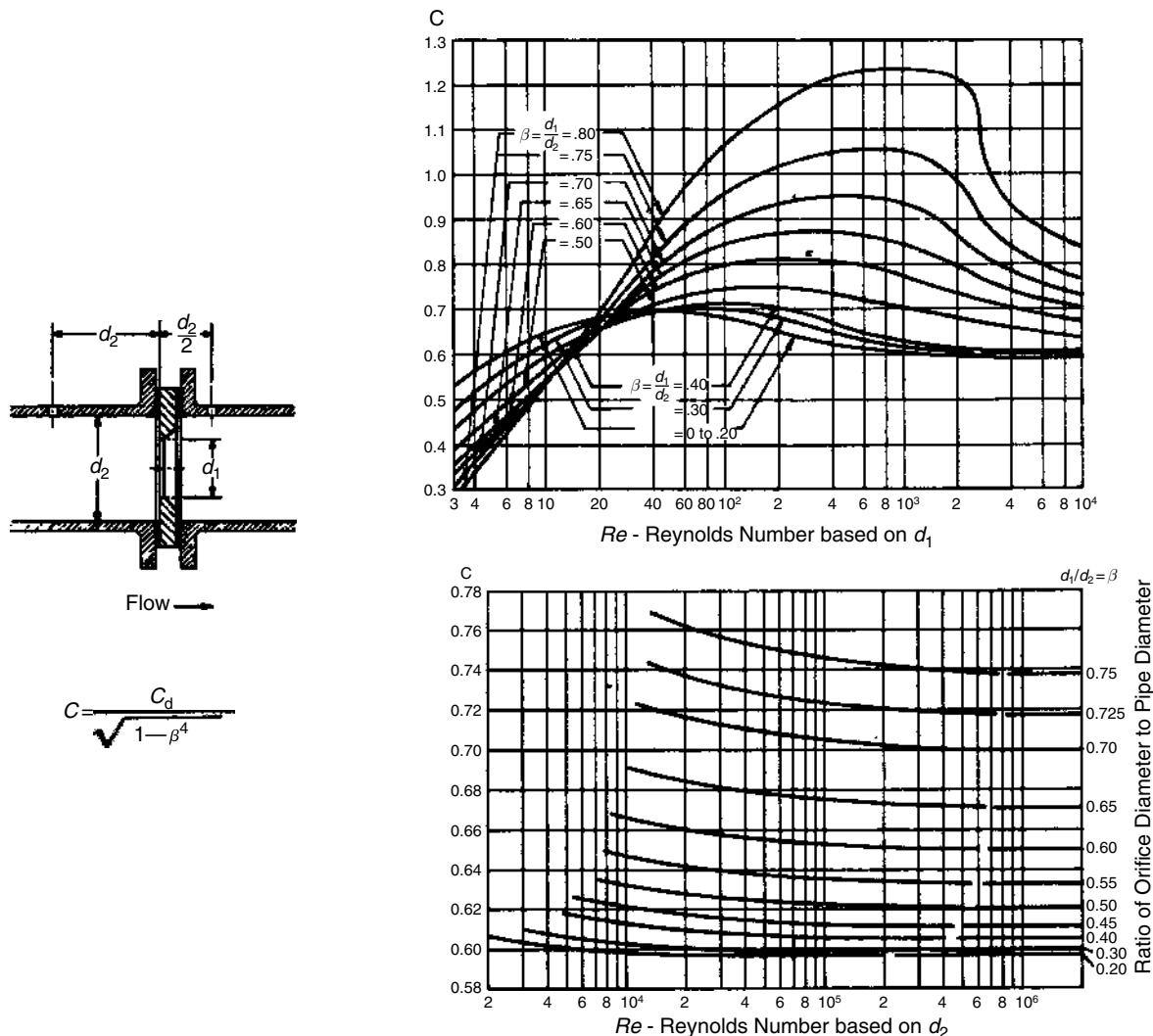
1. The specific heat ratio,  $\gamma$
2. The ratio ( $\beta$ ) of orifice or throat diameter to inlet diameter
3. Ratio of downstream to upstream absolute pressures.

$C'$  = flow coefficient from Figure 4-19 or 4-20, when discharging to atmosphere

$P$  = inlet gauge pressure (also see critical flow discussion).



**Figure 4-19** Flow coefficients "C" for nozzles.  $C$  based on the internal diameter of the upstream pipe. (By permission from Crane Co. [4]. Crane reference [18] is to Fluid Meters, American Society of Mechanical Engineers, Part 1-6th ed., 1971. Data used to construct charts. Chart not copied from ASME reference.)



**Figure 4-20** Flow coefficients “C” for square-edged orifices (By permission from Crane Co. [4], Technical Paper No. 410, Engineering Div. (1976) and Fluid Meters, Their Theory and Application Part 1, 6th ed., 1971, ASME and, Tuve, G.L. and R.E. Sprinkle, “Orifice Discharge Coefficients for Viscous Liquids,” Instruments Nov, 1993, p. 201.)

#### EXAMPLE 4-1

##### Pipe Sizing Using Resistance Coefficients, $K$

A plant decides to add a nitrogen blanket (at 5 psig) to a storage tank holding up to 25,000 gal of a hydrocarbon mixture having kerosene-like properties ( $S_g = 0.81$  and viscosity = 1.125 cP) and pumps this material into a process reactor operating at 30 psig (Figure 4-21). Liquid tank elevation is 10 ft.

The flow rate needs to be 20 gpm. Connections of pipe and valve are flanged, with the  $6 \times 90^\circ$  elbows added in the line.

*Solution*

Pump suction velocity = 2 ft/s (selected in accordance with good pump suction practice, from Table 4-12 or 4-13).

Estimated flow velocity for assumed 2 in. Sch. 40 pipe (see Appendix D-16)

$$= \frac{(20 \text{ gpm})(8.33 \text{ lb/gal})(0.81 \text{ Sp gr})}{(62.3 \times 0.81)(3.355 \text{ in.}^2)(60 \text{ s/min})/144}$$

$$= 1.91 \text{ ft/s}$$

$$\text{Velocity head, } \frac{v^2}{2g} = \frac{(1.91)^2}{2(32.2)} = 0.05664 \text{ ft of fluid}$$

$$\text{Reynolds number, } Re = \frac{50.6Q\rho}{d\mu} \quad (4-27)$$

$$= \frac{50.6(20)(0.81 \times 62.3)}{(2.067)(1.125 \text{ cP})}$$

$$Re = 21961 \text{ (turbulent)}$$

$$\epsilon/D = 0.0018/2.067 = 0.00087$$

(continued)

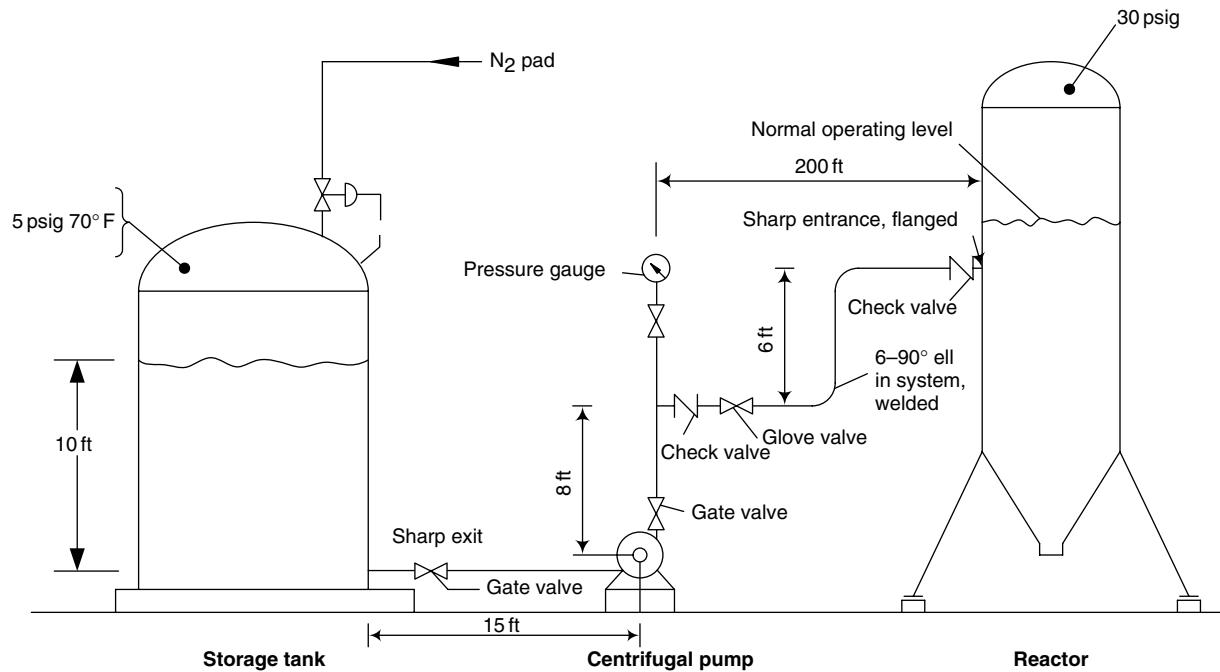


Figure 4-21 Pipe sizing using resistance coefficients,  $K$  illustration for Example 4-1.

#### EXAMPLE 4-1—(continued)

Using Chen's explicit Eq. (4-35) to calculate the friction factor,  $f_C$ :

$$\begin{aligned} A &= \frac{\varepsilon/D}{3.7} + \left( \frac{6.7}{Re} \right)^{0.9} \\ &= \left( \frac{0.00087}{3.7} \right) + \left( \frac{6.7}{21,961} \right)^{0.9} \\ &= 9.2059 \times 10^{-4} \end{aligned}$$

$$\begin{aligned} \frac{1}{\sqrt{f_C}} &= -4 \log \left\{ \frac{\varepsilon}{3.7D} - \frac{5.02}{Re} \log A \right\} \\ &= -4 \log_{10} \left\{ \frac{0.00087}{3.7} - \frac{5.02}{21,961} \log_{10} (9.2059 \times 10^{-4}) \right\} \\ &= 12.1277 \end{aligned}$$

$$f_C = 6.798934 \times 10^{-3}$$

$$\begin{aligned} f_D &= 4f_C \\ &= 4(6.798934 \times 10^{-3}) \\ &= 0.0272 \end{aligned}$$

Head loss due to friction (in pipe only—friction also in fitting and values) is calculated using Eq. (4-37).

$$\begin{aligned} h_f &= 0.0272 \left( \frac{15}{2.067/12} \right) \frac{(1.91)^2}{(2)(32.2)} \\ &= 0.134 \text{ ft of kerosene fluid, pipe friction for 15 ft} \end{aligned}$$

Loss through pump suction fittings:

- a. Square-edged inlet (tank to pipe),  $K = 0.5$ , Figure 4-14a
- b. Gate valve flanged, open, in suction line, from Table 4-6, with  $\beta = 1$ ,  $K = 8f_T$   
where  $f_T = 0.019$  (Table 4-6)  
 $K = 8 (0.019) = 0.152$

Frictional head loss (fittings and pipe entrance) is

$$\begin{aligned} h_f &= K v^2 / 2g = (0.5 + 0.152) (1.91)^2 / (2)(32.2) \\ &= 0.0370 \text{ ft fluid} \end{aligned}$$

Total suction pipe side friction loss:

$$\sum h_F = 0.134 + 0.0369 = 0.1709 \text{ ft kerosene}$$

Pressure drop per 100 ft from Eq. (4-65) is

$$\begin{aligned} \Delta P \text{ psi}/100 \text{ ft} &= 0.0216 f \rho Q^2 / d^5 & (4-65) \\ &= 0.0216 (0.0272) (0.81 \times 62.3) (20^2) / 2.067^5 \\ &= 0.314 \text{ psi}/100 \text{ ft} \end{aligned}$$

(continued)

**TABLE 4-12 Suggested Fluid Velocities in Pipe and Tubing: Liquids, Gases, and Vapors at Low/Moderate Pressure to 50 psig and 50° to 100° F**

Fluid	Suggested Trial Velocity	Pipe Material	Fluid	Suggested Trial Velocity	Pipe Material
Acetylene (observe pressure limitations)	4000 fpm	Steel	Sodium Hydroxide 0-30%	6 fps	Steel
Air, 0-30 psig	4000 fpm	Steel	30-50%	5 fps	and Nickel
Ammonia			50-73%	4	
Liquid Gas	6 fps	Steel	Sodium Chloride Solution No Solids	5 fps	Steel
Benzene	6000 fpm	Steel	With Solids	(6 min.- 15 max.)	
Bromine	6 fps	Steel		7.5 fps	Monel or nickel
Liquid Gas	4 fps	Glass	Perchlorethylene	6 fps	
Calcium chloride	2000 fpm	Glass	Steam	4,000-6,000 fpm	Steel
Carbon tetrachloride	4 fps	Steel	0-30 psi Saturated*		
Chloride (dry)	6 fps	Steel	30-150 psi Saturated or Superheated*	6,000-10,000 fpm	
Liquid Gas	5 fps	Steel, Sch. 80	150 psi up		
Chloroform	2000-5000 fpm	Steel, Sch. 80	Superheated	6,500-1,500 fpm	
Liquid Gas	6 fps	Copper and steel	*Short lines	15,000 fpm (max.)	
Ethylene Gas	2000 fpm	Steel			
Ethylene dibromide	6000 fpm	Glass	Sulfuric Acid 88-93%	4 fps	S.S.-N316, Lead
Ethylene dichloride	4 fps				Cast Iron and Steel
Ethylene glycol	6 fps	Steel			Sch. 80
Hydrogen		Steel			Steel
Liquid Gas	5 fps	Rubber Liquid	Sulfur Dioxide	4,000 fpm	
	4000 fpm	R.L., Saran	Styrene	6 fps	
Methyl chloride		Haveg	Trichlorethylene	6 fps	
Liquid Gas	6 fps	Steel	Vinyl Chloride	6 fps	
Natural gas	4000 fpm	Steel	Vinylidene Chloride	6 fps	
Oils, lubricating	6000 fpm	Steel	Water		
Oxygen	1800 fpm	Max.	Average service	3-8 (avg 6) fps	Steel
(ambient temperature)	4000 fpm	Type 304 SS	Boiler feed	4-12 fps	Steel
(Low temperature)			Pump suction lines	1-5 fps	Steel
Propylene glycol	5 fps	Steel	Steel (300 psig Max.)	7-10 fps	Steel
			Maximum economical (usual)		
			Sea and brackish water, lined pipe	5-8 fps	R.L., concrete Asphalt-line, saran-lined, transite
				3	
				5-12 fps (min.)	

Note: R. L. Rubber-lines steel.

The velocities are suggestive only and are to be used to approximate line size as a starting point for pressure drop calculations. The final line size should be such as to give an economical balance between pressure drop and reasonable velocity.

#### EXAMPLE 4-1—(continued)

Using Darby's 3-K method, Eq. (4-73)

$Re = 21961$  (turbulent)

Pipe nominal diameter = 2 in.

$$K_{\text{pipe}} = 4f_F \frac{L}{D} = f_D \frac{L}{D}$$

$$= 0.0272 \frac{15}{(2.067/12)} = 2.369$$

$$\text{Sum } K\text{'s} = 0.675 + 2.369 = 3.044$$

Fittings       $n$      $K_1$      $nK_1$      $K_i$      $nK_i$      $K_d$      $K_f$

Gate valve      1    300    300    0.037    0.037    3.9    0.168  
 Square-edged inlet      0.507

$$K_f = (160/Re + K_\infty)$$

$$K_\infty = 0.5$$

$$\text{Total} \quad 0.675$$

Frictional head loss (fittings and pipe entrance) is:

$$h_f = Kv^2/2g = (0.507 + 0.168)(1.91)^2/(2)(32.174)$$

$$= 0.038 \text{ ft of fluid}$$

(continued)

**EXAMPLE 4-1—(continued)****TABLE 4-13 Typical Design\* Velocities for Process System Applications**

Service	Velocity (ft/s)
Average liquid process	4–6.5
Pump suction (except boiling)	1–5
Pump suction, boiling	0.5–3
Boiler feed water (discharge, pressure)	4–8
Drain lines	1.5–4
Liquid to reboiler (no pump)	2–7
Vapor–liquid mixture out reboiler	15–30
Vapor to condenser	15–80
Gravity separator flows	0.5–1.5

\* To be used as guide; pressure drop and system environment govern final selection of pipe size. For heavy and viscous fluids, velocities should be reduced to about half the values shown. Fluids not to contain suspended solid particles.

Using Hooper's 2-K method

$$K_f = \frac{K_1}{Re} + K_\infty \left( 1 + \frac{1}{ID_{\text{inch}}} \right) \quad (4-71)$$

Pipe Internal diameter (ID) = 2.067 in.

Fittings	n	K <sub>1</sub>	nK <sub>1</sub>	K <sub>∞</sub>	nK <sub>∞</sub>	K <sub>f</sub>
Gate valve	1	300	300	0.1	0.1	0.162
Square-edged inlet						0.507
$K_f = (160/Re + K_\infty)$						
$K_\infty = 0.5$						
Total						0.669

The percentage error difference in the loss coefficient between the 2-K method and 3-K methods is  $(0.675 - 0.669)/0.669 \times 100 = 0.90\%$ . However, for the Crane method, the percentage error difference in the loss coefficient is  $(0.6752 - 0.652)/0.652 \times 100 = 3.5\%$  lower. (Note: The differences are greater for lower Reynolds numbers).

Applying the energy balance equation (In English Engineering units):

$$\frac{q}{g_c} + 144 \frac{P_1}{\rho} + z_1 \frac{g}{g_c} + \frac{v_1^2}{2g_c} = 144 \frac{P_2}{\rho} + z_2 \frac{g}{g_c} + \frac{v_2^2}{2g_c} + \frac{e_f}{g_c} + \frac{w}{g_c} \quad (4-69)$$

$g_c$  = conversion factor, 32.174 lb<sub>m</sub>/lb<sub>f</sub>. (ft/s<sup>2</sup>)  
 $g$  = acceleration due to gravity, 32.174 ft/s<sup>2</sup>

The general energy balance equation is:

$$\frac{P_2 - P_1}{\rho} + g(z_2 - z_1) + \frac{1}{2}(v_2^2 - v_1^2) + e_f + w = 0$$

where

$$e_f = (u_2 - u_1) - q \text{ for an incompressible fluid}$$

$u$  = internal energy per unit mass

$q$  = heat input into the system.

Inlet pressure  $P_2$  to the suction of the pump.

Assuming that there is no net heat transfer and no work is done by the system,  $q = w = 0$

$$P_2 = P_1 + \frac{\rho}{144} (z_1 - z_2) \frac{g}{g_c} + \frac{\rho}{2g_c} \left( \frac{1}{144} \right) (v_1^2 - v_2^2) - \frac{\rho}{144} \frac{(e_f)}{g_c}$$

The pressure,  $P_2$  (at the suction) is

$$P_2 = P_1 + \frac{\rho}{144} (z_1 - z_2) \frac{g}{g_c} + \frac{\rho}{2g_c} \left( \frac{1}{144} \right) (0 - v_2^2) - \frac{\rho}{144} \frac{(e_f)}{g_c}$$

$$= (5 \text{ psig} + 14.69 \text{ psia}) + \text{static head} - \text{kinetic energy}$$

$$\text{head} - \text{friction loss in suction line.}$$

$$e_f = \left( 4f_F \frac{L}{D} + \sum K_f \right) \frac{v^2}{2}$$

$$\text{or}$$

$$e_f = \left( f_D \frac{L}{D} + \sum K_f \right) \frac{v^2}{2}$$

$$e_f = \left( 0.0272 \left( \frac{15}{2.067/12} \right) + 0.662 \right) (0.05664)$$

$$= 5.552 \text{ ft}^2/\text{s}^2$$

$$\frac{\rho e_f}{144 g_c} = \frac{(0.81 \times 62.3) (5.552)}{(144) (32.174)} \left\{ \frac{\text{lb}_m}{\text{ft}^3} \cdot \frac{\text{ft}}{\text{s}^2} \cdot \frac{1}{\frac{\text{lb}_m}{\text{lb}_f} \cdot \frac{\text{ft}}{\text{s}^2}} \cdot \frac{\text{ft}^2}{\text{in}^2} \right\}$$

$$= 0.0605 \text{ lb}_f/\text{in}^2$$

Static head =  $(z_1 - z_2) = 10 \text{ ft}$

$$= \frac{(0.81 \times 62.3)}{144} (10 - 0) \left\{ \frac{\text{lb}_m}{\text{ft}^3} \cdot \frac{\text{ft}}{\text{s}^2} \cdot \frac{1}{\frac{\text{lb}_m}{\text{lb}_f} \cdot \frac{\text{ft}}{\text{s}^2}} \cdot \frac{\text{ft}^2}{\text{in}^2} \right\}$$

$$= 3.504 \text{ lb}_f/\text{in}^2$$

$$\text{Kinetic energy} = \frac{\rho}{2g_c} \left( \frac{1}{144} \right) (v_1^2 - v_2^2)$$

$$= \frac{(0.81 \times 62.3)(0 - 1.91^2)}{(2)(144)(32.174)} \left\{ \frac{\text{lb}_m}{\text{ft}^3} \cdot \frac{\text{ft}^2}{\text{s}^2} \cdot \frac{1}{\frac{\text{lb}_m}{\text{lb}_f} \cdot \frac{\text{ft}}{\text{s}^2}} \cdot \frac{\text{ft}^2}{\text{in}^2} \right\}$$

$$= 0.0605 \text{ lb}_f/\text{in}^2$$

$$P_2 = 19.69 \text{ psia} (4.92 \text{ psig}) + 3.504 - 0.0199 - 0.0605$$

$$= 8.344 \text{ psig.}$$

(continued)

**EXAMPLE 4-1—(continued)**

The total pressure drop ( $\Delta P_{\text{Total}}$ ) at the suction:

Suction (psig)	
Vessel pressure	5.0
Static-head pressure	3.504
Friction loss	-0.0605
$\Delta P_{\text{Total}}$	8.4435 psi

**Pump Discharge Line Sizing (only)**

The pump discharge can flow at a higher velocity than the suction line, due in part to NPSH conditions on the suction side of any pump (which are not considered directly in these pipe-sizing calculations). NPSH and its effect on pump-sizing are reviewed in Chapter 5.

From Table 4-12, select 6 ft/s as design velocity for estimating pipe size.

For 20 gpm, cross-section area for flow required.

$$\begin{aligned} A &= \frac{20 \text{ gpm}}{7.48 \text{ gal./ft}^3 (60 \text{ s/min}) (6 \text{ ft/s})} \\ &= 0.007427 \text{ ft}^2 \\ &= (0.007427)(144) = 1.0695 \text{ in.}^2 \end{aligned}$$

Pipe diameter is

$$D = \sqrt{4 \text{Area}/\pi}, \text{ in.}$$

$$\begin{aligned} D &= \sqrt{(4)(1.0695)/\pi} \\ &= 1.167 \text{ in.} \end{aligned}$$

(Note: Usually do not select this size. Could go to 1½ in. However, velocity would be even slower.)

The closest standard Sch. 40 pipe size is 1¼ in. (ID = 1.38 in.)

$$\text{ID in ft} = \frac{1.38}{12} = 0.115 \text{ ft}$$

$$\begin{aligned} \text{Pipe cross section area, } A &= \frac{\pi (\text{ID})^2}{4}, \text{ ft}^2 \\ &= \frac{\pi (0.115^2)}{4} = 0.010387 \text{ ft}^2 \end{aligned}$$

Actual velocity is  $Q = v \cdot A$   
where

$A$  = pipe cross section area,  $\text{ft}^2$

$Q$  = volumetric flow rate,  $\text{ft}^3/\text{s}$

$v$  = fluid velocity,  $\text{ft/s}$ .

$$\begin{aligned} v &= \frac{Q}{A} = \frac{20}{(0.010387)(7.48)(60)} \left\{ \frac{\text{gal}}{\text{min}} \cdot \frac{1}{\text{gal}} \cdot \frac{1}{\text{ft}^2} \cdot \frac{\text{min}}{\text{s}} \right\} \\ &= 4.29 \text{ ft/s} \end{aligned}$$

$$\text{Velocity head, } \frac{v^2}{2g} = \frac{(4.29)^2}{2(32.174)} = 0.2860 \text{ ft of fluid}$$

$$\begin{aligned} \text{Reynolds number, } Re &= \frac{50.6 Q \rho}{d \mu} \\ &= 50.6 (20) (0.81 \times 62.3) (1.38) (1.125) \\ &= 32,894 \text{ (turbulent)} (\approx 33,000) \end{aligned} \quad (4-27)$$

For 1¼ in. (ID = 1.38 in.),  $\varepsilon/D = 0.0013$

Using Chen's explicit Eq. (4-35) to calculate the friction factor,  $f_C$ :

$$\begin{aligned} A &= \frac{\varepsilon/D}{3.7} + \left( \frac{6.7}{Re} \right)^{0.9} \\ &= \left( \frac{0.0013}{3.7} \right) + \left( \frac{6.7}{32894} \right)^{0.9} \\ &= 8.2785 \times 10^{-4} \\ \frac{1}{\sqrt{f_C}} &= -4 \log \left\{ \frac{\varepsilon}{3.7D} - \frac{5.02}{Re} \log A \right\} \\ &= -4 \log_{10} \left\{ \frac{0.0013}{3.7} - \frac{5.02}{32894} \log_{10} (8.2785 \times 10^{-4}) \right\} \\ &= 12.3411 \\ f_C &= 6.5658 \times 10^{-3} \\ f_D &= 4f_C \\ &= 4 (6.5658 \times 10^{-3}) \\ &= 0.0263 \end{aligned}$$

Pressure drop per 100 ft from Eq. (4-65) is

$$\begin{aligned} \Delta P \text{ psi/100 ft} &= 0.0216 f \rho Q^2 / d^5 \\ &= 0.0216 (0.0263) (0.81 \times 62.3) (20^2) / 1.38^5 \\ &= 2.291 \text{ psi/100 ft.} \end{aligned} \quad (4-65)$$

Friction loss due to straight pipe:

$$h_f = \frac{e_f}{g} = \left( f_D \frac{L}{D} \right) (v^2/2g) \quad (4-37)$$

$$D = \text{pipe, ID, in. ft} = 1.38/12 = 0.1150 \text{ ft}$$

$$h_f = 0.0263 \left( \frac{8+6+200}{0.115} \right) (0.286)$$

= 14.0 ft of kerosene flowing (pipe only)

Loss coefficient through discharge fittings, valves, connections, using  $K$  factors in Table 4-6:

$$\begin{aligned} K_{\text{pipe}} &= 4 f_F L/D = f_D L/D \\ &= 0.0263 (8+6+200)/0.115 = 48.94 \end{aligned}$$

(continued)

**EXAMPLE 4-1—(continued)**

2 check valves (swing threaded),  $K = 100 f_T$   
where

$$f_T = 0.022 \text{ (Table 4-6)}$$

$$K = 100(0.022) = 2.2 = 4.4 \text{ (for 2)}$$

1 globe valve (open),  $\beta = 1$ ,  $K = 340 f_T = 340 (0.022) = 7.48$

6 × 90° elbows,  $r/d = 1.88/1.38 = 1.36$

$$K = 30 f_T = 30 (0.022) = 0.66$$

For 6:  $6 \times 0.66 = 3.96$

1 sharp-edged entrance (sudden enlargement),  $K = 1.0$

1 gate valve (open),  $\beta = 1.0$ ,  $K = 8 f_T = 8 (0.022) = 0.176$

Summation:

$$K = [4.4 + 7.48^* + 3.96 + 0.176^* + 1.0] = 17.016$$

For fittings:

$$\text{Then, } h = K \frac{v^2}{2g} = (17.016)(0.286) = 4.867 \text{ ft kerosene}$$

for 1 1/4 in. (ID = 1.38 in.),  $\epsilon/D = 0.0013$

Total friction loss for discharge side pump due to friction:

$$h = 14 + 4.867 = 18.867 \text{ ft fluid kerosene}$$

$$\Delta P_f = \frac{h \times \text{SpGr}}{2.31}$$

$$\Delta P_f = (18.867)(0.81)/2.31 = 6.61 \text{ psi}$$

Using Darby's 3-K method, Eq. (4-73)

Pipe nominal diameter  $D_n = 1.25$  in.

Fittings	n	K <sub>1</sub>	nK <sub>1</sub>	K <sub>i</sub>	nK <sub>i</sub>	K <sub>d</sub>	K <sub>f</sub>
90° ell (r/D = 1.5)	6	800	4800	0.071	0.426	4.2	2.245
Gate valve	1	300	300	0.037	0.037	3.9	0.181
Globe valve	1	1500	1500	1.70	1.70	3.6	7.469
Check valve	2	1500	3000	0.46	0.92	4.0	4.453
No exit loss if discharge is below liquid	0					0.0	
Total							14.348

$$K_{\text{pipe}} = 4f_F \frac{L}{D} = f_D \frac{L}{D}$$

$$= 0.0263 \frac{(8 + 6 + 200)}{(1.38/12)} = 48.94$$

\*Threaded, from Table 4-6.

$$\text{Sum } K's = 14.348 + 48.94 = 63.29$$

$$h_F = Kv^2/2g = (14.348)(4.29)^2/(2)(32.174)$$

$$= 4.10 \text{ ft of fluid}$$

Total friction loss at the discharge side of the pump (i.e. straight lengths of pipe plus fittings) is

$$h_f = 14 + 4.10 = 18.1 \text{ ft fluid kerosene}$$

$$\Delta P_f = \frac{h_f \times \text{SpGr}}{2.31} \quad (4-18)$$

$$\Delta P_f = (18.1)(0.81)/2.31 = 6.35 \text{ psi}$$

Using Hooper's 2-K method

$$K_f = \frac{K_1}{Re} + K_\infty \left( 1 + \frac{1}{ID_{\text{inch}}} \right) \quad (4-71)$$

Pipe ID = 1.38 in.

Fittings	n	K <sub>1</sub>	nK <sub>1</sub>	K <sub>∞</sub>	nK <sub>∞</sub>
90° ell (r/D = 1.5)	6	800	4800	0.2	1.2
Gate valve	1	300	300	0.10	0.1
Globe valve	1	1500	1500	4.0	4.0
Check valve	2	1500	3000	1.5	3.0
No exit loss if discharge is below liquid	0				
Total				9600	8.3

Substituting in Eq. (4-71)

$$K_f = \frac{9600}{32,894} + 8.30 \left( 1 + \frac{1}{1.38} \right)$$

$$= 14.606$$

$$h_F = Kv^2/2g = (14.606)(4.29)^2/(2)(32.174)$$

$$= 4.18 \text{ ft of fluid}$$

Using the energy balance equation at the pump discharge (i.e., between the pump outlet and the reactor):

$$\frac{q}{g_c} + 144 \frac{P_1}{\rho} + z_1 \frac{g}{g_c} + \frac{v_1^2}{2g_c} = 144 \frac{P_2}{\rho} + z_2 \frac{g}{g_c} + \frac{v_2^2}{2g_c} + \frac{e_f}{g_c} + \frac{w}{g_c} \quad (4-69)$$

Since the pipe configuration is the same,  $v_1 = v_2$ , and assuming that there is no net heat transfer or work done by the system,  $q = w = 0$

(continued)

**EXAMPLE 4-1—(continued)**

The pressure drop  $\Delta P$  between the pump discharge and the reactor is

$$-\Delta P = (P_1 - P_2) = \frac{\rho}{144} (z_2 - z_1) \frac{g}{g_c} + \frac{\rho}{144} \frac{e_f}{g_c}$$

where

$P_1$  = pressure at the pump discharge

$P_2$  = pressure of the reactor = 30 psig = 44.69 psia

$z_1$  = datum level at pump discharge

$z_2$  = piping elevation of the reactor (14 ft)

$g$  = acceleration due to gravity, 32.174 ft/sec<sup>2</sup>

$g_c$  = conversion factor, 32.174 lb<sub>m</sub>/lb<sub>f</sub>. (ft/sec<sup>2</sup>)

The pressure drop  $\Delta P$  is

$$-\Delta P = (P_1 - P_2) = \frac{\rho}{144} (z_2 - z_1) \frac{g}{g_c} + \frac{\rho}{144} \frac{e_f}{g_c}$$

The energy dissipated between the pump discharge and the reactor,  $e_f$  is

$$\begin{aligned} e_f &= \left( f_D \frac{L}{D} + \sum K_f \right) \frac{v^2}{2} \\ &= \left( (0.0263) \frac{214}{0.115} + 14.348 \right) \left( \frac{4.29^2}{2} \right) \\ &= 582.387 \text{ ft}^2/\text{sec}^2 \\ -\Delta P &= (P_1 - P_2) = \frac{(0.81)(62.3)}{144} (14 - 0) \\ &\quad + \frac{(0.81)(62.3)(582.387)}{(144)(32.174)} \\ &= 4.906 + 6.343 \\ &= 11.249 \text{ psi.} \end{aligned}$$

The total pressure drop ( $\Delta P_{\text{Total}}$ ) at the discharge:

Discharge (psig)	
Vessel pressure	30.0
Static-head pressure	4.906
Friction loss	6.343
$\Delta P_{\text{Total}}$	41.249 psi

Tables 4-14a and b show the computer results of frictional pressure drop calculations of the suction and discharge lines of Example 4-1. The Excel spreadsheet program (Example 4-1.xls) provides the pressure drop calculations of Example 4-1 for the suction and discharge lines.

**TABLE 4-14a Computer Results of Example 4-1 (Suction side)**

PRESSURE DROP CALCULATION IN A PIPE LINE

PIPE INTERNAL DIAMETER, inch:	2.067
MASS FLOW RATE, lb/h:	8089.46
VOLUMETRIC FLOW RATE, gal/min.:	20.00
FLUID VISCOSITY, cP:	1.1250
FLUID DENSITY, lb/ft. <sup>3</sup> :	50.4600
FLUID VELOCITY, ft/sec.:	1.910
VELOCITY HEAD LOSS DUE TO FITTINGS:	.6693
EQUIVALENT LENGTH OF PIPE, ft.:	4.2397
ACTUAL LENGTH OF PIPE, ft.:	15.0000
TOTAL LENGTH OF PIPE, ft.:	19.240
REYNOLDS NUMBER:	21981.
PIPE ROUGHNESS, ft.:	.0001500
DARCY FRICTION FACTOR:	.0272
EXCESS HEAD LOSS, ft.:	.172
PIPE PRESSURE DROP/100 ft, psi/100 ft:	.314
OVERALL PRESSURE DROP OF PIPE, psi:	.060

**TABLE 4-14b Computer Results of Example 4-1 (Discharge side)**

PRESSURE DROP CALCULATION IN A PIPE LINE

PIPE INTERNAL DIAMETER, inch:	1.380
MASS FLOW RATE, lb/h:	8089.46
VOLUMETRIC FLOW RATE, gal/min.:	20.00
FLUID VISCOSITY, cP:	1.1250
FLUID DENSITY, lb/ft. <sup>3</sup> :	50.4600
FLUID VELOCITY, ft/sec.:	4.285
VELOCITY HEAD LOSS DUE TO FITTINGS:	15.1109
EQUIVALENT LENGTH OF PIPE, ft.:	66.1510
ACTUAL LENGTH OF PIPE, ft.:	214.0000
TOTAL LENGTH OF PIPE, ft.:	280.151
REYNOLDS NUMBER:	32923.
PIPE ROUGHNESS, ft.:	.0001500
DARCY FRICTION FACTOR:	.0263
EXCESS HEAD LOSS, ft.:	18.244
PIPE PRESSURE DROP/100 ft, psi/100 ft:	2.288
OVERALL PRESSURE DROP OF PIPE, psi:	6.411

**EXAMPLE 4-2****Case Study**

Figure 4-22 shows the process flow diagram (PFD) of a crude distillation unit, debutanizer section, and Figure 4-23 further illustrates the debutanizer column C-1007, overhead gas line 8"-P10170-3101C-P to an air cooler condenser E-1031, accumulator V-1008, reflux pump P1017A/B, suction line 8"-P10174-3101C and discharge line 6"-P10176-3101C respectively. The operating pressure of the debutanizer C-1007 is 16.59 barg. The condensates from E-1031 are collected in the debutanizer accumulator V-1008 and separated from sour water, which is returned to vessel V-1002. The liquefied petroleum gas (LPG) from the accumulator V-1008 operating at 12.69 barg is refluxed to the top of the debutanizer C-1007 via a centrifugal pump P1017A/B.

Determine the frictional losses ( $\Delta P_f$ ) of both the suction and discharge of pump P1017 A/B carrying liquefied petroleum gas (LPG) at 1432.24 tonne/day from the accumulator vessel V-1008 to the suction of pump P1017 A/B, and from the discharge of pump P1017 A/B to the debutanizer unit C-1007. Other data obtained from the piping isometrics, piping data sheets, and fluid characteristics are given below.

Operating conditions:

Pressure at the accumulator = 12.69 barg

Vapor pressure = 14.17 bara

Temperature = 67°C

Viscosity of LPG,  $\mu = 0.23 \text{ cSt}$

Density of LPG,  $\rho = 488 \text{ kg/m}^3$

Suction Side of pump P1017A

Pipe line designation: 8"-P10174-3101C

Pipe size  $D = 8 \text{ in. Sch. 40}$

Pipe Length  $L = 19 \text{ m}$

Pipe elevation,  $\Delta z = 7.0 \text{ m}$

Total length of straight pipe,  $L = 26 \text{ m}$

Fittings	Number
90° ell ( $r/R = 1.5$ )	10
Ball valve	1
Gate valve	1
Tees (flow thru branch)	1
Reducer	1
Square edged inlet	1

Discharge side of pump P1017A

Pipe line designation: 6"-P10176-3101C

Pipe size = 6 in. Sch. 40

Straight length of pipe = 91 m

Pipe elevation,  $\Delta z_{\text{disch}} = 43 \text{ m}$

Total length of straight pipe =  $91 + 43 = 134 \text{ m}$

Fittings	Number
45° ell ( $r/R = 1.5$ )	6
90° ell ( $r/R = 1.5$ )	40
Ball valve	6
Globe valve	1
Check valve	2
Lift check valve	3
Tees (flow thru branch)	5

**Solution****The Frictional losses from the accumulator vessel V-1008 to the suction of pump P1017A/B**

LPG flow rate = 1432.24 tonne/day =  $122.5 \text{ m}^3/\text{h} = 59780 \text{ kg/h}$

The suction specification line is 8"-P10174-3101C

The viscosity of LPG in cP is

$$1 \text{ cSt} = 10^{-6} \text{ m}^2/\text{s}$$

Kinematic viscosity,  $v$ , is

$$v = 0.23 \text{ cSt} = 0.23 \times 10^{-6} \text{ m}^2/\text{s}$$

Dynamic viscosity = Kinematic viscosity × density

That is  $\mu = v\rho$

where

$\mu$  = fluid dynamic viscosity

$\rho$  = fluid density

$v$  = fluid kinematic viscosity.

$$\begin{aligned} \mu &= (0.23)(10^{-6})(488) \left( \frac{\text{kg}}{\text{m}^3} \cdot \frac{\text{m}^2}{\text{s}} \right) \\ &= 0.112 \times 10^{-3} \frac{\text{kg}}{\text{ms}} = 0.112 \text{ cP} \end{aligned}$$

Internal diameter of an 8" Sch 40 pipe, ID = 202.7 mm

Pipe area is

$$A = \frac{\pi d^2}{4} = \frac{\pi (0.2027)^2}{4} = 0.03227 \text{ m}^2$$

$$\begin{aligned} \text{Fluid velocity, } v &= \frac{Q}{\text{Area}} = \frac{122.5}{3600} \times \frac{1}{0.03227} \left\{ \frac{\text{m}^3}{\text{h}} \cdot \frac{\text{h}}{\text{s}} \cdot \frac{1}{\text{m}^2} \right\} \\ &= 1.054 \text{ m/s} \end{aligned}$$

The fluid's (LPG) Reynolds number using Eq. (4-28) is given by:

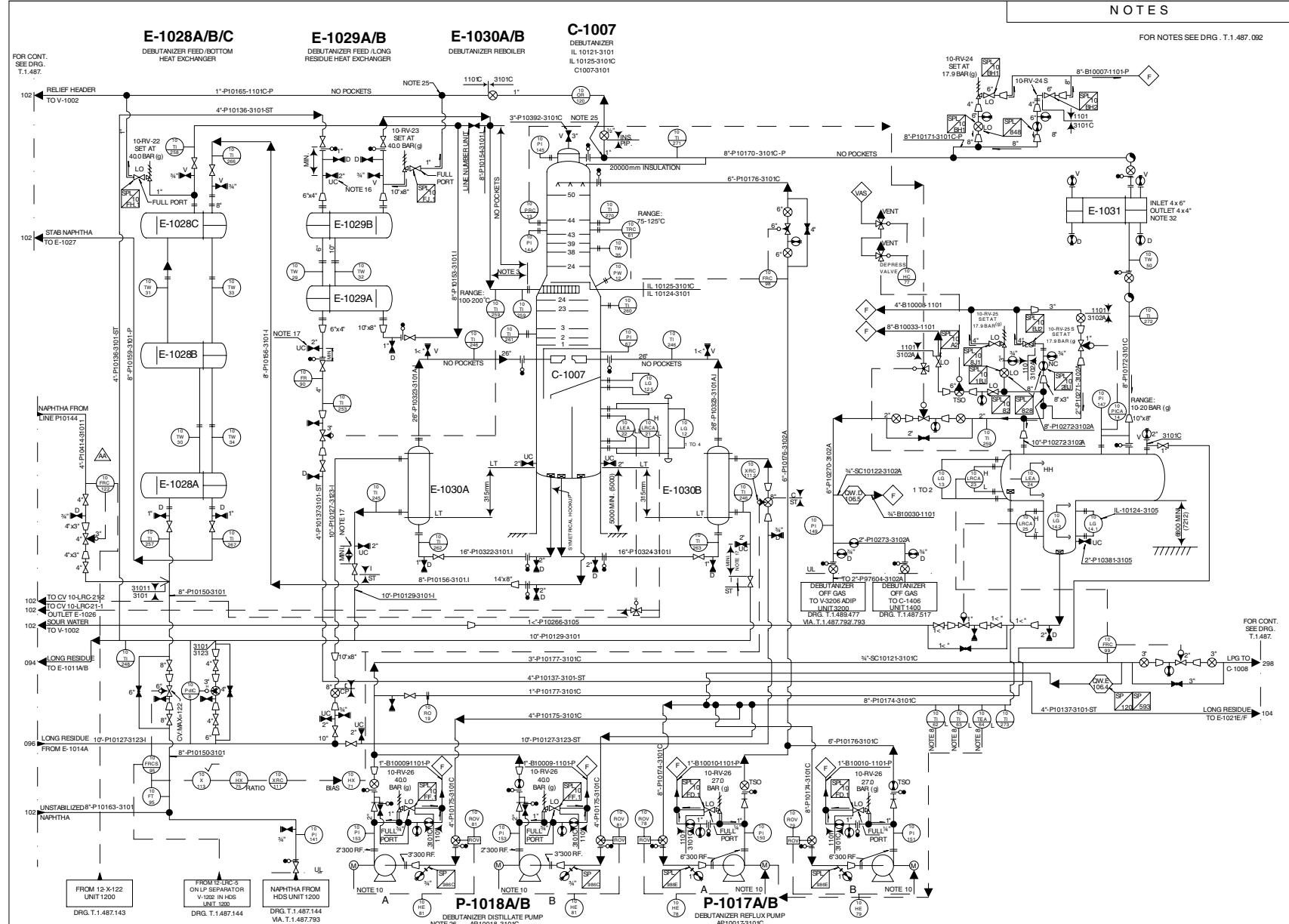
$$\begin{aligned} Re &= 354 \frac{597,800}{(202.7)(0.112)} \\ &= 932,153 \\ &= 9.3 \times 10^5 \text{ (turbulent)} \end{aligned}$$

Pipe relative roughness  $\epsilon/d = 0.046/202.7 = 0.000227$

Using Chen's explicit Eq. (4-35) to calculate the friction factor,  $f_C$ :

$$\begin{aligned} A &= \frac{\epsilon/D}{3.7} + \left( \frac{6.7}{Re} \right)^{0.9} \\ &= \left( \frac{0.000227}{3.7} \right) + 2 \left( \frac{6.7}{932153} \right)^{0.9} \\ &= 8.4844 \times 10^{-5} \end{aligned}$$

(continued)



**Figure 4-22** Process flow diagram of the debutanizer unit 1000 for Example 4-2.

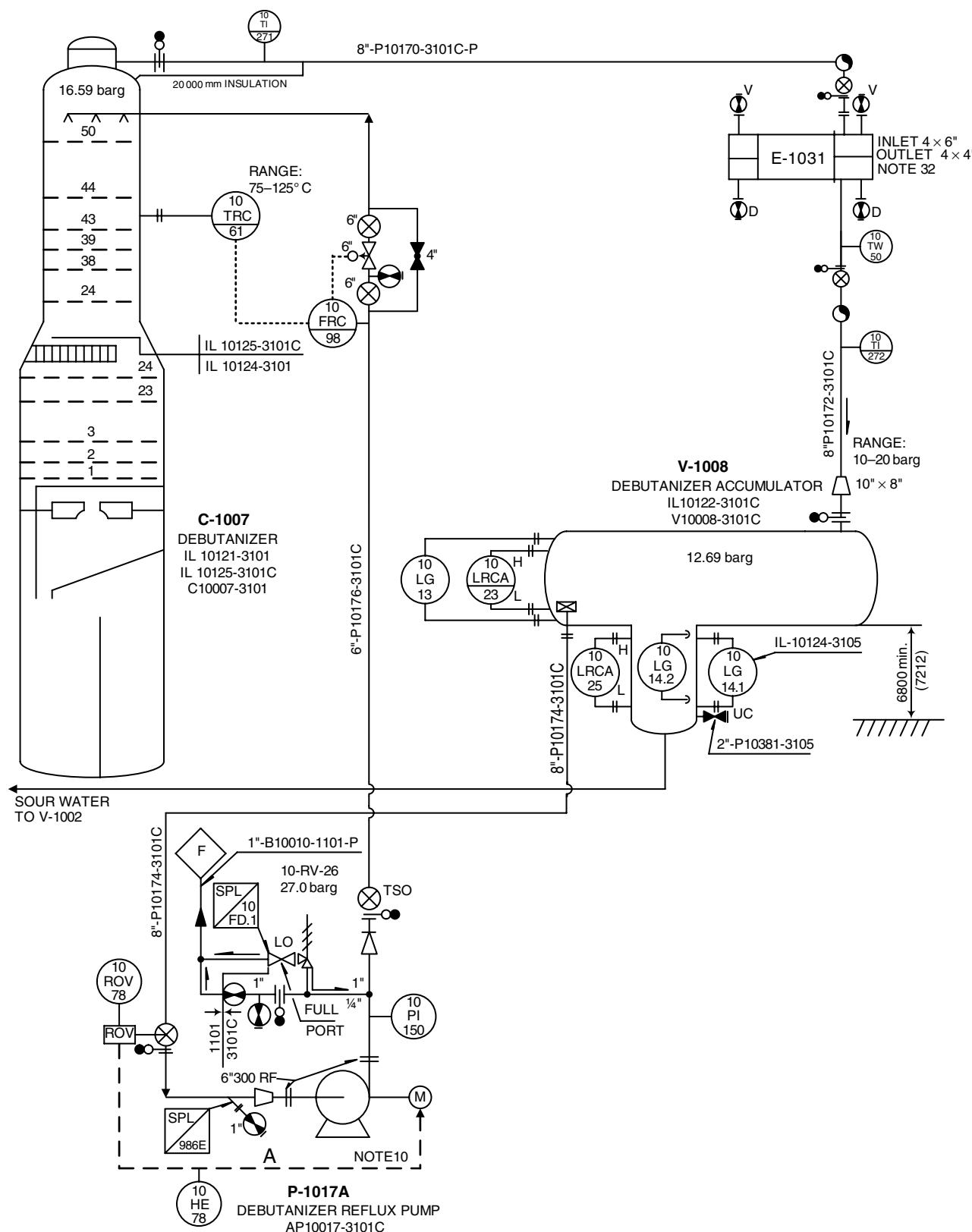


Figure 4-23 Process flow diagram of the debutanizer-pump-accumulator unit 1000 for Example 4-2.

**EXAMPLE 4-2—(continued)**

Substituting in Eq. (4-35) the friction factor  $f_C$  is

$$\begin{aligned}\frac{1}{\sqrt{f_C}} &= -4 \log_{10} \left[ \frac{0.000227}{3.7} - \frac{5.02}{932153} \log_{10}(8.4844 \times 10^{-5}) \right] \\ &= 16.3179 \\ &= 3.7377 \times 10^{-3} \\ f_D &= 4f_C \\ &= 4(3.7555 \times 10^{-3}) \\ &= 0.0150\end{aligned}$$

Pressure drop ( $\Delta P_{100}$ ) per 100 m from Eq. (4-66) is

$$\begin{aligned}\Delta P \text{ bar}/100 \text{ m} &= 225 f_D \rho Q^2 / d^5 \quad (4-66) \\ &= 225 (0.0150) (488) (2041.7)^2 / 202.7^5 \\ &= 0.02 \text{ bar}/100 \text{ m}\end{aligned}$$

In Eq. (4-69a), the frictional pressure drop for the plain pipe length (e.g., the first term on the right side of the equation):

$$\Delta P_f = f_D \left( \frac{L}{D} \right) \rho \frac{v^2}{2} \quad (4-41)$$

where

$$\left( \frac{L}{D} \right) = \frac{26 \text{ m}}{0.2027 \text{ m}} = 128.27$$

and

$$\rho \frac{v^2}{2} = \frac{(488)(1.054)^2}{2} = 271.06 \text{ N/m}^2$$

Frictional loss of the straight pipe,  $\Delta P_f$ , is

$$\begin{aligned}\Delta P_f &= (0.0150)(128.27)(271.06) \quad (4-41) \\ &= 521.53 \text{ N/m}^2\end{aligned}$$

The head loss  $h_f$  is

$$\begin{aligned}h_f &= f_D \left( \frac{L}{D} \right) \frac{v^2}{2g} \quad (4-37) \\ &= (0.0150)(128.27) \frac{1.054^2}{(2 \times 9.81)} \\ &= 0.109 \text{ m}\end{aligned}$$

The second term in Eq. (4-69a) represents the total loss coefficient  $K$  for all the fittings. Values of  $K$  for the 2- $K$  method (Hooper's) or 3- $K$  method (Darby's) are given in Tables 4-8 and 4-9 respectively.

Using Darby's 3- $K$  method as follows:

Nominal pipe size of an 8" pipe line is = 203.2 mm

$Re = 932,153$

The 3- $K$  (Darby's) method for determining the loss coefficient for pipe fittings

$$K_f = \frac{K_1}{Re} + K_i \left[ 1 + K_d \left( \frac{25.4}{D_{n,\text{mm}}} \right)^{0.3} \right] \quad (4-74)$$

Loss coefficient  $K$  for reduction in pipe size is

$$Re > 2500$$

$$\begin{aligned}K_f &= [0.6 + 0.48f] \left[ \frac{1-\beta^2}{\beta^4} \right] \left[ \sin\left(\frac{\theta}{2}\right) \right]^{1/2} \\ \beta &= \frac{D_1}{D_2} = \frac{154.1}{202.7} = 0.760\end{aligned}$$

Assume  $\theta = 45^\circ$

$$\begin{aligned}K_f &= [0.6 + (0.48)(0.0147)] \left[ \frac{1-0.76^2}{0.76^4} \right] \left[ \sin\left(\frac{45^\circ}{2}\right) \right]^{1/2} \\ &= 0.4759\end{aligned}$$

Fittings	n	$K_1$	$nK_1$	$K_i$	$nK_i$	$K_d$	$K_f$
90° ell ( $r/D = 1.5$ )	10	800	8000	0.071	0.71	4.2	2.317
Ball valve ( $\beta = 1$ )	1	300	300	0.017	0.017	4.0	0.054
Gate valve ( $\beta = 1$ )	1	300	300	0.037	0.037	3.9	0.115
Tees (Flow thru Branch)	1	800	800	0.14	0.14	4.0	0.441
Reducer	1						0.476
Square edged inlet $K_f =$ $(160/Re + K_\infty)$	$K_\infty = 0.5$						0.5
Total							3.357

$$\begin{aligned}\text{The pressure loss due to fittings } \Delta P_{f(\text{fittings})} &= K_{\text{Total}} \frac{\rho v^2}{2} \quad (4-67) \\ \Delta P_{f(\text{fittings})} &= (3.903)(271.06) \\ &= 1060.05 \text{ N/m}^2\end{aligned}$$

Head loss due to fittings is

$$\begin{aligned}h_{f(\text{fittings})} &= K_{\text{Total}} \frac{v^2}{2g} \quad (4-53) \\ h_{f(\text{fittings})} &= (3.903) \frac{1.054^2}{(2 \times 9.81)} \\ &= 0.221 \text{ m}\end{aligned}$$

Total frictional loss due to straight pipe + fittings is

$$\begin{aligned}\Delta P_{f_{\text{Total}}} &= \Delta P_f + \Delta P_{f(\text{fittings})} \\ &= 521.53 + 1060.05 \\ &= 1581.6 \text{ N/m}^2 (1.58 \text{ kN/m}^2)\end{aligned}$$

Total head loss due to straight pipe + fittings is

$$\begin{aligned}h_{\text{Total}} &= h_f + h_{f(\text{fittings})} \\ &= 0.109 + 0.221 \\ &= 0.33 \text{ m}\end{aligned}$$

(continued)

**EXAMPLE 4-2—(continued)**

Using the energy balance equation between the accumulator vessel V-1008 and the suction to reflux pump P1017A/B:

$$q + \frac{P_1}{\rho g} + z_1 + \frac{v_1^2}{2g} = \frac{P_2}{\rho g} + z_2 + \frac{v_2^2}{2g} + e_f + w \quad (4-69a)$$

where

$P_1$  = pressure at vessel V-1008

$P_2$  = pressure at pump suction P1017A

$z_1$  = pipe elevation to pump suction P1017A

$z_2$  = datum level at pump suction P1017A.

$e_f$  = irreversible energy dissipated between the vessel and the pump.

Since the pipe has the same configuration  $v_1 = v_2$ , and assuming that there is no net heat transfer and no work is done by the system,  $q = w = 0$ .

$$-\Delta P = \frac{(P_1 - P_2)}{\rho} = g(z_2 - z_1) + e_f$$

$$-\Delta P = (P_1 - P_2) = \rho g(z_2 - z_1) + \rho e_f$$

$$e_f = \left( 4f_D \frac{L}{D} + \sum K_f \right) \frac{v^2}{2}$$

or

$$e_f = \left( f_D \frac{L}{D} + \sum K_f \right) \frac{v^2}{2}$$

$$e_f = \left( 0.0150 \times \frac{26}{0.2027} + 3.903 \right) \frac{1.054^2}{2}$$

$$= 3.24 \text{ m}^2/\text{s}^2$$

The pressure drop between the vessel and the suction to pump P-1017A is:

$$-\Delta P = (P_1 - P_2) = \rho g(z_2 - z_1) + \rho e_f$$

$$-\Delta P = (488)(9.81)(0 - 7) + (488)(3.24)$$

$$= -33510.96 + 1581.12$$

$$= -31929.84 \text{ N/m}^2$$

$$= -0.32 \text{ bar}$$

The total pressure drop ( $\Delta P_{\text{Total}}$ ) at the suction is

Suction (barg)	
Vessel pressure	12.69
Static-head pressure	0.335
Friction loss	-0.016
$\Delta P_{\text{Total}}$	13.01 bar

**The Frictional losses from the discharge of pump P1017A/B to the debutanizer unit C-1007:**

The discharge specification line is 6"-P10176-3101C

The flow rate of LPG to the debutanizer unit C-1007 is 122.5  $\text{m}^3/\text{h}$  (59,780 kg/h).

Internal diameter of 6" Sch. 40, ID = 154.1 mm  
Pipe area is

$$A = \frac{\pi d^2}{4} = \frac{\pi (0.1541)^2}{4} = 0.0187 \text{ m}^2$$

$$\text{Fluid velocity, } v = \frac{Q}{\text{Area}} = \frac{122.5}{3600} \times \frac{1}{0.0187} \left( \frac{\text{m}^3}{\text{h}} \cdot \frac{\text{h}}{\text{s}} \cdot \frac{1}{\text{m}^2} \right) \\ = 1.82 \text{ m/s}$$

The fluid's (LPG) Reynolds number,  $Re$ , from Eq. (4-28) is

$$Re = 354 \frac{59,780}{(154.1)(0.112)} \\ = 1,226,136 \\ = 1.22 \times 10^6 (\text{fully turbulent})$$

Pipe relative roughness  $\varepsilon/d = 0.046/154.1 = 0.0002985$

Using Chen's explicit Eq. (4-35) to calculate the friction factor,  $f_C$ :

$$A = \frac{\varepsilon/D}{3.7} + \left( \frac{6.7}{Re} \right)^{0.9} \\ = \left( \frac{0.0002985}{3.7} \right) + \left( \frac{6.7}{1226136} \right)^{0.9} \\ = 9.90319 \times 10^{-5}$$

Substituting in Eq. (4-35) the friction factor  $f_C$  is

$$f_C = -4 \log_{10} \left[ \frac{0.0002985}{3.7} - \frac{5.02}{1,226,136} \log_{10} (9.90319 \times 10^{-5}) \right] \\ = 16.0616$$

$$f_C = 3.88114 \times 10^{-3}$$

$$f_D = 4 f_C$$

$$= 4 (3.88114 \times 10^{-3})$$

$$= 0.0155$$

Pressure drop ( $\Delta P_{100}$ ) per 100 m from Eq. (4-66) is

$$\Delta P \text{ bar/100 m} = 225 f \rho Q^2 / d^5 \quad (4-66) \\ = 225 (0.0155) (488) (2041.7)^2 / 154.1^5 \\ = 0.082 \text{ bar/100 m}$$

$$\left( \frac{L}{D} \right) = \frac{134 \text{ m}}{0.1541 \text{ m}} = 869.57$$

and

$$\rho \frac{v^2}{2} = \frac{(488)(1.82)^2}{2} \\ = 808.23 \text{ N/m}^2$$

Frictional pressure loss of straight pipe,  $\Delta P_f$ , is

$$\Delta P_f = (0.0155) (869.57) (808.23) \quad (4-41) \\ = 10893.59 \text{ N/m}^2 \\ = 10.89 \text{ kN/m}^2$$

(continued)

**EXAMPLE 4-2—(continued)**

The head loss,  $h_f$ , is

$$h_f = f_D \left( \frac{L}{D} \right) \frac{v^2}{2g}$$

$$= (0.0155)(869.57) \frac{1.82^2}{(2 \times 9.81)}$$

$$= 2.28 \text{ m}$$
(4-37)

The fourth term in Eq. (4-69a) represents the total loss coefficient  $K$  for all the fittings. Values of  $K$  for the 2- $K$  method (Hooper's) or 3- $K$  method (Darby's) are given in Tables 4-8 and 4-9 respectively.

Using Darby's 3- $K$  method as follows:

Nominal pipe size of a 6" pipe line = 152.4 mm

$Re = 1,226,136$  (fully turbulent)

$$K_f = \frac{K_1}{Re} + K_i \left( 1 + K_d \left( \frac{25.4}{D_{n,\text{mm}}} \right)^{0.3} \right)$$
(4-74)

The pressure loss due to fittings  $\Delta P_{f(\text{fittings})} = K_{\text{Total}} \frac{\rho v^2}{2}$

$$\Delta P_{f(\text{fittings})} = (47.103)(808.23)$$

$$= 38070.06 \text{ N/m}^2$$

$$= 38.07 \text{ kN/m}^2 (0.38 \text{ bar})$$
(4-67)

Fittings	n	$K_1$	$nK_1$	$K_i$	$nK_i$	$K_d$	$K_f$
45° ell ( $r/D = 1.5$ )	6	500	3,000	0.052	0.312	4.0	1.044
90° ell ( $r/D = 1.5$ )	40	800	32,000	0.071	2.84	4.2	9.834
Ball valve ( $\beta = 1$ )	6	300	1,800	0.017	0.102	4.0	0.342
Globe valve ( $\beta = 1$ )	1	1,500	1,500	1.7	1.7	3.6	5.276
Check valve	2	1,500	3,000	0.46	0.92	4.0	3.072
Lift check valve	3	2,000	6,000	2.85	8.55	3.8	27.535
No exit loss, if discharge is above liquid in the column					0		
Total					47.103		

Head loss due to fittings is

$$h_{f(\text{fittings})} = K_{\text{Total}} \frac{v^2}{2g}$$
(4-53)

$$h_{f(\text{fittings})} = (47.103) \frac{1.82^2}{(2 \times 9.81)}$$

$$= 7.95 \text{ m}$$

Total frictional pressure loss due to straight pipe + fittings is

$$\Delta P_{f(\text{Total})} = \Delta P_f + \Delta P_{f(\text{fittings})}$$

$$= 10.89 + 38.07$$

$$= 49.96 \text{ kN/m}^2 (0.49 \text{ bar})$$

Total head loss due to straight pipe + fittings is

$$h_{\text{Total}} = h_f + h_{f(\text{fittings})}$$

$$= 2.28 + 7.95$$

$$= 10.23 \text{ m}$$

Using the energy balance equation between the reflux pump P1017A/B discharge and the distillation column C-1007 unit:

$$q + \frac{P_1}{\rho g} + z_1 + \frac{v_1^2}{2g} = \frac{P_2}{\rho g} + z_2 + \frac{v_2^2}{2g} + e_f + w$$
(4-69a)

where

$P_1$  = pressure at pump discharge P-1017A/B

$P_2$  = pressure at distillation column C-1007

$z_1$  = datum level at pump discharge P-1017A/B

$z_2$  = pipe elevation to the distillation column C-1007

$e_f$  = irreversible energy dissipated between the vessel and the pump.

Since the pipe has the same configuration  $v_1 = v_2$ , and assuming that there is no net heat transfer, and no work is done by the system,  $q = w = 0$

$$-\Delta P = \frac{(P_1 - P_2)}{\rho} = g(z_2 - z_1) + e_f$$

The pressure drop is

$$-\Delta P = (P_1 - P_2) = \rho g(z_2 - z_1) + \rho e_f$$

where

$$e_f = \left( 4f_D \frac{L}{D} + \sum K_f \right) \frac{v^2}{2}$$

or

$$e_f = \left( f_D \frac{L}{D} + \sum K_f \right) \frac{v^2}{2}$$

$$e_f = \left[ (0.0155) \left( \frac{134}{0.1541} \right) + 47.103 \right] \left( \frac{1.82^2}{2} \right)$$

$$= 100.33 \text{ m}^2/\text{s}^2$$

$$-\Delta P = (P_1 - P_2) = (488)(9.81)(43 - 0) + (488)(100.33)$$

$$= 205853.04 + 48963.33$$

$$= 254816.37 \text{ N/m}^2 (2.5 \text{ bar})$$

The total pressure drop ( $\Delta P_{\text{Total}}$ ) at the discharge:

Discharge (barg)	
Vessel pressure	16.59
Static-head pressure	2.058
Friction loss	0.489
$\Delta P_{\text{Total}}$	19.137 bar

(continued)

**EXAMPLE 4-2—(continued)**

Tables 4-15a and b show computer results of the suction and discharge frictional pressure drop calculations of Example 4-2. The Excel spreadsheet program (Example 4-2.xls) provides the pressure drop calculations of the suction and discharge sides of pump P1017A/B of Example 4-2.

**TABLE 4-15a Computer Results of Example 4-2 (Suction side)**

PRESSURE DROP CALCULATION IN A PIPE LINE	
*****	
PIPE INTERNAL DIAMETER, mm:	202.70
MASS FLOW RATE, Kg/h:	59760.00
VOLUMETRIC FLOW RATE, 1/min.:	2040.98
FLUID VISCOSITY, cP:	.1120
FLUID DENSITY, Kg/m <sup>3</sup> :	488.0000
FLUID VELOCITY, m/sec.:	1.0542
VELOCITY HEAD LOSS DUE TO FITTINGS:	2.7346
EQUIVALENT LENGTH OF PIPE, m:	36.8998
ACTUAL LENGTH OF PIPE, m:	19.0000
TOTAL LENGTH OF PIPE, m:	55.8998
REYNOLDS NUMBER:	931842.
PIPE ROUGHNESS, mm:	.000046
DARCY FRICTION FACTOR:	.0150
EXCESS HEAD LOSS, m:	7.2346
PIPE PRESSURE DROP/100 m, bar/100m:	.0201
OVERALL PRESSURE DROP OF PIPE, bar:	.3463
-----	

**TABLE 4-15b Computer Results of Example 4-2 (Discharge side)**

PRESSURE DROP CALCULATION IN A PIPE LINE	
*****	
PIPE INTERNAL DIAMETER, mm:	154.10
MASS FLOW RATE, Kg/h:	59760.00
VOLUMETRIC FLOW RATE, 1/min.:	2040.98
FLUID VISCOSITY, cP:	.1120
FLUID DENSITY, Kg/m <sup>3</sup> :	488.0000
FLUID VELOCITY, m/sec.:	1.8240
VELOCITY HEAD LOSS DUE TO FITTINGS:	21.3443
EQUIVALENT LENGTH OF PIPE, m:	211.8641
ACTUAL LENGTH OF PIPE, m:	91.0000
TOTAL LENGTH OF PIPE, m:	302.8641
REYNOLDS NUMBER:	1225725.
PIPE ROUGHNESS, mm:	.000046
DARCY FRICTION FACTOR:	.0155
EXCESS HEAD LOSS, m:	48.1736
PIPE PRESSURE DROP/100 m, bar/100m:	.0817
OVERALL PRESSURE DROP OF PIPE, bar:	2.3060
-----	

**EXAMPLE 4-3  
Laminar Flow Through Piping System**

A heavy weight oil, No. 5 fuel oil, is to be pumped through 350 ft of existing 4-in. Sch. 40 pipe at 350 gpm to a tank, which is 40 ft above the pump level. Determine the frictional pressure loss.

Oil data:

Temperature = 100° F  
Viscosity = 150 cP  
sp gr = 0.78 = 48.6 lb/ft<sup>3</sup>  
Pipe ID = 4.026 in. = 0.3355 ft

$$\begin{aligned} \text{Reynolds number} &= 50.6 Q \rho / (d \mu) \quad (4-27) \\ &= 50.6 [(350)(48.6)/(4.026)(150)] \\ &= 1425 \end{aligned}$$

Reynolds number,  $Re < 2000$

$$\begin{aligned} f &= 64/Re \quad (4-32) \\ &= 64/1425 = 0.0449 \end{aligned}$$

The loss coefficient  $K$  for a straight pipe

$$K = f_D \left( \frac{L}{D} \right) \quad (4-55)$$

Frictional loss due to straight line of pipe (350 ft) is

For 350 ft pipe,  $K = f_D (L/D) = 0.0449 (350/0.3355) = 46.84$   
Using Darby's 3-K method, Eq. (4-73)

$Re = 1425$

Nominal pipe diameter of a 4-in. pipe line is = 4.0 in.

Fittings	n	K <sub>1</sub>	nK <sub>1</sub>	K <sub>i</sub>	nK <sub>i</sub>	K <sub>d</sub>	K <sub>f</sub>
90° ell ( $r/D = 1.5$ )	4	800	3200	0.14	0.56	4.0	4.283
Gate valve	1	300	300	0.037	0.037	3.9	0.343
Entrance to tank projecting inward (Table 4-6)	1						0.78
Total							5.406

Total K valves = 46.84 + 5.406 = 52.246

Fluid velocity is:

$$v = 0.408 \frac{Q}{d^2}, \text{ ft/s} \quad (4-30)$$

where

$Q$  = fluid flow rate, gpm

$d$  = pipe internal diameter = 4.026 in.

$$v = 0.408 \frac{350}{(4.026)^2} = 8.81 \text{ ft/s}$$

(continued)

**EXAMPLE 4-3—(continued)**  
Velocity head in ft is

$$\frac{v^2}{2g} = \frac{(8.81)^2}{(2)(32.2)} = 1.205 \text{ ft}$$

Applying the energy balance equation (English Engineering units):

$$\frac{q}{g_c} + 144 \frac{P_1}{\rho} + z_1 \frac{g}{g_c} + \frac{v_1^2}{2g_c} = 144 \frac{P_2}{\rho} + z_2 \frac{g}{g_c} + \frac{v_2^2}{2g_c} + \frac{e_f}{g_c} + \frac{w}{g_c} \quad (4-69)$$

where

Inlet pressure  $P_2$  to the suction of the pump.

Since the pipe is horizontal,  $v_1 = v_2$ , assuming that there is no net heat transfer and no work is done by the system,  $q = w = 0$

$$\begin{aligned} -\Delta P &= (P_1 - P_2) = \frac{\rho}{144} (z_2 - z_1) \frac{g}{g_c} + \frac{\rho}{144} \frac{e_f}{g_c} \\ e_f &= \left( 4f_F \frac{L}{D} + \sum K_f \right) \frac{v^2}{2} \end{aligned}$$

or

$$\begin{aligned} e_f &= \left( f_D \frac{L}{D} + \sum K_f \right) \frac{v^2}{2} \\ e_f &= \left( 0.0449 \left( \frac{350}{0.3355} \right) + 5.406 \right) \left( \frac{8.81^2}{2} \right) \\ &= 2027.59 \text{ ft}^2/\text{s}^2 \end{aligned}$$

Assuming that there is no elevation,  $z_1 = z_2$

Therefore the pressure drop  $\Delta P$  is

$$\begin{aligned} -\Delta P &= (P_1 - P_2) = \frac{\rho}{144} \frac{e_f}{g_c} \\ \Delta P &= \frac{(48.6)(2027.59)}{(144)(32.174)} \\ &= 21.27 \text{ psi} \end{aligned}$$

## 4.29 ALTERNATE CALCULATION BASIS FOR PIPING SYSTEMS FRICTION HEAD LOSS: LIQUIDS

Pressure loss in a piping system (not including the tanks, heat exchangers, distillation columns, etc.) is usually expressed in units of "feet (meters) of flowing fluid", or the "equivalent converted to pounds per square inch (bar)". Some published pressure loss data are expressed as per "100 equivalent feet (100 equivalent metres)" of the pipe size being used or estimated.

## 4.30 EQUIVALENT LENGTH CONCEPT FOR VALVES, FITTINGS AND SO ON

With pipe of any specified size as the basis, the total footage of "straight pipe" in a system is just the measured length (totaled).

For fittings, valves, and so on, in the same system, these can be expressed as "equivalent straight pipe," and then added to the straight pipe described above, to arrive at a total equivalent straight length of pipe of the specific size in question.

Figure 4-24a presents equivalent lengths of straight pipe in feet (meters) for various pipe system components. For example, a standard threaded 6-in. 90° elbow is equivalent to adding 17 ft

The frictional pressure drop  $\Delta P$  from Eq. (4-63a) with pipe elevation is:

$$\begin{aligned} \Delta P &= 0.000179 \frac{K\rho Q^2}{d^4} + \frac{\rho \Delta z}{144} \frac{g}{g_c} \\ &= 0.0000179 \frac{(52.246)(48.6)(350)^2}{4.026^4} + \frac{(48.6)(40)}{144} \\ &= 34.69 \text{ psi.} \end{aligned}$$

Table 4-16 shows the computer results of example 4-3. The Excel spreadsheet program (Example 4-3.xls) provides the pressure drop calculations of Example 4-3.

**TABLE 4-16 Computer Results of Example 4-3**

PRESSURE DROP CALCULATION IN A PIPE LINE	
*****	
PIPE INTERNAL DIAMETER, inch:	4.026
MASS FLOW RATE, lb/h:	136347.30
VOLUMETRIC FLOW RATE, gal/min.:	350.00
FLUID VISCOSITY, cP:	150.0000
FLUID DENSITY, lb/ft.^3:	48.6000
FLUID VELOCITY, ft/sec.:	8.810
VELOCITY HEAD LOSS DUE TO FITTINGS:	4.1891
EQUIVALENT LENGTH OF PIPE, ft.:	31.3280
ACTUAL LENGTH OF PIPE, ft.:	350.0000
TOTAL LENGTH OF PIPE, ft.:	381.328
REYNOLDS NUMBER:	1427.
PIPE ROUGHNESS, ft.:	.0001500
DARCY FRICTION FACTOR:	.0449
EXCESS HEAD LOSS, ft.:	101.620
PIPE PRESSURE DROP/100 ft, psi/100 ft:	5.455
OVERALL PRESSURE DROP OF PIPE, psi:	34.303

straight pipe to the system. This 17 ft is additive to the lengths of nominal 6-in. straight pipe in the system (dotted line). However, there is an important consideration in the use of this chart, that is, use only for threaded or screwed pipe/fittings, and only for sizes under 2-in. nominal size. It is not practical in current industry practice to thread a process or utility system much greater in nominal diameter than 2 in. For special situations, the larger sizes can be used, but from a handling standpoint, sizes greater than 3 or 4 in. are not practical.

For pipe sizes greater than 2 in. nominal, industry practice is to weld the pipe and fittings into one continuous system, and then use flanged or special bolted connections for attaching the valves, orifices, and connections to vessels or other equipment. For special lethal, high pressure, and steam power plant high temperature/high pressure utility systems, even the valves and connections to vessels are welded into the system (see ASME and ANSI Codes). For these situations of about 1.5-in. to 2-in. nominal pipe size and larger, use Figure 4-25 to determine the equivalent pipe lengths for these fittings, valves, and so on. For example, a 45° welding elbow or an open 6-in. gate valve (see line on chart) have an equivalent length of 6-in. pipe of 4 ft (straight), which is an addition to

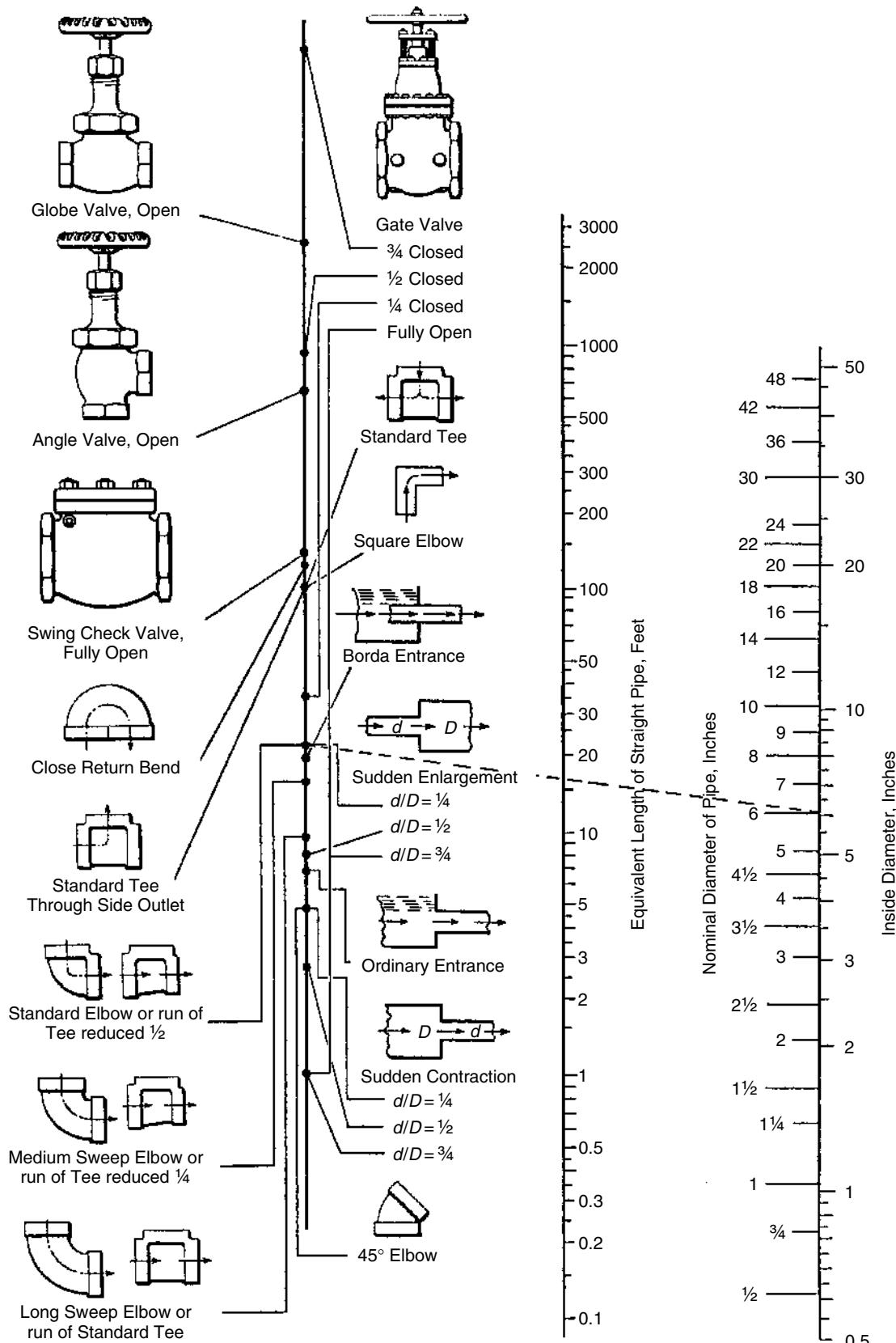


Figure 4-24a Equivalent length resistance of valves and fittings to flow of fluids. Note: Apply to 2 in. and similar threaded pipe for process applications (Ludwig [19]). (By permission from Crane Co., Technical Paper No. 409, Engineering Div., 1942, also see [4].)

### Equivalent Lengths $L$ and $L/D$ and Resistance Coefficient $K$

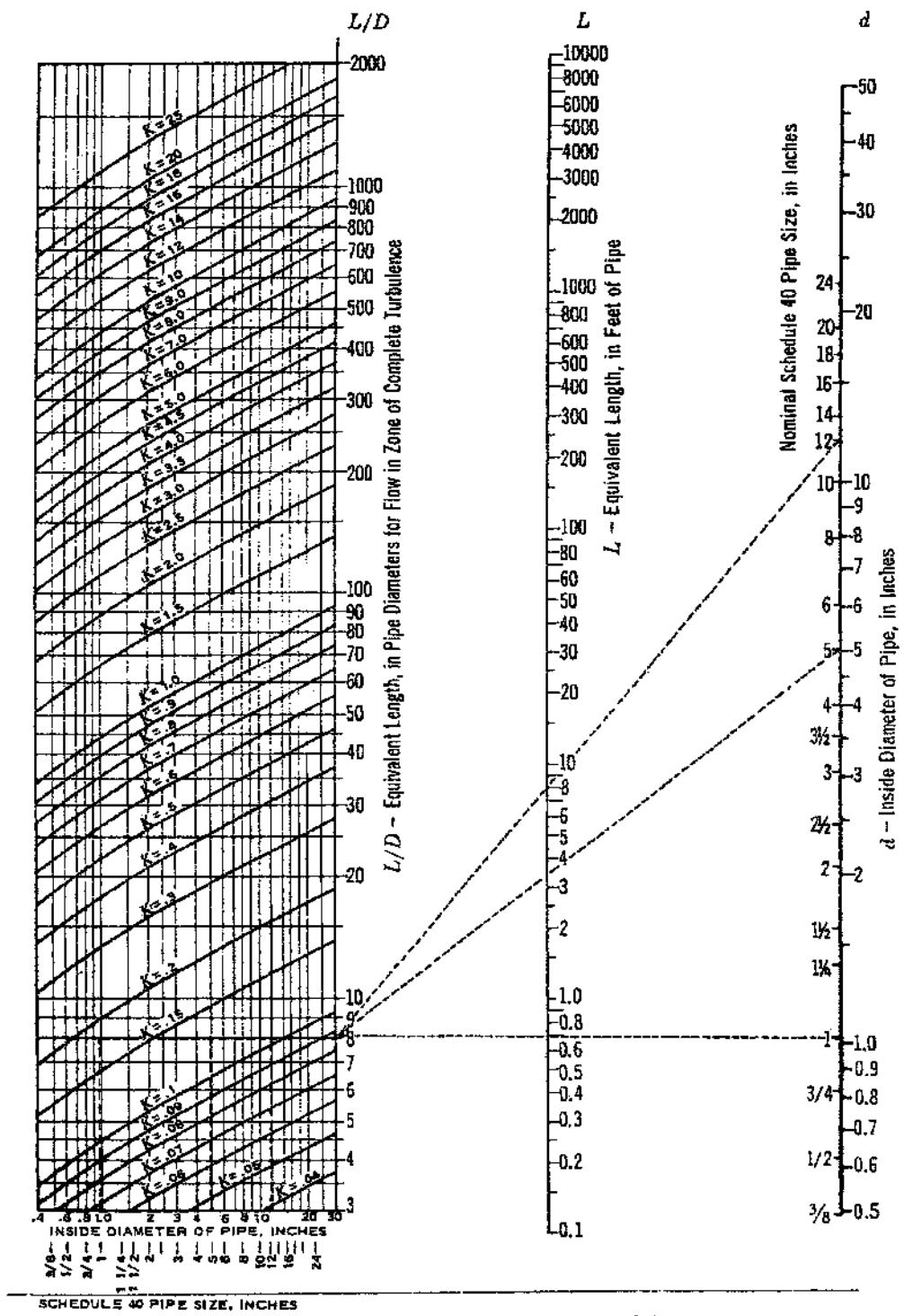
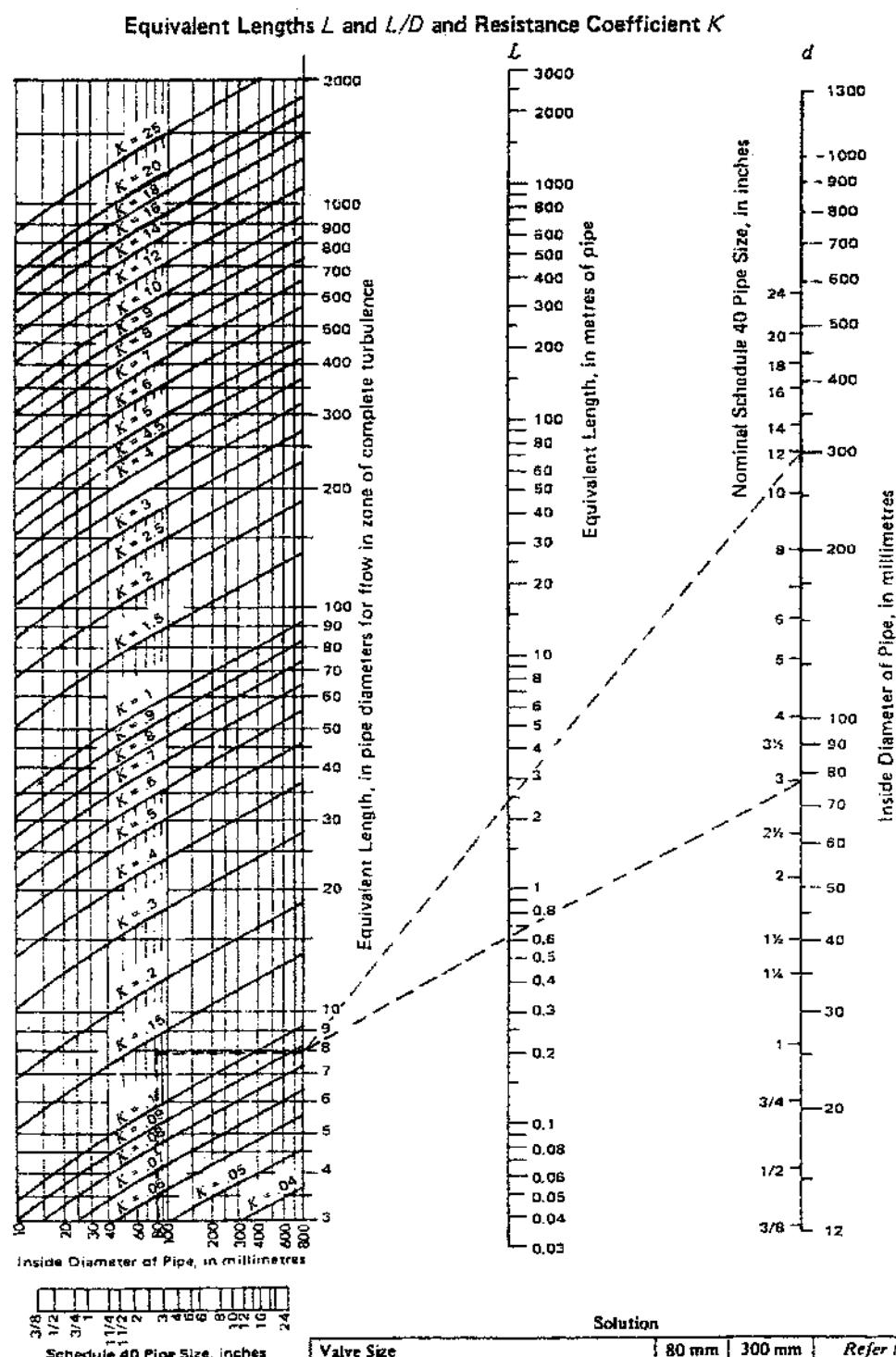


Figure 4-24b Equivalent length and resistance coefficient for nominal Sch. 40 pipe size, in inches. (Reprinted/Adapted with permission from "Flow of fluids Through Valves, Fittings, and Pipe", Technical Paper No. 410, 1999, Crane Co. All rights reserved.)



**Problem:** Find the equivalent length in pipe diameters and metres of Schedule 40 clean commercial steel pipe, and the resistance factor  $K$  for 80 mm and 300 mm fully opened gate valves, ANSI Class 300, with flow in zone of complete turbulence.

For discussion on  $D/D$  and  $K$  see pages 2-8 to 2-10

Valve Size	80 mm	300 mm	Refer to
Equivalent length, pipe diameters	8	8	Page A-27
Equivalent length, metres of Sch. 40 pipe	0.62	2.43	Dotted lines
Resist Factor $K$ , based on Sch. 40 pipe	0.14	0.10	on chart

**Figure 4-24c** Equivalent length and resistance coefficient for nominal Sch. 40 pipe size, in mm. (Reprinted/Adapted with permission from "Flow of fluids Through Valves, Fittings, and Pipe", Technical Paper No. 410M, 1999, Carne Co. All rights reserved.)

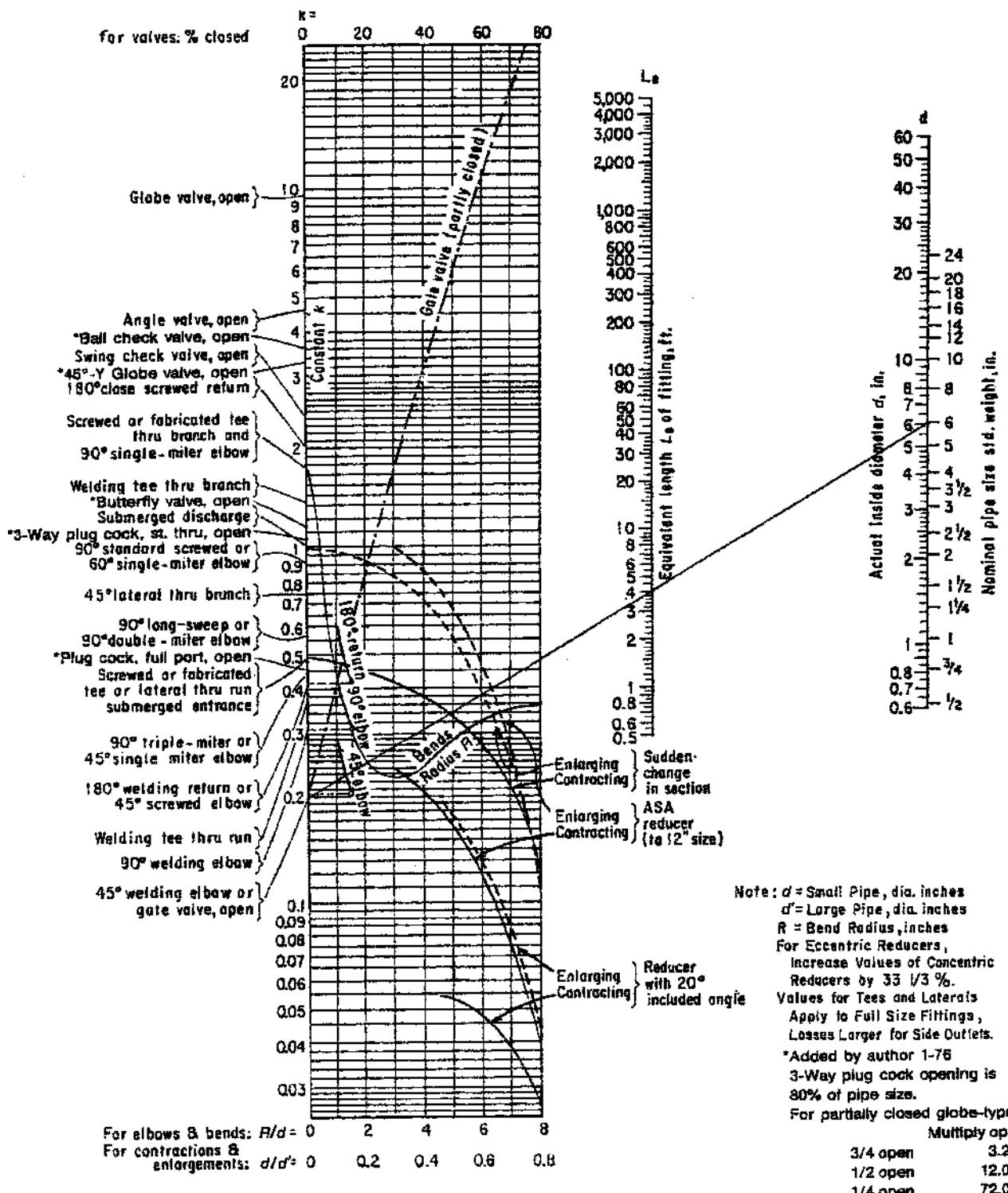


Figure 4-25 Equivalent length of fittings for pipe systems. Note: Preferred use for  $1\frac{1}{2}$  in. and larger pipe butt- or socket-welded connections (Ludwig [19]). (By permission from Tube Turns Div., Chemitron Corp, Bull. TT 725, 1952, reference now to Tube Turns from Technologies, Inc.)

the actual straight pipe in the system. In summation, these equivalent lengths for all the components determine the total pipe length to use in the pressure loss (pressure drop) equations to be described later.

### 4.31 FRICTION PRESSURE DROP FOR NON-VISCOUS LIQUIDS

The only significance in differentiating between water and liquids of different densities and viscosities is the convenience in having a separate simplified table for water.

- Using known flow rate in gallons per minute and a suggested velocity from Tables 4-12, 4-13, 4-17, 4-18, and 4-19 or Figures 4-26a and b (SI), first estimate the pipe size. Mean velocity of any liquid flowing in a pipe [4] is given by Figures 4-26a and b and Eq. (4-30).

English Engineering units

$$v = q/A = w_s(A)(\rho), \text{ ft/s}$$

$$v = 0.408 Q/d^2 = 0.0509 W/(\rho d^2) = 183.3 q/d^2, \text{ ft/s}$$
(4-30)

In SI units,

$$v = 21.22 Q/d^2 = 354 W/(\rho d^2) = 1,273,000 q/d^2, \text{ m/s}$$
(4-31)

$$d = (0.408 Q/v)^{0.5} = (0.0509 W/v\rho)^{0.5} = (183.3 q/v)^{0.5}, \text{ in.}$$
(4-94)

In SI units,

$$d = (21.22 Q/v)^{0.5} = (354W/v\rho)^{0.5} = (1,273,000 q/v)^{0.5}$$
(4-95)

- Estimate or otherwise determine the linear feet (meters) of straight pipe in the system,  $L$ .
- Estimate (or use actual tabulation) number of fittings, valves, and so on, in system. Convert these to equivalent straight pipe using Figure 4-24 or 4-25,  $L_{eq}$ , or head by Figures 4-14 through 4-18 and Table 4-6.

Note preferred pipe size type from charts.

- Determine expansions or contraction losses, if any, including tank or vessel entrance or exit losses from Figures 4-14a, 4-17, or 4-18. Convert units to psi (bar), head loss in feet times 0.4331 = psi (for water); head loss in meters times 0.0980 = bar (for water), or adjust for sp gr of other liquids.
- Estimate pressure drop through orifices, control valves, and other items in the system, but not equipment. For control valves, estimate  $\Delta P$  from paragraph to follow.
- Determine pressure drop per unit length.

- a. Calculate Reynolds number [4]  
English Engineering units

$$Re = 50.6Q\rho/(d\mu) = 631 W/d\mu = 123.9 dv\rho/\mu$$
(4-27)

In SI units,

$$Re = 21.22Q\rho/(d\mu) = 354 W/(d\mu) = dv\rho/\mu$$
(4-28)

**TABLE 4-17 Typical Design Vapor Velocities\* (ft/s)**

Fluid	Line Sizes		
	≤ 6 in.	8–12 in.	≥ 14 in.
Saturated vapor			
0–50 psig	30–115	50–125	60–145
Gas or superheated vapor			
0–10 psig	50–140	90–190	110–250
11–100 psig	40–115	75–165	95–225
101–900 psig	30–85	60–150	85–165

\* Values listed are guides, and final line sizes and flow velocities must be determined by appropriate calculations to suit circumstances. Vacuum lines are not included in the table, but usually tolerate higher velocities. High vacuum conditions require careful pressure drop evaluation.

**TABLE 4-18 Usual Allowable Velocities for Duct and Piping Systems**

Service/Application	Velocity (ft/min)
Forced draft ducts	2,500–3,500
Induced-draft flues and breeching	2,000–3,000
Chimneys and stacks	2,000
Water lines (max.)	600
High pressure steam lines	10,000
Low pressure steam lines	12,000–15,000
Vacuum steam lines	25,000
Compressed air lines	2,000
Refrigerated vapor lines	
High pressure	1,000–3,000
Low pressure	2,000–5,000
Refrigerated liquid	200
Brine lines	400
Ventilating ducts	1,200–3,000
Register grilles	500

(Source: By permission from *Chemical Engineer's Handbook*, 3rd ed. McGraw-Hill Book Co., New York, NY, p. 1642.)

**TABLE 4-19 Suggested Steam Pipe Velocities in Pipe Connecting to Steam Turbines**

Service—Steam	Typical Range (ft/s)
Inlet to turbine	100–150
Exhaust, non-condensing	175–200
Exhaust, condensing	500–400

- b. From Reynolds Number-Friction Factor Chart, Figure 4-5, read friction factor,  $f$  at  $\epsilon/d$  value taken from Figure 4-13.  
c. Calculate pressure drop per 100 ft of (straight and/or equivalent) pipe [4] as psi/100 ft (bar/100 m). Establish piping system friction pressure drop (loss), liquids (Figures 4-27a and 4-27b (Metric)).

For turbulent flow:

$$\Delta P/100 \text{ ft} = 0.0216 f \rho Q^2/d^5 = 0.000336 f W^2/(\rho d^5)$$

$$= 0.1294 f \rho v^2/d = 4350 f \rho q^2/d^5$$
(4-65)

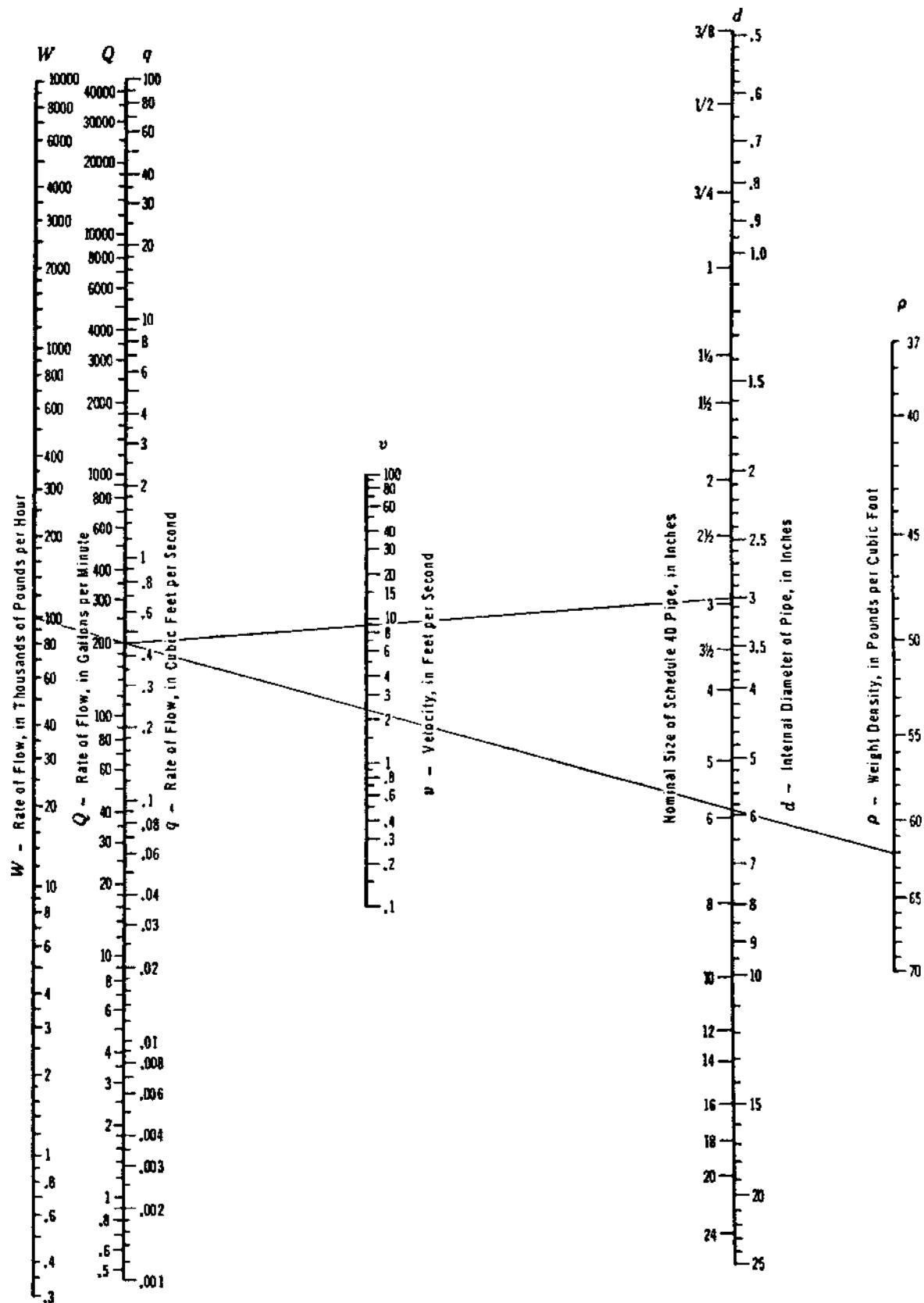


Figure 4-26a Velocity of liquid in pipe. (Reprinted/Adapted with permission from "Flow of Fluids Through Valves, Fittings, and pipe," Technical Paper No. 410, 1999, Crane Co. All rights reserved.)

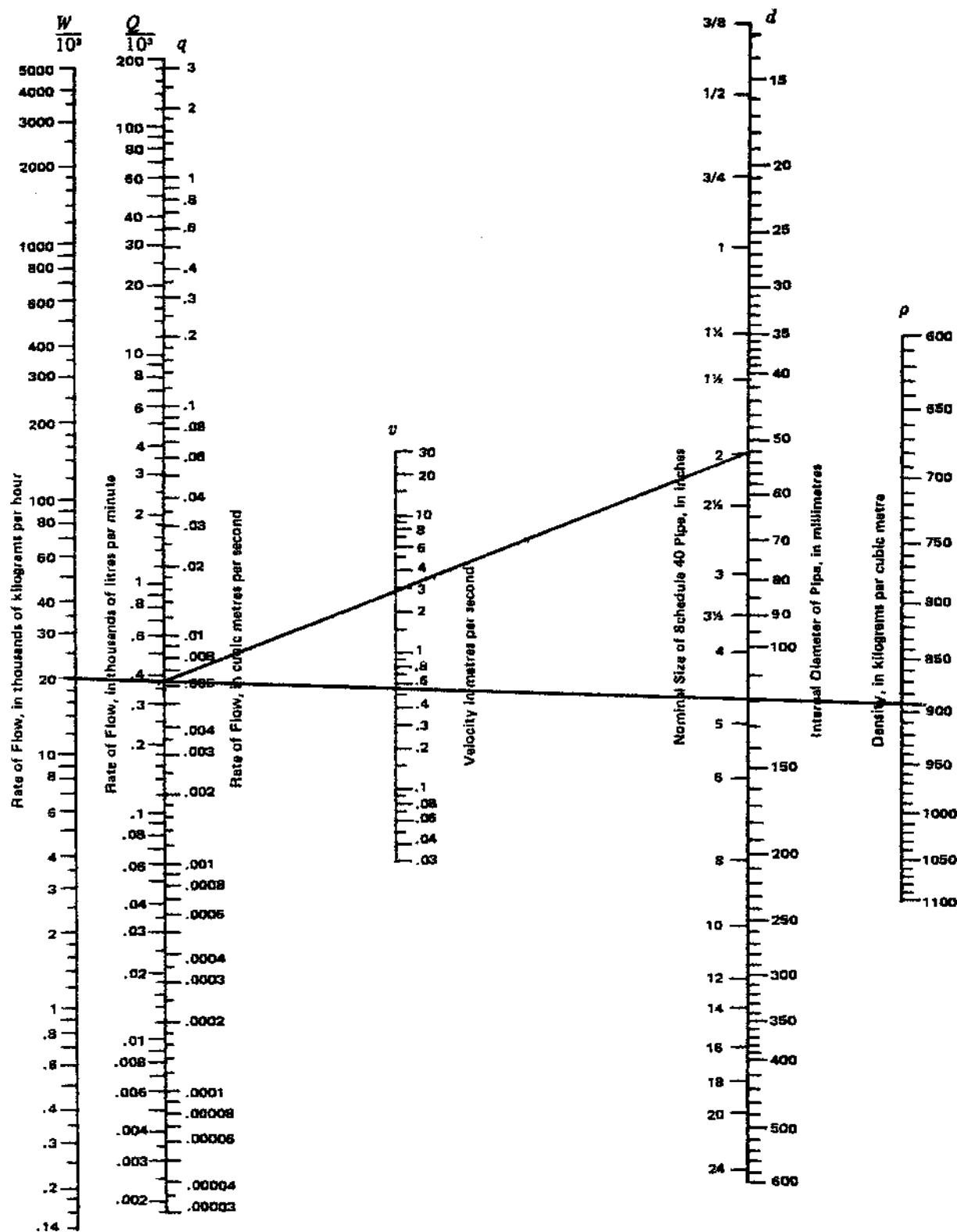
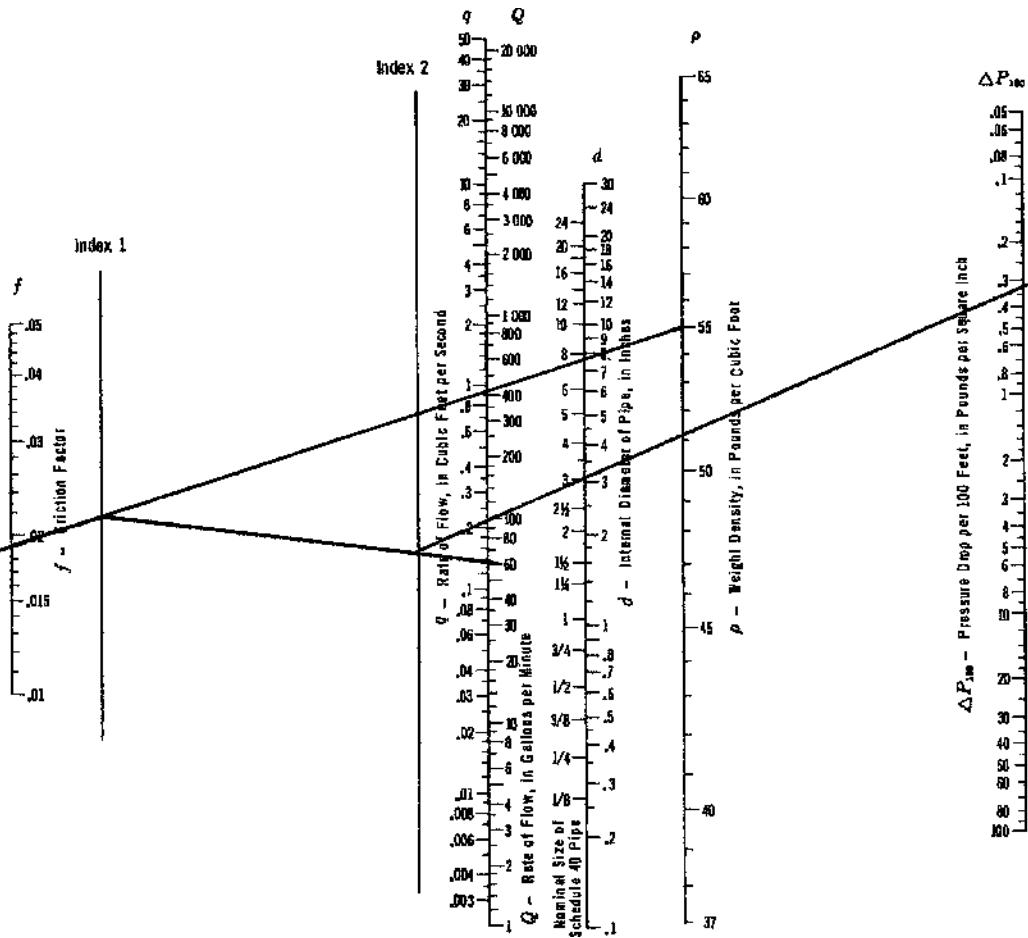


Figure 4-26b Velocity of liquid in pipe. (Reprinted/Adapted with permission from "Flow of Fluids Through Valves, Fittings, and Pipe", Technical Paper No. 410M, 1999, Crane Co. All rights reserved.)



**Figure 4-27a** Pressure drop in liquid lines for turbulent flow. (Reprinted/Adapted with permission from "Flow of Fluids Through Valves, Fittings, and Pipe", Technical Paper No. 410, 1999, Crane Co. All rights reserved.)

For laminar flow:

$$\begin{aligned}\Delta P/100 \text{ ft} &= 0.0668 \mu v/d^2 = 0.0273 \mu Q/d^4 \\ &= 12.25 \mu q/d^4\end{aligned}\quad (4-63)$$

In SI units,

For turbulent flow:

$$\begin{aligned}\Delta P/100 \text{ m} &= 225 f \rho Q^2/d^5 = 62,530 f W^2/d^5 \rho \\ &= 0.5 f \rho v^2/d = 81,055 \times 10^7 f \rho q^2/d^5\end{aligned}\quad (4-66)$$

For laminar flow:

$$\begin{aligned}\Delta P/100 \text{ m} &= 32 \mu v/d^2 = 679 \mu Q/d^4 \\ &= 4074 \times 10^4 \mu q/d^4\end{aligned}\quad (4-63)$$

Where  $\Delta P_{100}$  is the pressure drop in psi (bar) for 100 ft (100 m) of pipe.

#### 7. Total pressure drop for system:

$$\begin{aligned}\Delta P, \text{ psi} &= (L + \sum L_{eq}) (\Delta P/100 \text{ ft from 6c above}) \\ &\quad + 4 \text{ above} + 5 \text{ above}\end{aligned}\quad (4-96)$$

In SI units,

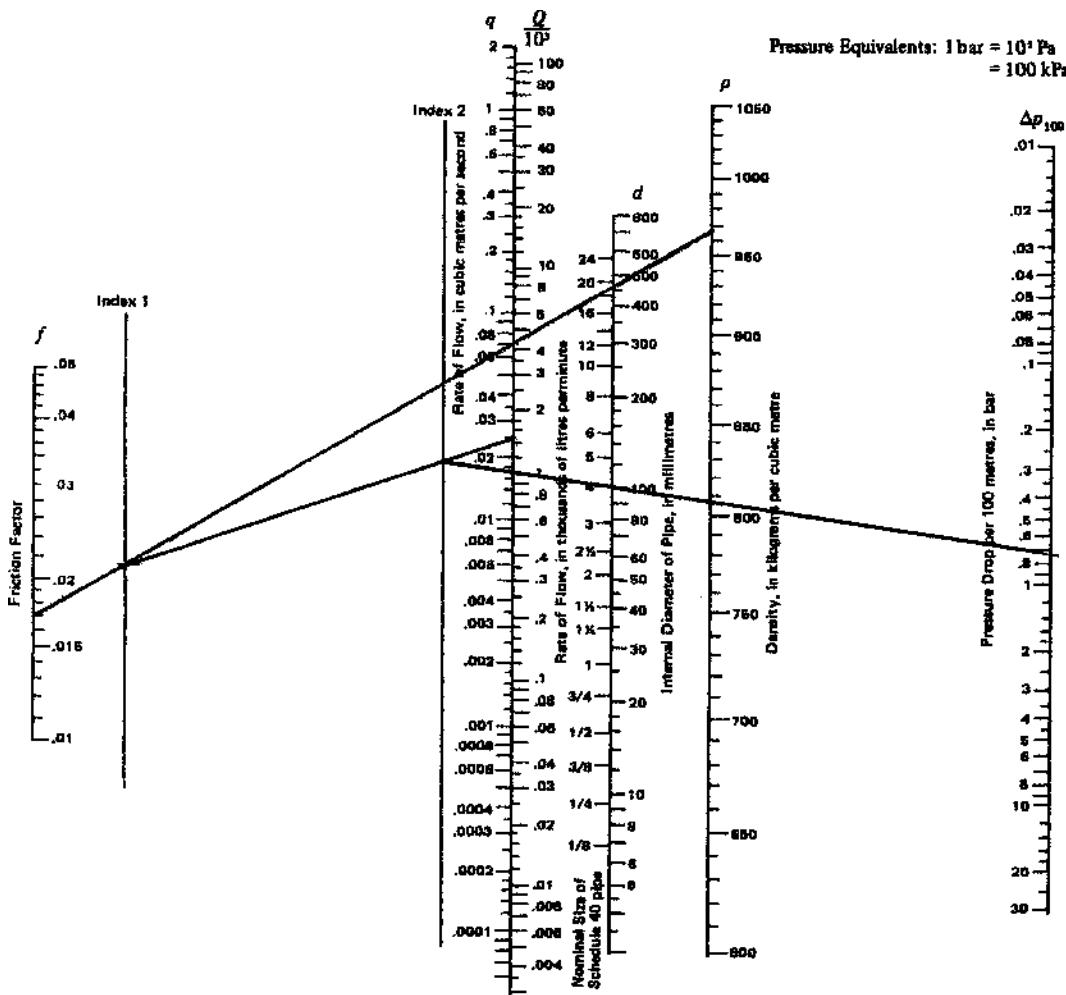
$$\begin{aligned}\Delta P, \text{ bar} &= (L + \sum L_{eq}) (\Delta P/100 \text{ m from 6c above}) \\ &\quad + 4 \text{ above} + 5 \text{ above}\end{aligned}\quad (4-97)$$

(Note:  $L_{eq}$  is from 3 above.)

If this pressure drop is too large or too small, recheck the steps using larger or smaller pipes as may be indicated. The tables in Cameron [58], Table 4-46, or Figure 4-28 are very convenient to use, although they give much more conservative results (about twice unit head loss) than the method outlined above. When using Figure 4-28, the results agree acceptably well with tests on 15- to 20-year-old steel pipe. Also see Table 4-46. For brine, Table 4-20 gives multipliers to use with the water unit losses of Figure 4-28. Figure 4-29 gives direct reading values with Dowtherm® liquid.

It is important to note that comparison of results from these charts does not yield exact checks on any particular fitting. Calculations should never be represented as being more accurate than the basic information. Therefore, rounded values to no more than one decimal place are limits for such head loss calculations.

The head losses calculated using  $K$  coefficients by these figures can be added directly to the total friction head loss for the straight pipe portions of a system. When equivalent lengths are determined, they must be added to the straight pipe before



**Figure 4-27b** Pressure drop in liquid lines for turbulent flow. (Reprinted/Adapted with permission from "Flow of Fluids Through Valves, Fittings, and Pipe", Technical Paper No. 410M, 1999, Crane Co. All rights reserved.)

determining the total head loss, as shown in the example calculations for a water system.

Friction loss in rubber-lined pipe is usually considered equivalent to that in new steel pipe of one-half to one nominal size smaller, with little or no change due to aging, unless known conditions can be interpolated. For a given inside diameter, the friction loss is the same (or slightly less) than clean steel pipe.

In the turbulent flow range, friction loss in glass pipe is 70–85% of clean steel. For 2-in. (nominal) and larger vinyl, saran, or hard rubber pipe, the friction loss does not exceed clean steel. With saran and rubber-lined pipe the loss is about equal to clean steel at the 2.5-in. size, increasing between 2 and 4 times the loss at the 1-in. size.

#### 4.32 ESTIMATION OF PRESSURE LOSS ACROSS CONTROL VALVES

##### LIQUIDS, VAPORS, AND GASES

Despite the need for good control in many process systems, most engineers do not allow the proper pressure drop for the control valves into their calculations. Many literature sources ignore the problem, and many plant operators and engineers wonder why the actual plant has control problems.

Rather than assuming a pressure drop across the control as 25%, 33%, or 40% of the other friction losses in the system, a logical approach [18] is summarized here. The control valve pressure drop has nothing to do with the valve size, but is determined by the pressure balance (Eq. (4-99)) [18].

Control valve pressure drop:

$$P_s = P_e + F_D + P_c \quad (4-98)$$

$$\text{Available } \Delta P_c = (P_s - P_e) - F_D, \text{ psi} \quad (4-99)$$

where

$P_s$  = total pressure at beginning (higher pressure) of system, psig, including any static heads to reach final pressure,  $P_e$

$P_e$  = pressure at lower end of system, psig

$F_D$  = friction loss at design basis, total, for the system, psi, including equipment and piping, at  $Q_D$  rate

$Q_M$  = maximum anticipated flow rate for system, gpm, or ACFM

$F_M$  = friction pressure drop at maximum flow rate  $Q_M$ , psi

$Q_D$  = design flow rate, gpm or ACFM

$\Delta P_c$  = pressure drop across control valve

$F_M$  = friction pressure drop at maximum flow rate, psi.

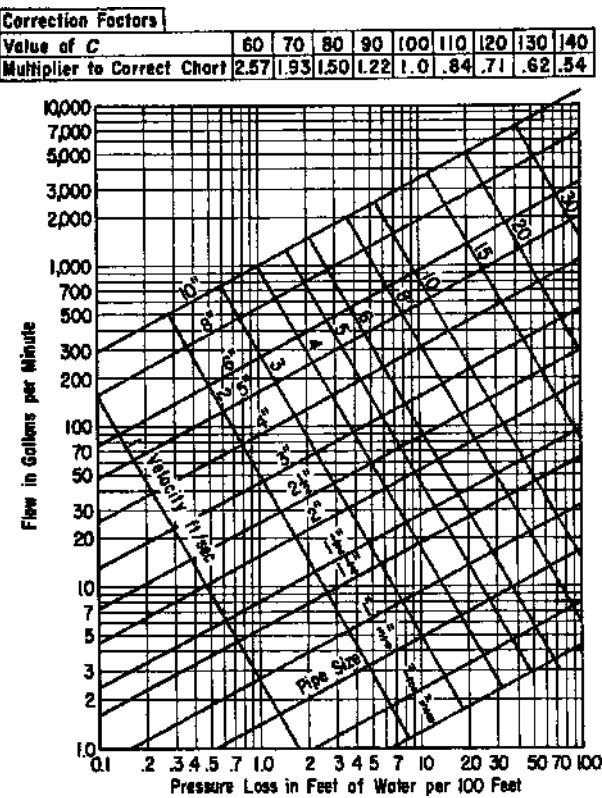


Figure 4-28 Friction loss for flow of water in steel pipes. Note:  $C$  = pipe roughness factor. See Table 4.20 (Courtesy of Carrier Corp.).

Friction loss or drop at higher flow rates than design:

$$\text{Increased pressure drop} = \left[ F_D (Q_M/Q_D)^2 - F_D \right],$$

$$\text{or } \left[ \left( \frac{Q_M}{Q_D} \right)^2 - 1 \right] (F_D) \quad (4-100)$$

Allowing 10% factor of safety, expected maximum increase in friction pressure drop allowance:

$$= 1.1 \left[ (Q_M/Q_D)^2 - 1 \right] (F_D) \quad (4-101)$$

At maximum flow rate,  $Q_M$ , the *friction drop* will become

$$F_M = F_D (Q_M/Q_D)^2 \quad (4-102)$$

The friction loss or pressure drop,  $F_D$ , is determined at the design flow rate,  $Q_D$ , for the piping, valves and friction producing equipment (such as tubular heat exchangers, tubular furnaces/heaters), orifice or other meters and control valves. Because the system friction pressure loss changes with flow rate through the system, recognition must be given to the changes in flow rate (increase or decrease) as it affects the pressure loss through the control valves. For any design, the beginning and end points of the system should be relatively constant for good process operations.

For good control by the valve, the pressure drop across (or through) the valve must always be greater than the friction losses of the system by perhaps 10–20% (see [18]).

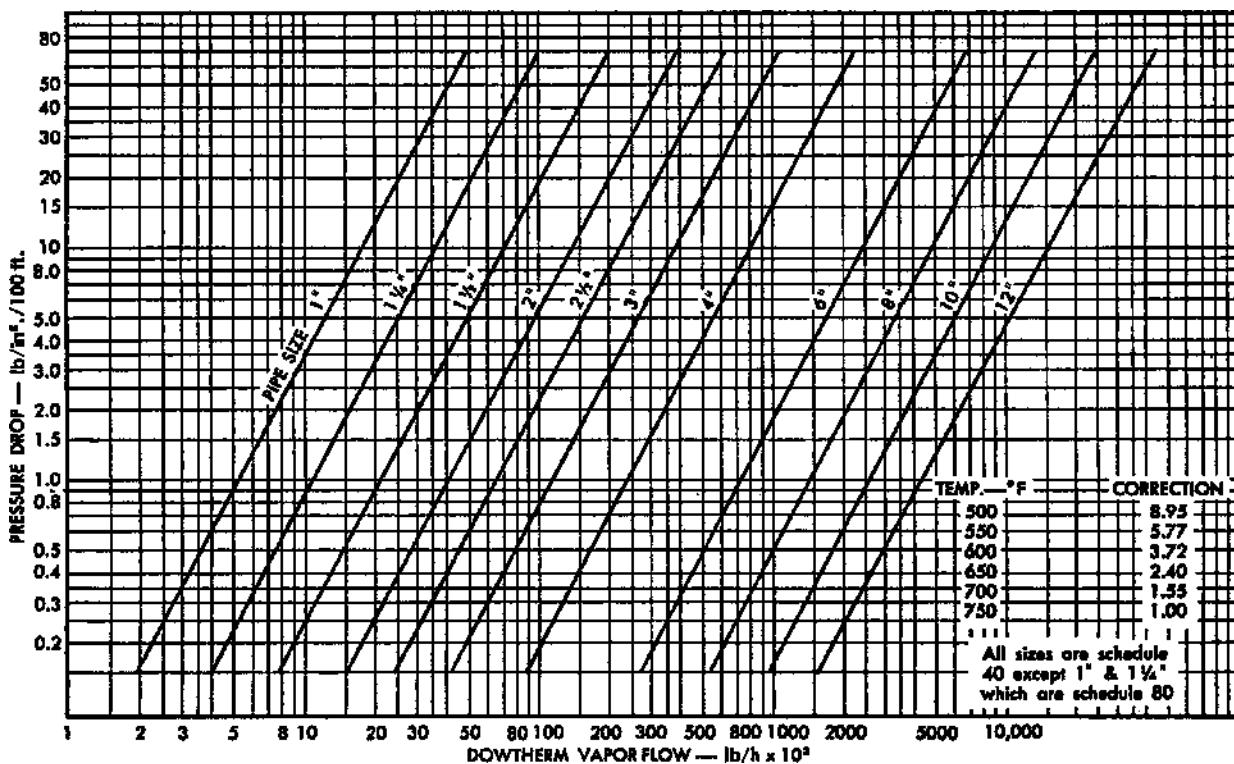


Figure 4-29 Pressure drop for Dowtherm® liquid in Schedule 40 pipe. (By permission from Struthers Wells Corp. Bull., 1956, pp. 4–45).

**TABLE 4-20 Brine Pipe Friction Multiples for Use with Water Friction Data (Figure 4-28)**

Brine	Specific Gravity	Brine Temperature °F						
		0	10	20	30	40	50	60
Sodium chloride	1.10	—	1.23	1.20	1.18	1.16	1.14	1.13
	1.15	1.43	1.33	1.29	1.26	1.24	1.22	1.21
	1.20	1.53	1.44	1.38	1.35	1.32	1.30	1.28
Calcium chloride	1.05	—	—	1.15	1.12	1.10	1.08	1.07
	1.10	—	1.28	1.23	1.20	1.18	1.16	1.14
	1.15	1.41	1.35	1.31	1.28	1.25	1.22	1.21
	1.20	1.49	1.43	1.39	1.36	1.33	1.30	1.28
	1.25	1.56	1.53	1.49	1.45	1.42	1.40	1.38
	1.30	1.65	1.61	1.58	1.55	1.52	1.50	1.49

(Source: By permission from Crocker, S., *Piping Handbook*, McGraw-Hill Book Co.)

Note: To find brine friction loss, multiply loss from Figure 4-12 by multiplier from above table

**EXAMPLE 4-4****Establishing Control Valve Estimated Pressure Drop, using Connell's Method [18]**

Refer to Figure 4-30 for an example to determine the pressure loss (drop) through the control valve.

where

$$P_e = \text{system end pressure} = 22 + 15 = 37 \text{ psig (not friction)}$$

Piping system pipe friction at  $Q_D$  flow rate = 6 psi

Heater, friction = 65 psi

Separator, friction = 1 psi

Preheaters, 10 + 12 (friction) = 22 psi

Orifice, allow, friction = 2 psi

Total friction, excluding control valve,  $F_D = 96$  psi.

Assuming pressure loss through control valve = 35 psi

$$\Delta P_c = (P_s - P_e) - F_D, \text{ psi} \quad (4-99)$$

$$35 = (P_s - 37) - 96$$

$P_s = 168$  psi, at pump discharge, using assumed control valve pressure drop of 35 psi

Note that  $P_e = 22$  psig + 15 psi static  $H_d = 37$  psigAssume that allowance must be made for a 10% increase in process flow rate, above design,  $Q_D$ . Pressure drop varies as the square of the flow rate.New flow rate = 110% ( $Q_D$ )Friction pressure drop will increase to 121% of  $F_D$ :

$$1.21(96) = 116 \text{ psi} = F_M$$

Friction increase = 116 - 96 = 20 psi added for relatively constant  $P_s$  and  $P_e$ Available  $\Delta P_c = (168 - 37) - 116$ 

$\Delta P_c = 15$  psig through the control valve, which means that the valve has to open more and reduce its sensitivity of response, from its design  $\Delta P_c$  of 35 psig.

For design purposes, the assumed 35 psi for the control valve could be used; however, decreasing the pipe friction of 6 psi to perhaps  $\frac{1}{2}$  or  $\frac{1}{3}$  by increasing the line size will help the control of the valve. It would be better to have the *available* valve pressure drop equal to or greater than the assumed.

Factors to consider in evaluating the control valve pressure drop are:

**A. Allowance for increase in friction drop**

Establish the ratio of the maximum anticipated flow rate for system,  $Q_M$ , to the design basis rate,  $Q_D$  or  $Q_M/Q_D$ . When  $Q_M$  is not known, nor can it be anticipated, use:  $Q_M/Q_D$  of 1.1 for flow control and 1.25 for level pressure and temperature control valves to anticipate the flow rate transients as the control loop recovers from a disturbance [18].

At the maximum flow rate  $Q_M$ , the friction drop will become:

$$F_M = F_D (Q_M/Q_D)^2 \quad (4-102)$$

The *increase* in pressure drop will be:

$$\Delta F_M = F_D (Q_M/Q_D)^2 - F_D \quad (4-103)$$

$$\text{or, } \Delta F_M = F_D [(Q_M/Q_D)^2 - 1] \quad (4-104)$$

$F_D$  may not necessarily be very accurate at the design stage where final drawing dimensions for the system are being estimated. For this reason a 10% increase allowance is suggested to  $\Delta F_M$ .

**B. Allowance for possible fall off in: overall system pressure drop,  $P_s - P_e$** 

If there is an increase in system flow rate, overall system pressure drop

$$\text{PF (all)} = 0.05 P_s \quad (4-105)$$

**C. Allowance for control valve (base pressure drop at full-open position [18])**

This varies with the type and design of valve and can be obtained from the manufacturer. It is identified here as base pressure drop  $B$  for the valve itself. Using average line velocities and assuming that the control valve will be one pipe size smaller than

(continued)

**EXAMPLE 4-4—(continued)**  
the pipe line it is connected to, using average  $B$  values over a range of sizes, the  $B$  values for estimating purposes are [18]:

Control Valve Type	B (psi)
Single plug	11
Double plug	7
Cage (unbalanced)	4
Cage (balanced)	4
Butterfly	0.2
V-ball	1

Then, incorporating the requirements of A, B, and C above, the estimated overall control valve drop is

$$\text{Required } \Delta P_c = 0.05 P_s + 1.1 \left[ \left( Q_M / Q_D \right)^2 - 1 \right] + F_D + B, \text{ psi} \quad (4-106)$$

$B$  = base pressure drop for control valve with valve in wide-open position, psi (see list above).

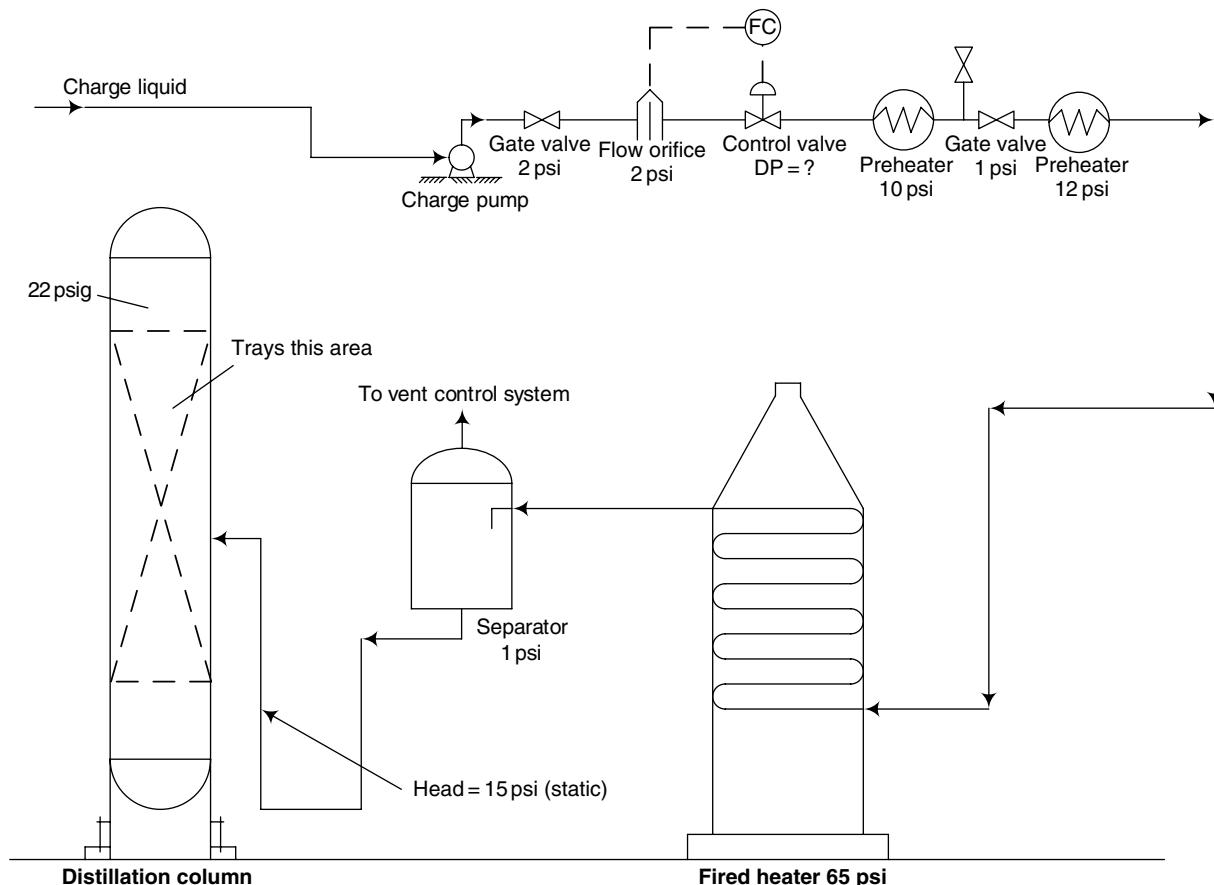


Figure 4-30 Establishing control valve estimated pressure drop.

### 4.33 THE DIRECT DESIGN OF A CONTROL VALVE

This does not require the system balance as outlined in A through C above; however, without first preparing a pressure balance, the designer cannot be confident of properly estimating the valve pressure drop. From Shinskey [20],

where

$a'$  = fractional opening of control valve, generally assume  $60\% = 0.6$

$C'_v$  = standard valve coefficient from manufacturer's catalog

$\Delta P_c$  = pressure drop across valve, psi

sp gr = specific gravity of fluid, relative to water at same temperature.

$$\text{GPM} = a' (C'_v) \sqrt{\Delta P_c / \text{sp gr}} \quad (4-107)$$

or, from [21], for gases or vapors:

**EXAMPLE 4-5****Using Figure 4-30, Determine Control Valve Pressure Drop and System Start Pressure (see Example 4-4)**

To determine  $P_s$ , the value of  $\Delta P$  through the control valve must be known.

$$P_e = 37 \text{ psi}$$

$$F_D = 96 \text{ psi (all except control valve), psi}$$

Assume  $Q_M = 120\%$  of  $Q$

$$Q_M/Q_D = 1.2$$

Use cage-type valve,  $B = 4$   
From Eq. (4-99)

$$\begin{aligned} \text{Available } \Delta P_c &= (P_s - P_e) - F_D \\ &= (P_s - 37) - 96 = P_s - 133 \end{aligned}$$

From Eq. (4-106)

$$\text{Required } \Delta P = 0.05 P_s + 1.1 \left[ (Q_M/Q_D)^2 - 1 \right] F_D + B$$

$$\begin{aligned} &= 0.05 P_s + 1.1 \left[ (1.2)^2 - 1 \right] (96) + 4 \\ &= 0.05 P_s + 46.5 + 4 = 0.05 P_s + 50 \end{aligned}$$

Substituting:

$$P_s - 133 = 0.05 P_s + 50$$

$$0.95 P_s = 183$$

$P_s = 192 \text{ psi}$ , start pressure at the pump  
Control valve pressure drop:

$$\Delta P_c = 0.05 P_s + 50 = 0.05 (192) + 50 = 59.6 \text{ psi}$$

Use this as estimated control valve pressure drop for the system design.

Flow, SCFH\*,†,

$$q'_h = \frac{42.2 C'_v \sqrt{(P_1 - P_2)(P_1 + P_2)}}{\sqrt{S_G}} \quad (4-108)$$

Flow, SCFH (temperature corrected)<sup>†</sup>

$$q'_h = \frac{963 C'_v \sqrt{(P_1 - P_2)(P_1 + P_2)}}{\sqrt{S_G T}} \quad (4-108a)$$

where

$S_G$  = specific gravity relative to air = 1.0

$P_1$  = inlet pressure (14.7 + psig)

$P_2$  = outlet pressure (14.7 + psig)

$q'_h$  = flow rate, standard ft<sup>3</sup>/h (SCFH)

$T$  = flowing temperature, °R abs (°F + 460)

$C'_v$  = valve coefficient of flow, full open (from manufacturer's tables).

2. Estimate (or use actual drawing or measured tabulation) total linear feet of pipe,  $L$ .
3. Estimate (or use actual tabulation) number of elbows, tees, crosses, globe valves, gate valves, and other fittings in system. Convert these to equivalent straight pipe using Figure 4-23 or 4-24,  $L_{eq}$ , or to head loss using Figures 4-14 through 4-18. Note preferred pipe size/type for charts.
4. Determine expansion and contraction losses (if any) from Figures 4-14, 4-17, and 4-18. Convert units: head loss in feet times 0.4331 = psi or in meters times 0.098 = bar. (This term can usually be neglected for most liquids at reasonable velocities < 10 ft/s).
5. Estimate pressure drop through orifices, control valves, and other items that may be in system, per prior discussion.
6. Total pressure drop.

$$\Delta P = (L + \sum L_{eq}) (\Delta P / 100 \text{ ft from Table 4-21}) + \text{Item (4)} + \text{Item (5)} \quad (4-97)$$

If this pressure drop is too large (or too small), recheck the steps using larger or smaller pipe as may be indicated. Table 4-37 [7] or Figure 4-28 is convenient to use, although they give much more conservative results (about twice unit head loss) than the method and figures just referenced. When using Figure 4-28 the results agree acceptably well with tests on 15- to 20-year-old steel pipe.

#### 4.34 FRICTION LOSS FOR WATER FLOW

Table 4-21 is quite convenient for reading friction loss in standard Sch. 40 pipe. It is based upon Darcy's rational analysis (equivalent to Fanning).

Suggested procedure:

1. Using known flow rate in gallons per minute, and a suggested velocity from Tables 4-10, 4-11, 4-12, 4-13, and 4-14 select an approximate line size.

\*The effect of flowing temperatures on gas flow can be disregarded for temperatures between 30° F and 150° F. Corrections should apply to other temperatures above or below [21].

† When outlet pressure  $P_2$  is less than 1/2 inlet pressure  $P_1$ , the square root term becomes 0.87  $P_1$  [21].

#### 4.35 FLOW OF WATER FROM OPEN-END HORIZONTAL PIPE

The equation of Brooke [22] is useful in estimating water or similar fluids flow from the end of open pipes:

$$GPM = 1.04 a (l) \quad (4-109)$$

where

GPM = flow rate, gpm

$a$  = internal cross-sectional area for flow in pipe, in.<sup>2</sup>

$l$  = horizontal distance from pipe opening to point where flow stream has fallen one ft, in.

TABLE 4-21 Flow of Water Through Sch. 40 Steel Pipe\*

Discharge		Pressure Drop per 100 feet and Velocity in Schedule 40 Pipe for Water at 60°F.																				
		Velocity		Press. Drop.		Velocity		Press. Drop.		Velocity		Press. Drop.		Velocity		Press. Drop.		Velocity				
Gallons per Minute	Cubic Ft. per Second	Feet	Lbs. per Second Sq. In.	Feet	Lbs. per Second Sq. In.	Feet	Lbs. per Second Sq. In.	Feet	Lbs. per Second Sq. In.	Feet	Lbs. per Second Sq. In.	Feet	Lbs. per Second Sq. In.	Feet	Lbs. per Second Sq. In.	Feet	Lbs. per Second Sq. In.	Feet	Lbs. per Second Sq. In.			
		$\frac{1}{8}''$		$\frac{1}{4}''$		$\frac{3}{8}''$		$\frac{1}{2}''$		$\frac{5}{8}''$		$\frac{3}{4}''$		$\frac{7}{8}''$		$1''$		$1\frac{1}{4}''$				
.2	0.000446	1.13	1.86	0.616	0.359	0.504	0.159	0.317	0.061	0.448	0.160	0.371	0.048	0.526	0.249	0.429	0.044	0.644	0.090	0.473	0.043	
.4	0.000668	1.69	4.22	0.924	0.903	0.672	0.345	0.422	0.086	0.743	0.164	0.743	0.164	0.864	0.429	0.844	0.164	0.936	0.158	0.563	0.071	
.6	0.000891	2.26	6.98	1.23	1.61	0.672	0.345	0.422	0.086	0.743	0.164	0.743	0.164	0.864	0.429	0.844	0.164	0.936	0.158	0.563	0.071	
.8	0.001114	2.82	10.5	1.54	2.39	0.640	0.539	0.528	0.167	0.701	0.033	0.673	0.240	0.661	0.041	0.758	0.223	0.788	0.104	0.446	0.126	
1	0.001337	3.37	14.7	1.85	2.29	1.01	0.751	0.673	0.240	0.661	0.041	0.646	0.223	0.635	0.041	0.758	0.223	0.788	0.104	0.446	0.126	
2	0.001758	4.72	25.0	2.46	4.44	1.34	1.25	0.944	0.408	0.481	0.102	1	1	1	1	1	1	1	1	1	1	
3	0.002282	11.29	134.4	9.35	64.1	5.04	13.9	3.17	4.33	1.81	1.09	1.14	0.336	0.644	0.090	0.473	0.043	0.518	1.26	0.241	0.361	
4	0.002805	12.33	111.2	6.72	23.9	4.22	7.42	2.41	1.83	1.49	0.565	0.858	0.150	0.630	0.774	1.58	0.32	0.361	0.27	0.755	0.104	
5	0.01114	2	8.40	36.7	5.28	11.2	3.01	2.75	1.86	0.835	0.102	0.758	0.223	0.788	0.104	0.446	0.126	0.446	0.126	0.446	0.126	
6	0.01337	0.574	0.044	21/2"	10.08	51.9	6.33	15.8	3.61	3.84	2.23	1.17	1.29	0.309	0.946	0.145	0.446	0.126	0.446	0.126	0.446	0.126
8	0.01782	0.763	0.073	11.44	91.1	8.45	27.7	4.81	6.60	2.97	1.99	1.72	0.518	1.26	0.241	0.361	0.27	0.755	1.58	0.32	0.361	0.27
10	0.02226	0.956	0.108	0.670	0.046	10.56	42.4	6.02	9.99	3.71	2.99	2.15	0.774	1.58	0.32	0.361	0.27	0.755	1.58	0.32	0.361	0.27
15	0.03342	1.43	0.224	1.01	0.094	3	1.01	0.056	0.056	0.03	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	
20	0.04456	1.91	0.375	1.34	0.188	0.868	0.056	0.31	0.117	0.117	0.117	0.117	0.117	0.117	0.117	0.117	0.117	0.117	0.117	0.117	0.117	
25	0.05570	2.39	0.561	1.68	0.233	1.09	0.083	0.812	0.041	4"	9.28	16.7	5.37	4.22	3.94	1.93	11.14	23.8	6.44	4.73	2.72	
30	0.06684	2.87	0.786	2.01	0.327	1.30	0.114	0.974	0.056	4"	11.14	23.8	5.32	7.31	7.90	5.32	3.64	11.14	23.8	6.44	4.73	2.72
35	0.07798	3.35	1.05	2.35	0.436	1.52	0.161	1.14	0.704	0.882	0.041	12.99	32.2	5.21	7.90	5.32	3.64	11.14	23.8	6.44	4.73	2.72
40	0.08912	3.83	1.35	2.68	0.556	1.74	0.192	1.30	0.905	1.01	0.052	14.85	41.5	5.10	10.24	6.30	4.65	9.67	12.80	7.09	5.35	3.64
45	0.10036	4.30	1.67	3.02	0.668	1.95	0.239	1.46	0.117	0.064	12.03	37.8	7.43	10.9	4.29	1.78	3.16	1.28	12.03	37.8	7.43	1.78
50	0.1114	4.78	2.03	3.15	0.839	2.17	0.288	1.62	0.142	1.30	0.076	5"	10.74	15.66	7.88	7.15	11.14	23.8	6.44	4.73	2.72	
60	0.1337	5.74	2.87	4.02	1.18	2.00	0.406	1.97	0.284	1.51	0.107	1.28	22.1	9.47	10.21	11.05	13.71	12.62	17.59	12.62	17.59	12.62
70	0.1560	6.70	3.84	4.69	1.59	3.04	0.540	2.77	0.361	1.43	0.143	1.17	0.047	11.14	23.8	6.44	4.73	2.72	12.62	17.59	12.62	
80	0.1782	7.65	4.97	5.36	2.03	3.47	0.687	2.00	0.334	2.02	0.180	1.28	0.060	6	14.20	22.0	11.14	23.8	6.44	4.73	2.72	
90	0.2005	8.60	6.09	5.25	2.53	3.91	0.861	2.92	0.416	2.27	0.224	1.44	0.074	8"	14.20	22.0	11.14	23.8	6.44	4.73	2.72	
100	0.2228	9.56	7.59	6.70	3.09	4.14	1.05	2.25	0.509	2.52	0.272	1.60	0.090	1.11	0.036	15.78	26.9	1.11	0.036	15.78	26.9	
125	0.2785	11.97	11.76	8.38	4.71	6.43	1.61	4.06	0.769	3.15	0.415	2.01	0.135	1.39	0.055	19.72	41.4	1.39	0.055	19.72	41.4	
150	0.3342	14.16	16.70	10.05	6.69	9.51	2.24	4.87	1.08	3.75	0.580	2.41	0.190	1.67	0.077	1.67	0.077	1.67	0.077	1.67	0.077	
175	0.3899	16.75	22.3	11.73	8.97	7.60	3.00	5.68	1.44	4.41	0.774	2.81	0.253	1.94	0.102	1.94	0.102	1.94	0.102	1.94	0.102	
200	0.4456	19.14	28.8	13.42	11.68	8.08	3.87	6.49	1.85	5.04	0.985	3.21	0.323	2.22	0.130	2.22	0.130	2.22	0.130	2.22	0.130	
225	0.5013	...	15.09	14.63	9.77	4.83	7.30	3.32	5.67	1.23	3.61	4.01	1.50	1.62	1.44	0.043	1.50	1.62	1.44	0.043	1.50	
250	0.557	...	10.09	14.63	9.77	4.83	7.30	3.32	5.67	1.23	3.61	4.01	1.50	1.62	1.44	0.043	1.50	1.62	1.44	0.043	1.50	
275	0.6127	...	11.94	7.14	8.93	3.40	6.93	1.79	4.41	0.41	5.83	3.05	1.24	1.76	0.061	1.24	1.76	0.061	1.24	1.76	0.061	
300	0.6684	...	13.00	8.38	8.93	3.40	6.93	1.79	4.41	0.41	5.83	3.05	1.24	1.76	0.061	1.24	1.76	0.061	1.24	1.76	0.061	
325	0.7241	...	14.12	9.89	10.53	4.09	4.09	8.19	2.47	5.21	0.797	7.61	0.320	2.08	2.08	1.11	0.075	1.11	0.075	1.11	0.075	
350	0.7798	...	...	...	...	11.36	5.41	8.82	2.84	5.62	0.919	7.89	0.367	2.24	0.095	2.24	0.095	2.24	0.095	2.24	0.095	
375	0.8355	...	...	...	...	12.17	5.41	8.82	2.84	5.62	0.919	7.89	0.367	2.24	0.095	2.24	0.095	2.24	0.095	2.24	0.095	
400	0.8912	...	...	...	...	12.98	7.03	10.08	3.68	6.42	1.19	4.44	0.471	2.56	0.121	2.56	0.121	2.56	0.121	2.56	0.121	
425	0.9469	...	...	...	...	13.80	8.89	10.71	4.02	7.56	2.11	4.81	0.683	3.33	0.275	1.92	0.072	1.92	0.072	1.92	0.072	
450	1.003	10"	...	...	...	14.91	8.80	11.34	4.60	8.00	2.47	5.21	0.797	7.61	0.360	2.24	0.095	2.24	0.095	2.24	0.095	
475	1.059	1.93	0.054	...	...	...	...	...	...	11.97	5.12	6.84	0.853	3.04	0.166	1.66	0.166	1.66	0.166	1.66	0.166	
500	1.114	2.03	0.059	...	...	...	...	...	...	12.60	5.85	8.02	1.81	5.55	0.720	3.11	0.181	3.11	0.181	3.11	0.181	
550	1.225	2.24	0.071	...	...	...	...	...	...	12.85	6.79	8.82	2.17	6.11	0.861	3.53	0.219	3.53	0.219	3.53	0.219	
600	1.337	2.44	0.083	...	...	...	...	...	...	13.12	8.04	9.63	2.55	6.69	1.02	3.85	0.258	3.85	0.258	3.85	0.258	
650	1.448	2.64	0.097	12"	...	...	...	...	...	10.43	2.98	7.22	1.18	4.17	0.301	2.08	0.130	2.08	0.130	2.08	0.130	
700	1.560	2.86	0.112	2.01	0.047	...	...	...	...	11.23	3.43	7.76	1.35	4.49	0.343	2.24	0.130	2.24	0.130	2.24	0.130	
750	1.671	3.05	0.127	2.15	0.054	14"	...	...	...	12.01	3.92	8.33	1.55	4.81	0.392	2.24	0.130	2.24	0.130	2.24	0.130	
800	1.782	3.25	0.143	2.20	0.061	...	...	...	...	12.87	4.43	8.88	1.75	5.13	0.443	2.24	0.130	2.24	0.130	2.24	0.130	

**EXAMPLE 4-6****Water Flow in Pipe System**

The system of Figure 4-31 consists of 125 ft of unknown size Sch. 40 steel pipe on the discharge side of a centrifugal pump. The flow rate is 500 gpm at 75°F. Although the tank is located above the pump, note that this elevation difference does not enter into the pipe size-friction drop calculations. However, it will become a part of selection of the pump for the service (see Chapter 5). For quick estimate follow these steps:

- From Table 4-6, select 6 fps as a reasonable and usually economical water rate.

From Table 4-21, a 6-in. pipe has a velocity of 5.55 fps at 500 gpm and a head loss of 0.72 psi/100 ft. The 5-in. pipe has a velocity of 8.02 fps and might be considered; however 5-in. pipe is not commonly stocked in many plants and the velocity is above usual economical pumping velocities. Use the 6-in. pipe (rough estimate).

- Linear feet of straight pipe,  $L = 125$  ft.
- From Figure 4-24, the equivalent length of fitting is 6-in.-90° ell  $\cong 14$  ft straight pipe (using medium sweep elbow to represent a welding elbow). Note that this is given as 6.5 ft from Figure 4-25. This illustrates the area of difference in attempting to obtain close or exact values.

Three 90° ell = 3 (14) =  $L_{eq} = 42$  ft (conservative)

One tee = 1 (12) =  $L_{eq} = 12$  (run of standard tee)

One 6 in. open gate valve = (1) (3.5) =  $L_{eq} = 3.5$

One sudden enlargement in tank at  $d/d' = 0$ ; = 10 ft, Figure 4-25

Total  $L_{eq} = 67.5$  ft

- Neglect expansion loss at entrance to tank, since it will be so small.
- No orifice or control valves in system.
- From Table 4-21, at 500 gpm, loss = 0.72 psi/100 eq. ft

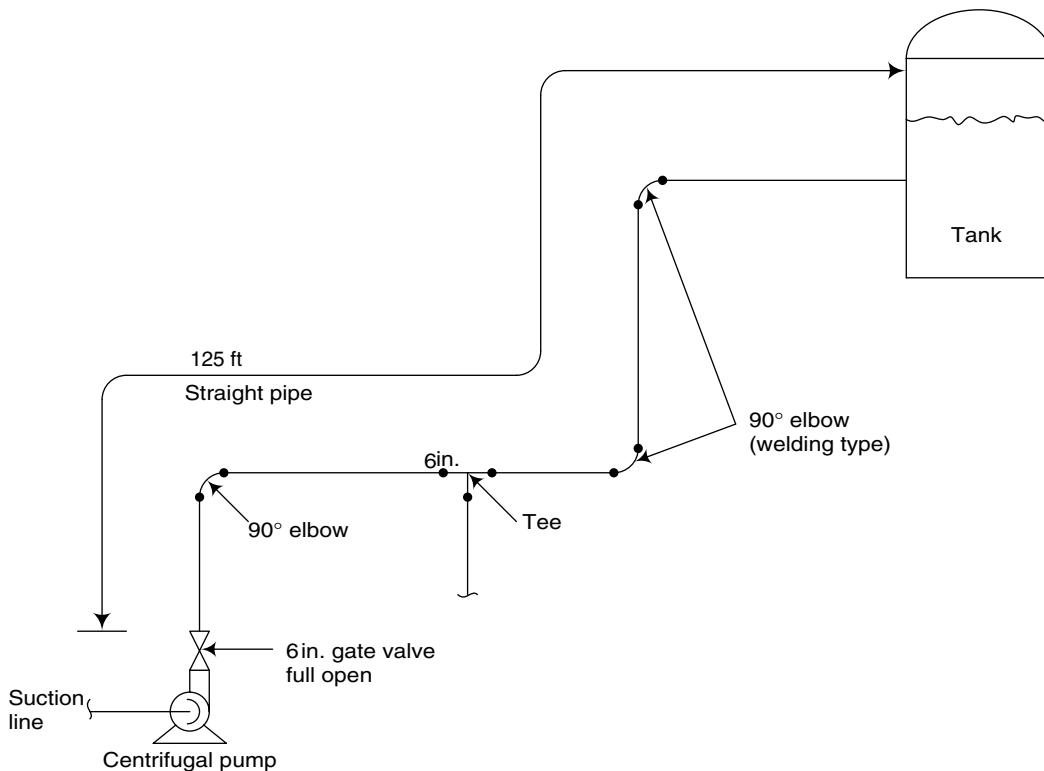
Total pressure drop from face to discharge flange on pump to nozzle connection on tank:

$$\Delta P = (125 + 67.5) [0.720/100] + 0$$

$$\Delta P = 1.386 \text{ psi}$$

$$\Delta P = 1.386 \text{ psi} (2.31 \text{ ft/psi}) = 3.20 \text{ ft water}$$

Note that a somewhat more accurate result may be obtained by following the detailed loss coefficients given in Figures 4-14–4-18. However, most preliminary engineering design calculations for this type of water system do not warrant the extra detail.



**Figure 4-31** Pipe system for pipe sizing calculations (Example 4-6).

### 4.36 WATER HAMMER [23]

Water hammer is an important problem that occurs in some liquid control systems. It is defined as hydraulic shock that occurs when a non-viscous liquid flowing in a pipe experiences a sudden change in velocity, such as the fast closing of a valve. The kinetic energy of the moving mass of liquid upon sudden stoppage or abrupt change of direction is transformed into pressure energy, thereby causing an abrupt pressure rise in the system, often resulting in severe mechanical damage [7].

The pressure that can develop from the shock wave can be destructive to the containing system hardware, particularly in long pipe. Examples of conditions that can develop water hammer are

1. start, stop, or an abrupt change in a pump's speed
2. power failure
3. rapid closing of a valve (usually a control valve, which can slam shut in one or two seconds).

The magnitude of this shock wave can be expressed [23, 24]:

$$h_{wh} = a_w (v_w) / g \\ = \frac{4660 (v_w)}{g \sqrt{1 + K_{hs} B_r}} \quad (4-110)$$

For water:

$$a_w = \frac{4660}{(1 + K_{hs} B_r)^{1/2}}, \text{ ft/s} \quad (4-111)$$

where

$h_{wh}$  = maximum pressure developed by hydraulic shock, ft of water

$v_w$  = reduction in velocity, ft/s (actual flowing velocity, ft/s)

$g$  = acceleration due to gravity, 32.2 ft/s<sup>2</sup>

$K_{hs}$  = ratio of elastic modulus of water to that of the pipe material (see table below)

$B_r$  = ratio of pipe diameter (ID) to wall thickness  
 $a_w$  = velocity of propagation of elastic vibration in the discharge pipe, ft/s.

Some typical  $K_{hs}$  values of water/metal are [23]

Metal	$K_{hs}$
Copper	0.017
Steel	0.100
Brass	0.017
Wrought iron	0.012
Malleable cast iron	0.012
Aluminium	0.03

The time interval  $t_s$  required for the pressure wave to travel back and forth in the pipe is

$$t_s = 2L/a_w, \text{ s} \quad (4-112)$$

$L$  = length of pipe, ft (not equivalent ft)

When the actual abrupt closing of a device to stop the flow has a time shorter than  $t_s$ , then the maximum pressure,  $h_{wh}$ , will be exerted on the closed device and line.

Note that the value of  $h_{wh}$  is added to the existing static pressure of the system.

### 4.37 FRICTION PRESSURE DROP FOR COMPRESSIBLE FLUID FLOW

#### VAPORS AND GASES

In contrast to non-compressible fluids, the densities of gases and vapors cannot be considered constant during flow through pipes as represented by the ideal gas law

$$V = RT/P \quad \text{or} \quad PV = RT \quad (4-113)$$

$$\rho = \frac{PM_w}{RT} \quad (4-114)$$

#### EXAMPLE 4-7 Water Hammer Pressure Development

An 8-in. process pipe for transferring 2000 gpm of methanol of sp gr = 0.75 from the manufacturing plant site to a user plant location is 2000 ft long, and the liquid is flowing at 10.8 ft/s.

Maximum pressure developed (preliminary solution) when an emergency control valve suddenly closes:

$$h_{wh} = a_w (v_w) g \\ = \frac{4660 (v_w)}{g \sqrt{1 + K_{hs} B_r}} \quad (4-110)$$

Since methanol has many properties similar to water:

$$a_w = \frac{4660}{(1 + K_{hs} B_r)^{1/2}} \\ = \frac{4660}{[1 + 0.01 (24.7^*)]^{1/2}} = 4175 \text{ ft/s}$$

\* For 8-in. standard pipe,  $B_r = 7.981/0.322 = 24.78$

Time interval for pressure wave travel:

$$t_s = 2L/a_w = 2(2000)/4175 = 0.95 \text{ s} \quad (4-112)$$

If the shutoff time for the valve (or a pump) is less than 0.95 s, the water hammer pressure will be:

$$h_{wh} = 4175(10.8)/32.2 = 1400 \text{ ft of methanol} \\ = (1400)/[(2.31)/0.75)] = 454 \text{ psi hydraulic shock}$$

Then total pressure on the pipe system

$$= 454 + (\text{existing pressure from process/or pump})$$

This pressure level would most likely rupture an 8-in. Sch. 40 pipe. For a more exact solution, refer to specialty articles on the subject.

**EXAMPLE 4-8****Pipe Flow System With Liquid of Specific Gravity Other Than Water**

This is illustrated by a line sizing sheet (Figure 4-32).

Figure 4-33 represents a liquid reactor system discharging crude product similar to glycol through a flow control valve and orifice into a storage tank. The reactor is at 350 psig and 280°F with the liquid of 0.93 specific gravity and 0.91 centipoise viscosity. There is essentially no flashing of liquid across the control valve.

Flow rate: 11,000 lb/h

$$\text{GPM actual} = 11,000/(60)(8.33)(0.93) = 23.7$$

$$\text{Design rate} = 23.7(1.05) = 25 \text{ gpm}$$

1. From Table 4-10, selected velocity = 6 fps.

$$\begin{aligned} \text{Estimated pipe diameter, } d &= (0.408 Q/v)^{1/2} \\ &= [(0.408)25/6]^{1/2} = 1.3 \text{ in.} \end{aligned}$$

Try 1½ in. (ID = 1.61), since 1¼ in. (ID = 1.38) is not stocked in every plant. If it is an acceptable plant pipe size, then it should be considered and checked, as it would probably be as good pressure drop-wise as the 1½-in. The support of 1¼-in. pipe may require shorter support spans than the 1½ in. Most plants prefer a minimum of 1½-in. valves on pressure vessels, tanks, and so on. The valves at the vessels should be 1½ in. even though the pipe might be 1¼ in. The control valve system of gate and globe valves could very well be 1¼ in. For this example, use 1½-in. pipe, Sch. 40:

2. Linear length of straight pipe,  $L = 254$  ft.

3. Equivalent lengths of fittings, valves, and so on.

Estimated Fittings	Type	(From Figure 4-24)
10	1½" – 90° elbows	4'(10) = 40
8	1½" – Tees	3'(8) = 24
4	1½" – Gate valves	1'(4) = 4
68' (Rounded total to 75')		

4. No expansion or contraction losses (except control valve).

5. Pressure drop allowance assumed for orifice plate = 5 psig.

Control valve loss will be by difference, trying to maintain minimum 60% of pipe friction loss as minimum drop through valve, but usually not less than 10 psi.

$$6. \text{ Reynolds number, } Re = 50.6Q\rho/d\mu \quad (4-27)$$

$$\begin{aligned} &= \frac{50.6(25)[0.93(62.3)]}{(1.61)(0.91)} \\ &= 50,025 \text{ (turbulent)} \end{aligned}$$

7. From Figure 4.13,  $\varepsilon/d = 0.0012$  for 1½-in. steel pipe.

From Figure 4-5, at  $Re = 50,025$ , read  $f = 0.021$  or use Eq. (4-35).

8. Pressure drop per 100 ft of pipe:

$$\begin{aligned} \Delta P \text{ 100 ft} &= 0.0216 f Q^2 d^5 \quad (4-65) \\ &= 0.0216 (0.021) (62.3) (0.93) (25)^2 / (1.61)^5 \\ &= 1.52 \text{ psi/100 eq.ft} \end{aligned}$$

9. Total pressure drop

The control valve must be sized to take the residual pressure drop, as long as it is an acceptable minimum. Pressure drop accounted for:

$$\text{Total psi drop} = (245 + 75)(1.52/100) + 5 = 10 \text{ psi}$$

Drop required across control valve:

Reactor	= 350 psig
Storage	= 0 psig
Differential	= 350 psi
$\Delta P$	= 10 psi (sys. friction)
Control Valve $\Delta P$	= 340 psi

Note that this control valve loss exceeds 60% of this system loss, since the valve must take the difference. For other systems where this is not the situation, the system loss must be so adjusted as to assign a value (see earlier section on control valves) of approximately 10–20 psi or 25–60% of the system other than friction losses through the valve. For very low pressure systems, this minimum value of control valve drop may be lowered with the sacrifice of sensitive control.

where

$$V = \text{molar volume, ft}^3/(\text{lb mol})$$

$$\rho = \text{density, lb/ft}^3$$

$$M_w = \text{molecular weight, lb/(lb mol)}$$

$$P = \text{pressure, lb}_f/\text{ft}^2$$

$$T = \text{temperature, } {}^\circ\text{R} = ({}^\circ\text{F} + 460)$$

$$R = \text{universal gas constant, } 1545 (\text{ft lb}_f)/[{}^\circ\text{R (lb mol)}].$$

The effect of a change in pressure or temperature on the density of a gas causes the determination of pressure loss experienced by flowing compressible fluids in conduits, orifices, and nozzles to differ from that for non-compressible fluids.

The first step in analyzing the flow of a compressible fluid is to determine which type of polytropic process is being considered,

especially whether it is an isothermal or an adiabatic process. In an isothermal process, the gas flows at a constant temperature through a pipe and an ideal gas follows the relationship

$$PV = \text{constant (isothermal)} \quad (4-115)$$

To attain a constant temperature, the gas must exchange heat with its surroundings which act as a uniform temperature heat sink. The isothermal flow assumption is typically applied to most uninsulated lines carrying gases flowing at a low velocity. The lack of insulation and the low velocity allow the gas to approach thermal equilibrium with its surroundings. For example, most plant air and gas supply lines may be treated as isothermal fluids. However, this does not apply when the gas travels at high velocities and in insulated pipelines.



SHEET NO. \_\_\_\_\_

By AKC LINE SIZE SHEET Job No. \_\_\_\_\_  
Date \_\_\_\_\_ Charge No. \_\_\_\_\_  
Line No. LP - 51 Flow Sheet Drawing No. \_\_\_\_\_  
Line Description Reactor Discharge

Fluid in line	Crude Product		Temperature	280	° F
GPM (Calc.)	23.7	GPM (des.) 25	Pressure	350	psig
CFM (Calc.)		CFM (des.)	sp gr	0.93	
lb/h (Calc.)		lb/h (des)	sp vol		ft <sup>3</sup> /lb
Recommended Velocity	6.0		Viscosity	0.91	cP
	fps				

Item	Pressure Drop in psi
Pipe & Equivalent	5)
Orifice	5) = 10
Motor Valve (control)	340
Miscellaneous	
Total	350

\* Rounded total to 75 feet

\*\* By difference

Inlet = 350 psig; Outlet = 0 psig  
Friction Loss = 10 (includes orifice loss)  
Balance for Valve = 340 psi

Estimated line size 1½" (verified)

Unit Loss per 100 ft of liquid	1.52 psi (see below)
Total head loss in feet	869
Total pressure drop in psi	350

Selected pipe size	1½"	Material & Weight	Schedule 40 steel
Calculations:	$Re = 50.6 Q \rho / d \mu = 50.6 (25) (0.93 \times 62.3) / (1.61) (0.91) = 50,025$		
	$\varepsilon/d = 0.0012; f = 0.021$ (Figures 4-5 and 4-13)		
	$\Delta P/100' = 0.0216 f_p Q^2 / d^5 = 0.0216 (0.021) (62.3) (0.93) (25)^2 / (1.61)^5 = 1.52 \text{ psi}/100 \text{ ft}$		
Total Pipe System Friction	$\Delta P = [(329) (1.52/100)] + 5^* = 10 \text{ psi}$ for friction; *orifice		
Total Loss, Feet Liquid	$= 350 (2.31 \text{ ft/psi}) (1/0.93) = 869 \text{ ft of Liquid}$		

Checked by: \_\_\_\_\_ Date: \_\_\_\_\_

**Figure 4-32** Line sizing sheet for example problem (Example 4-8).

Gases and vapors flowing through measuring devices such as orifices, through pipes at high velocities or in insulated pipelines cannot maintain a constant temperature. Thus the condition should not be treated as an isothermal process. This is because the effect of velocity acceleration due to changes in flow cross-sectional area and the effect of friction create changes in temperature. High gas velocities and insulation prevent the compressible fluid from attaining thermal equilibrium with its surroundings. When heat cannot be exchanged with the surroundings, compressible flow conditions may be assumed to be essentially an adiabatic process, and ideal fluids then follow the relationship

$$PV^k = \text{constant (adiabatic)} \quad (4-116)$$

where

$$k = \text{specific heat ratio} = C_p/C_v = C_p/(C_p - R)$$

$C_p$  = specific heat at constant pressure

$C_v$  = specific heat at constant volume

$R$  = universal gas constant, 1544/M<sub>w</sub> (ft.lbf)/[° R.lbmol].

Comparing Eqs (4-115) and (4-116), it can be seen that the isothermal process is equivalent to having an exponent  $k$  of unity for the volume term. The isothermal and adiabatic processes are special cases of polytropic fluid flow process. The isothermal process represents thermal equilibrium with the environment, while the adiabatic process illustrates no heat exchange with the environment. Actual flow processes can often be approximated by these extremes, but

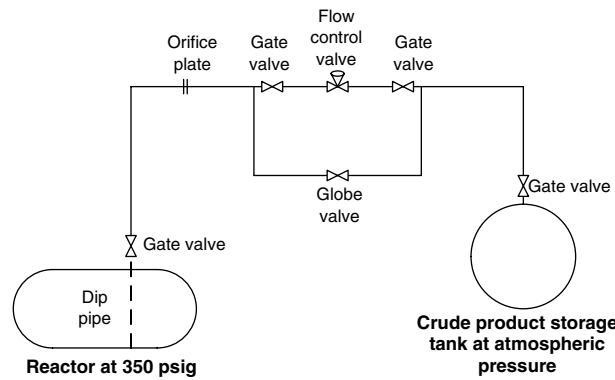


Figure 4-33 Liquid flow system (Example 4-8).

there are other situations which can be represented by the general relationship

$$PV^n = \text{constant} \quad (\text{polytropic}) \quad (4-117)$$

where  $n$  = polytropic coefficient. The exponent  $n$  can be substituted for  $k$  in Eq. (4-116) and can have a value between zero and infinity, depending on the amount of heat added or removed from the flowing system. In the special polytropic process known as the adiabatic case, heat is neither added nor removed, so that  $n = k$ . In the isothermal process, sufficient heat is added to a flowing system to maintain it at constant temperature and  $n$  is unity. For other polytropic processes, the value of  $n$  is in the range

$$(\text{Isothermal}) 1 < n < k \quad (\text{adiabatic})$$

#### 4.38 COMPRESSIBLE FLUID FLOW IN PIPES

The flow of compressible fluids (e.g., gases and vapors) through pipelines and other restrictions is often affected by changing conditions of pressure, temperature, and physical properties. The densities of gases and vapors vary with temperature and pressure ( $PV = \text{constant}$ ). Conversely, in adiabatic flow, that is no heat loss ( $PV^k = \text{constant}$ ), a decrease in temperature occurs when pressure decreases, resulting in a density increase. At high pressures and temperatures, the compressibility factor can be less than unity, which results in an increase in the fluid density.

The condition of high pressure drop ( $\Delta P$ ) in compressible flow frequently occurs in venting systems, vacuum distillation equipment, and long pipelines. Some design situations involve vapor flows at very high velocities resulting in  $\Delta P > 10\%$  of the upstream pressure. Such cases are vapor expanding through a valve, high speed vapor flows in narrow pipes, and vapors flowing in process lines under vacuum conditions.  $\Delta P$ , in many cases, is critical and requires accurate analysis and design. For instance, the inlet pipe  $\Delta P$ , of a safety relief valve should not exceed 3% of the relief valve set pressure (gauge) at its relieving capacity for stable operation. This limit is to prevent the rapid opening and closing of the valve, resulting in lowered fluid capacity and subsequent damage of the valve seating surfaces. Conversely, the tail pipe or vent line of a relief valve should be designed such that  $\Delta P < 10\%$  of the relieving pressure, that is set pressure + over pressure in gauge. Figure 4-34 shows a typical tail pipe and relief valve connected to a heat exchanger.

#### 4.39 MAXIMUM FLOW AND PRESSURE DROP

Determining the maximum fluid flow rate or pressure drop for process design often has the dominant influence on density. As

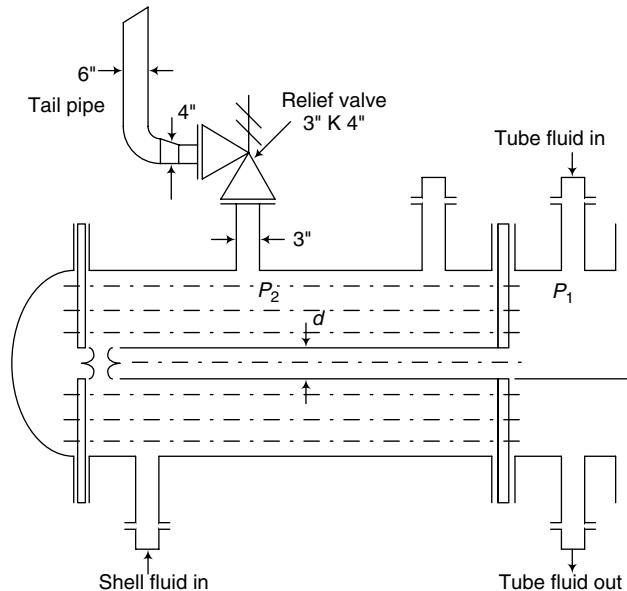


Figure 4-34 Fluid flow through a heat exchanger, relief valve, and tail pipe.

pressure decreases due to piping and component resistance, the gas expands and its velocity increases. A limit is reached when the gas or velocity cannot exceed the sonic or critical velocity. Even if the downstream pressure is lower than the pressure required to reach sonic velocity, the flow rate will still not increase above that evaluated at the critical velocity. Therefore, for a given pressure drop ( $\Delta P$ ), the mass discharge rate through a pipeline is greater for an adiabatic condition (i.e., insulated pipes, where heat transfer is nil) than the rate for an isothermal condition by as much as 20%. There is, however, no difference if the pipeline is more than 1000 pipe diameters long [25]. In practice, the actual flows are between the two conditions, and the inflow rates are often below 20% even for lines less than 1000 pipe diameters.

#### 4.40 SONIC CONDITIONS LIMITING FLOW OF GASES AND VAPORS

The flow rate of a compressible fluid in a pipe with a given upstream pressure will approach certain maximum rate that it cannot exceed even with reduced downstream pressure. The maximum possible velocity of a compressible fluid in a pipe is sonic (speed of sound) velocity, as:

$$v_s = \sqrt{kgZRT} = \sqrt{kg144P'\bar{V}}, \text{ ft/s} \quad (4-118)$$

$$v_s = [(C_p/C_v)(32.2)(Z)(1544/M_w)(460+t)]^{1/2} \quad (4-119)$$

$$= 68.1[(C_p/C_v)P'/\rho]^{1/2}, \text{ ft/s} \quad (4-120)$$

where

$k$  = ratio of specific heat at constant pressure to specific heat at constant volume ( $C_p/C_v$ )

$C_p$  = specific heat at constant pressure

$C_v$  = specific heat at constant volume

$R$  = individual gas constant =  $MR/M_w = 1544/M_w$

$M_w$  = molecular weight of the gas

$MR$  = universal gas constant = 1544

$T$  = temperature of gas,  $R = (460+t)$

$t$  = temperature, °F

$P'$  = pressure, psia

$v_s$  = sonic or critical velocity of flow of a gas, ft/s

$\bar{V}_1$  = specific volume of fluid,  $\text{ft}^3/\text{lb}$  ( $\text{m}^3/\text{kg}$ ) at  $T$  and  $P'$

$g$  = acceleration of gravity =  $32.2 \text{ ft/s}^2$

$Z$  = compressibility factor.

In SI units,

$$v_s = \sqrt{\gamma ZRT} = \sqrt{\gamma P' \bar{V}} = 316.2 \sqrt{\gamma p' \bar{V}}, \text{ m/s} \quad (4-121)$$

where

$R$  = Individual gas constant =  $8314/M_w, \text{ J/kg K}$

$M_w$  = molecular weight of the gas

$\gamma$  = ratio of specific heat at constant pressure to specific heat at constant volume =  $C_p/C_v$

$T$  = absolute temperature,  $K = 273 + t$

$t$  = temperature,  $^\circ\text{C}$

$P'$  = pressure,  $\text{N/m}^2$

$p'$  = pressure, bar

$\bar{V}$  = specific volume of fluid,  $\text{m}^3/\text{kg}$

$W$  = fluid flow rate,  $\text{kg/h}$

$\rho$  = density of fluid,  $\text{kg/m}^3$  at  $T$  and  $P'$  (or  $p'$ ).

Thus the maximum flow in a pipe occurs when the velocity at the exit becomes sonic. The sonic location may be other than the exit, can be at restrictive points in the system, at expansion points in the system (e.g., an expander from a 2-in. to 3-in. pipe or a 6 in.-pipe into a larger header), or at control/safety relief valves.

A recommended compressible fluid velocity for trouble-free operation is  $v \leq 0.6v_s$ . The design criteria for compressible fluid process lines (e.g., carbon steel) are shown in Table 4-22.

Shock waves are stationary, however if they were to travel, it would be at sonic velocity and exhibit a near discontinuity in pressure, density, and temperature, and a great potential exists for damage from such waves [26]. A discussion of shock waves is beyond the scope of this chapter.

Velocity considerations are important in rotating or reciprocating machinery systems, because if the compressible fluid velocity exceeds the speed of sound in the fluid, shock waves can be set up and the results of such conditions are much different than the velocities below the speed of sound. The ratio of the actual fluid velocity to its speed of sound is called the Mach number [27].

The velocity of sound at  $68^\circ\text{F}$  in air is  $1126 \text{ ft/s}$ .

For any gas, the speed of sound is

$$v_s = \sqrt{kgp''/\rho}, \text{ ft/s} \quad (4-122)$$

where

$k$  = ratio of specific heat of gas, at constant pressure to that at constant volume,

=  $C_p/C_v$  (Table 4-23)

$g = 32.2 \text{ ft/s}^2$

$p''$  = pressure, pounds per square feet, absolute (Psfa) (note units)

$\rho$  = the specific weight,  $\text{lb}/\text{ft}^3$  at  $T$  and  $p''$ .

In SI units,

$$v_s = \sqrt{\gamma P'/\rho} \quad (4-123)$$

This sonic velocity occurs in a pipe system in a restricted area (e.g., valve, orifice, venturi) or at the outlet end of the pipe (open-ended), as long as the upstream pressure is high enough. The physical properties in the above equations are at the point of maximum velocity.

**TABLE 4-22 Recommended Fluid Velocity and Maximum  $\Delta P$  for Carbon Steel Vapor Lines**

Type of Service	Recommended Velocity (ft/s)	Maximum $\Delta P$ (psi/100 ft)
<b>1. General Recommendation</b>		
Fluid pressure in psig		
Sub-atmospheric	0.18	
$0 < P \leq 50$	0.15	
$50 < P \leq 150$	0.30	
$150 < P \leq 200$	0.35	
$200 < P \leq 500$	1.0	
$P > 500$	2.0	
<b>2. Tower overhead</b>		
Pressure ( $P > 50 \text{ psia}$ )	40–50	0.2–0.5
Atmospheric	60–100	
Vacuum ( $P < 10 \text{ psia}$ )		0.05–0.1
<b>3. Compressor piping suction</b>		
	75–200	0.5
<b>4. Compressor piping discharge</b>		
Gas lines with battery limits		
Refrigerant suction lines	15–35	0.5
Refrigerant discharge lines	35–60	
<b>Steam lines</b>		
<b>1. General recommendation</b>		
Maximum: Saturated superheated	200	
Steam pressure in psig	250	
0–50	167	0.25
50–150	117	0.40
150–300		1.0
>300		1.5
<b>2. High pressure steam lines</b>		
Short ( $L < 600 \text{ ft}$ )		1.0
Long ( $L > 600 \text{ ft}$ )		0.5
<b>3. Exhaust steam lines (<math>P &gt;</math> atmosphere)</b>		
Leads to exhaust header		0.5
<b>4. Relief valve discharge</b>		
Relief valve, entry point at silencer	$0.5 v_s$	$v_s$

With a high velocity vapor flow, the possibility of attaining critical or sonic flow conditions in a process pipe should be investigated. These occur whenever the resulting pressure drop approaches the following values of  $\Delta P$  as a percentage of the upstream pressure [6] where the properties are evaluated at the condition of sonic flow.

This applies regardless of the downstream pressure for a fixed upstream pressure. This limitation must be evaluated separately from pressure drop relations, as it is not included as a built-in limitation.

Sonic velocity will be established at a restricted point in the pipe or at the outlet, if the pressure drop is great enough to establish the required velocity. Once the sonic velocity has been reached, the flow rate in the system will not increase, as the velocity will remain at this value even though the fluid may be discharging into a vessel at a lower pressure than that existing at the point where sonic velocity is established.  $\Delta P$  can be increased by continuing to lower the discharge pressure. But no additional flow rate will result. The usual pressure drop equations do not hold at the sonic velocity, as in an orifice. Conditions or systems exhausting to atmosphere (or vacuum) from medium-to-high pressures should be examined

**TABLE 4-23 Approximate  $k$  Values for Some Common Gases (68° F, 14.7 psia)**

Gas	Chemical Formula or Symbol	Approximate Molecular Weight	$k(C_p/C_v)$
Acetylene (ethyne)	$C_2H_2$	26.0	1.30
Air	—	29.0	1.40
Ammonia	$NH_3$	17.0	1.32
Argon	A	39.9	1.67
Butane	$C_4H_{10}$	58.1	1.11
Carbon dioxide	$CO_2$	44.0	1.30
Carbon monoxide	CO	28.0	1.40
Chlorine	$Cl_2$	70.9	1.33
Ethane	$C_2H_6$	30.0	1.22
Ethylene	$C_2H_4$	28.0	1.22
Helium	He	4.0	1.66
Hydrogen chloride	HCl	36.5	1.41
Hydrogen	$H_2$	2.0	1.41
Methane	$CH_4$	16.0	1.32
Methyl chloride	$CH_3Cl$	50.5	1.20
Natural gas	—	19.5	1.27
Nitric oxide	NO	30.0	1.40
Nitrogen	$N_2$	28.0	1.41
Nitrous oxide	$N_2O$	44.0	1.31
Oxygen	$O_2$	32.0	1.40
Propane	$C_3H_8$	44.1	1.15
Propylene (propene)	$C_3H_6$	42.1	1.14
Sulfur dioxide	$SO_2$	64.1	1.26

for critical flow, otherwise the calculated pressure drop may be in error.

All flowing gases and vapors (compressible fluids) including steam (which is a vapor) are limited or approach a maximum in mass flow velocity or rate, that is lb/s or lb/h (kg/s or kg/h), through a pipe depending upon the specific upstream or starting pressure. This maximum rate of flow cannot be exceeded regardless of how much the downstream pressure is further reduced. The mean velocity of fluid flow in a pipe by continuity equation is [4]

$$v = \frac{0.0509 W \bar{V}}{d^2} \text{ or } \frac{0.0509 W}{\rho d^2}, \text{ ft/s} \quad (4-124)$$

In SI units,

$$v = \frac{354 W \bar{V}}{d^2} \text{ or } v = \frac{354 W}{\rho d^2}, \text{ m/s} \quad (4-125)$$

where

$d$  = internal pipe diameter, in. (mm)

$W$  = rate of flow, lb/h (kg/h)

$\bar{V}$  = specific volume of fluid,  $\text{ft}^3/\text{lb}$  ( $\text{m}^3/\text{kg}$ )

$\rho$  = fluid density,  $\text{lb}/\text{ft}^3$  ( $\text{kg}/\text{m}^3$ ).

This maximum velocity of a compressible fluid in a pipe is limited by the velocity of propagation of a pressure wave that travels at the speed of sound in the fluid [4]. This speed of sound is specific for each individual gas or vapor and is a function of the ratio of specific heats of the fluid. The pressure reduces and the velocity increases as the fluid flows downstream through the pipe, with the maximum velocity occurring at the downstream end of the pipe. When or if the pressure drop is great enough, the discharge or exit or outlet velocity will reach the velocity of sound for that fluid.

If the outlet or discharge pressure is lowered further, the pressure upstream at the origin will not detect it because the pressure

wave can only travel at sonic velocity. Therefore, the change in pressure downstream will not be detected upstream. The excess pressure drop obtained by lowering the outlet pressure after the maximum discharge has been reached takes place beyond the end of the pipe [4]. This pressure is lost in shock waves and turbulence of the jetting fluid. See [26, 28–30] for further expansion of shock waves and detonation waves through compressible fluids.

In the case of a high pressure header, the flow may be sonic at the exit. Therefore, it is often necessary to check that the outlet pressure of each pipe segment is not critical. If  $P_c$  is less than terminal  $P_2$ , the flow is subcritical. If, however,  $P_c$  is greater than  $P_2$ , then the flow is critical. Although, it may be impractical to keep the flow in high pressure subheaders below sonic, Mak [31] suggests that the main flare header should not be sized for critical flow at the outlet of the flare stack. This would obviate the undesirable noise and vibration resulting from sonic flow. Crocker's [32] equation for critical pressure can be expressed as:

$$P_c = \frac{G}{11,400 d^2} \left( \frac{RT}{k[k+1]} \right)^{1/2}, \text{ psia} \quad (4-126)$$

or

$$P_c = 2.45 \times 10^{-3} \left( \frac{ZG}{d^2} \right) \left( \frac{T}{kM_w} \right)^{0.5} \quad (4-127)$$

where

$$R = \left( \frac{1544}{29 \text{ sp gr}} \right), \text{ molar gas constant} \quad (4-128)$$

$$\text{sp gr} = \frac{\text{Molecular weight of gas}}{\text{Molecular weight of air}}$$

$Z$  = compressibility factor

$d$  = internal pipe diameter, in.

$M_w$  = fluid molecular weight

$G$  = fluid flow rate, lb/h

$T$  = fluid temperature, °R

$\rho$  = fluid density,  $\text{lb}/\text{ft}^3$ .

#### 4.41 THE MACH NUMBER, $Ma$

The Mach number,  $Ma$ , is the velocity of the gas divided by the velocity of the sound in the gas and can be expressed as:

$$Ma = v/v_s \quad (4-129)$$

The exit Mach number for compressible isothermal fluid has been shown to be  $Ma \neq 1$ , but  $1/\sqrt{k}$ , where  $k$  is the ratio of the fluid-specific heat capacities at constant pressure to that at constant volume. Table 4-23 shows the  $k$  values for some common gases. The following cases are such:

$$1. Ma < 1/\sqrt{k}, \text{ the flow is subsonic} \quad (4-129a)$$

$$2. Ma = 1/\sqrt{k}, \text{ the flow is sonic} \quad (4-129b)$$

$$3. Ma > 1/\sqrt{k}, \text{ the flow is supersonic} \quad (4-129c)$$

Case 3 is produced under certain operating conditions in the throttling processes (e.g., a reduction in the flow cross-sectional area). Kirkpatrick [33] indicates that there is a maximum length for which continuous flow is applied for an isothermal condition, and this corresponds to  $Ma = 1/\sqrt{k}$ . The limitation for isothermal flow, however, is the heat transfer required to maintain a constant temperature. Therefore, when  $Ma < 1/\sqrt{k}$ , heat must be added to the stream to maintain constant temperature. For  $Ma > 1/\sqrt{k}$ , heat must be rejected from the stream. Depending on the ratio of specific heats, either condition could occur with subsonic flow. Therefore, to maintain isothermal flow during heat transfer, high temperatures require high Mach numbers and low temperatures require low Mach numbers.

#### 4.42 MATHEMATICAL MODEL OF COMPRESSIBLE ISOTHERMAL FLOW

The derivations of the maximum flowrate and pressure drop of compressible isothermal flow are based on the following assumptions:

1. Isothermal compressible fluid.
2. No mechanical workdone on or by the system.
3. The gas obeys the perfect gas laws.
4. A constant friction factor along the pipe.
5. Steady state flow or discharge unchanged with time.

Figure 4-35 illustrates the distribution of fluid energy with work done by the pump and heat added to the system. Table 4-6 gives friction factors for clean commercial steel pipes with flow in zones of complete turbulence.

#### 4.43 FLOW RATE THROUGH PIPELINE

Bernoulli's equation for the steady flow of a fluid can be expressed as:

$$\int_1^2 \frac{dP}{\rho} + \alpha \frac{\Delta v^2}{2g_c} + \frac{g}{g_c} \Delta z + \delta e_f + \delta W_s = 0 \quad (4-130)$$

where  $\alpha$  (the dimensionless velocity distribution) = 1 for turbulent or plug flow. Assuming no shaft work is done (e.g.,  $\delta W_s = 0$ ), then Eq. (4-130) becomes

$$\frac{P_2 - P_1}{\rho} + \alpha \frac{v_2^2 - v_1^2}{2g_c} + \frac{g}{g_c} (z_2 - z_1) + e_f = 0 \quad (4-131)$$

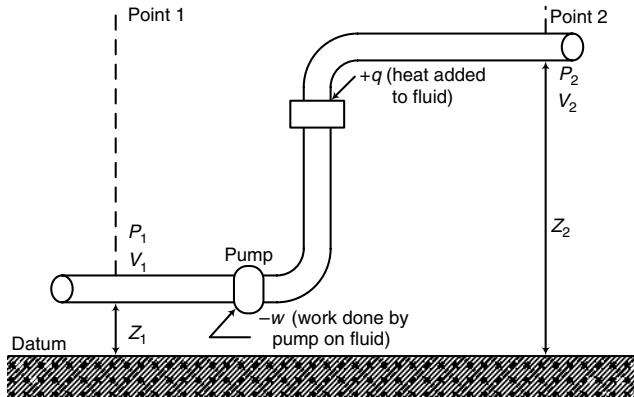


Figure 4-35 Energy aspects of a single-stream piping system.

For expanding gas flow,  $v_2 \neq v_1$ ; with horizontal pipe,  $Z_2 = Z_1$ . Hence, the differential form of Bernoulli's equation can be expressed as

$$-\frac{dP}{\rho} = \frac{1}{g_c} v dv + e_f \quad (4-132)$$

Where  $\rho$  is constant and the velocity head is

$$e_f = K \frac{v^2}{2g_c} \quad (4-133)$$

The mass flowrate through the pipe is

$$G = \rho v A$$

or

$$\frac{G}{A} = \rho v = \text{constant} \quad (4-134)$$

Because both density and velocity change along the pipeline, Eq. (4-134) can be expressed in differential form as:

$$\rho dv + v d\rho = 0 \quad (4-135)$$

In expanding gas flow, the pressure and density ratio are constant:

$$\frac{P}{\rho} = \frac{dP}{d\rho} = \frac{P_1}{\rho_1} \quad (4-136)$$

and, with respect to initial condition,  $P_1$  and  $\rho_1$

$$\frac{P}{\rho} = \frac{dP}{d\rho} = \frac{P_1}{\rho_1} \quad (4-137)$$

From Eq. (4-135)

$$dv = -\frac{v}{\rho} d\rho \quad (4-138)$$

and

$$\frac{dP}{P} = \frac{d\rho}{\rho} \quad (4-139)$$

therefore

$$\frac{1}{\rho} = \frac{1}{P} \left( \frac{dP}{d\rho} \right) = \frac{1}{P} \left( \frac{P_1}{\rho_1} \right) \quad (4-140)$$

Substituting Eq. (4-139) into Eq. (4-138) gives

$$dv = -\frac{v}{\rho} \left( \frac{dP}{P} \right) \rho \quad (4-141)$$

Therefore

$$dv = -v \left( \frac{dP}{P} \right) \quad (4-142)$$

Substituting Eq. (4-142) into Eq. (4-132) gives

$$-\frac{dP}{\rho} = \frac{1}{g_c} v \left( -v \frac{dP}{P} \right) + K \frac{v^2}{2g_c} \quad (4-143)$$

$$-\frac{dP}{\rho} = -\frac{1}{g_c} v^2 \left( \frac{dP}{P} \right) + K \frac{v^2}{2g_c} \quad (4-144)$$

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Rearranging Eq. (4-144) gives

$$-dP = \frac{v^2}{g_c} \rho \left( \frac{K}{2} - \frac{dP}{P} \right) \quad (4-145)$$

From the mass flowrate,  $G = \rho v A$

$$v = \frac{G}{\rho A} \quad (4-146)$$

Substituting Eq. (4-146) into Eq. (4-145) gives

$$-dP = \frac{G^2}{g_c A^2 \rho^2} \rho \left( \frac{K}{2} - \frac{dP}{P} \right) \quad (4-147)$$

In terms of the initial conditions of pressure and density, that is substituting Eq. (4-140) into Eq. (4-147) yields

$$-dP = \left( \frac{G}{A} \right)^2 \frac{1}{g_c} \left[ \frac{1}{P} \left( \frac{P_1}{\rho_1} \right) \right] \left[ \frac{K}{2} - \frac{dP}{P} \right] \quad (4-148)$$

Integrating Eq. (4-148) between points 1 (inlet) and 2 (outlet) gives

$$-2 \int_1^2 P dP = \left( \frac{G}{A} \right)^2 \left( \frac{P_1}{\rho_1} \right) \frac{1}{g_c} \left( K - 2 \int_1^2 \frac{dP}{P} \right) \quad (4-149)$$

which is

$$P_1^2 - P_2^2 = \left( \frac{G}{A} \right)^2 \frac{P_1}{\rho_1} \left( \frac{1}{g_c} \right) \left[ K + 2 \ln \left( \frac{P_1}{P_2} \right) \right] \quad (4-150)$$

Therefore, for maximum flowrate through the pipe,

$$G = \left[ \left( \frac{A^2 \rho_1 g_c}{K_{\text{Total}} + 2 \ln \left( \frac{P_1}{P_2} \right)} \right) \left( \frac{P_1^2 - P_2^2}{P_1} \right) \right]^{0.5} \quad (4-151)$$

where  $K_{\text{Total}}$  is the total resistance (loss) coefficient due to friction, fittings, and valves.

$$K_{\text{Total}} = f_D \frac{L}{D} + \sum K_f \text{ (pipe fittings + valves)} \quad (4-152)$$

$\sum K_f$  (pipe fittings + valves) is the sum of the pressure loss coefficient for all the fittings and valves in the line. Expressing the maximum fluid rate in pounds per hour, Eq. (4-151) becomes

$$G = 1335.6 d^2 \left[ \left( \frac{\rho_1}{K_{\text{Total}} + 2 \ln \left( \frac{P_1}{P_2} \right)} \right) \left( \frac{P_1^2 - P_2^2}{P_1} \right) \right]^{0.5}, \text{ lb/h} \quad (4-153)$$

In SI units,

$$G = 2.484 \times 10^{-4} d^2 \left[ \left( \frac{\rho_1}{K_{\text{Total}} + 2 \ln \left( \frac{P'_1}{P'_2} \right)} \right) \left( \frac{(P'_1)^2 - (P'_2)^2}{P'_1} \right) \right]^{0.5}, \text{ kg/s} \quad (4-154)$$

where

$d$  = pipe internal diameter, mm

$\rho_1$  = upstream gas density,  $\text{kg/m}^3$

$P'_1$  = upstream gas pressure, bara

$P'_2$  = downstream gas pressure, bara.

## 4.44 PIPELINE PRESSURE DROP ( $\Delta P$ )

If  $\Delta P$  due to velocity acceleration is relatively small compared with the frictional drop, then  $\ln \left( \frac{P_1}{P_2} \right)$  may be neglected. Therefore Eq. (4-153) becomes

$$G = 1335.6 d^2 \left[ \frac{\rho_1}{K_{\text{Total}}} \left( \frac{P_1^2 - P_2^2}{P_1} \right) \right]^{0.5} \quad (4-155)$$

$$\text{Putting } C = 1335.6 d^2 \quad (4-156)$$

$$\frac{G^2}{C^2} = \frac{\rho_1}{K_{\text{Total}}} \frac{P_1^2 - P_2^2}{P_1} \quad (4-157)$$

That is,

$$P_1^2 - P_2^2 = \frac{P_1 G^2 K_{\text{Total}}}{\rho_1 C^2} \quad (4-158)$$

$$P_2 = \left( P_1^2 - \frac{P_1 G^2 K_{\text{Total}}}{\rho_1 C^2} \right)^{0.5} \quad (4-159)$$

Therefore, the pressure drop

$$\Delta P \cong P_1 - P_2 \quad (4-160)$$

That is,

$$\Delta P \cong P_1 - \left( P_1^2 - \frac{P_1 G^2 K_{\text{Total}}}{\rho_1 C^2} \right)^{0.5} \quad (4-161)$$

In SI units,

$$G = 2.484 \times 10^{-4} d^2 \left[ \left( \frac{\rho_1}{K_{\text{Total}} + 2 \ln \left( \frac{P'_1}{P'_2} \right)} \right) \left( \frac{(P'_1)^2 - (P'_2)^2}{P'_1} \right) \right]^{0.5} \quad (4-154)$$

$$\text{Putting } C_1 = 2.484 \times 10^{-4} d^2 \quad (4-162)$$

$$\frac{G^2}{C_1^2} = \frac{\rho_1}{K_{\text{Total}}} \frac{P_1^2 - P_2^2}{P_1} \quad (4-163)$$

That is,

$$P_1^2 - P_2^2 = \frac{P_1 G^2 K_{\text{Total}}}{\rho_1 C_1^2} \quad (4-164)$$

$$P_2 = \left( P_1^2 - \frac{P_1 G^2 K_{\text{Total}}}{\rho_1 C_1^2} \right)^{0.5} \quad (4-165)$$

Therefore, the pressure drop

$$\Delta P \cong P_1 - P_2 \quad (4-166)$$

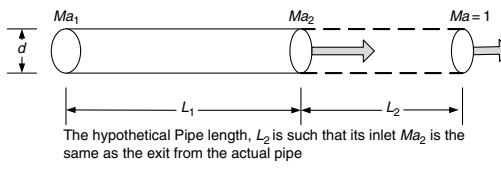
That is,

$$\Delta P \cong P_1 - \left( P_1^2 - \frac{P_1 G^2 K_{\text{Total}}}{\rho_1 C_1^2} \right)^{0.5} \quad (4-167)$$

Table 4-24 shows the conditions at the pipe exit as a function of the Mach number.

**TABLE 4-24 Conditions at the Pipe Exit as a Function of the Mach Number**

Isothermal Flow		Adiabatic Flow
$P = \rho \times \text{constant}$		$\frac{1}{2} \frac{G^2}{\rho^2} + \frac{k}{(k-1)} \frac{P}{\rho} = \text{constant}$
<b>Subsonic Flow</b>		
$P_2$	$P_1 \times Ma_1 / Ma_2$	$P_1 \frac{Ma_1}{Ma_2} \sqrt{\frac{2+(k-1)Ma_1^2}{2+(k-1)Ma_2^2}}$
$T_2$	$T_1$	$T_1 \frac{2+(k-1)Ma_1^2}{2+(k-1)Ma_2^2}$
$\rho_2$	$\rho_1 \times Ma_1 / Ma_2$	$\rho_1 \frac{Ma_1}{Ma_2} \sqrt{\frac{2+(k-1)Ma_2^2}{2+(k-1)Ma_1^2}}$
$v_2$	$v_1 \times Ma_2 / Ma_1$	$v_1 \times \rho_1 / \rho_2$
<b>Critical or Sonic Flow</b>		
$P_2$	$P_1 \times Ma_1 \times \sqrt{k}$	$P_1 Ma_1 \sqrt{\frac{2+(k-1)Ma_1^2}{k+1}}$
$T_2$	$T_1$	$T_1 \frac{2+(k-1)Ma_1^2}{(k-1)}$
$\rho_2$	$\rho_1 \times Ma_1 \times \sqrt{k}$	$\rho_1 Ma_1 \sqrt{\frac{(k+1)}{2+(k-1)Ma_1^2}}$
$v_2$	$v_s / \sqrt{k}$	$v_s$



#### 4.45 CRITICAL PRESSURE RATIO

The maximum attainable mass flux for given supply conditions is when  $\psi$  is a maximum, and is represented by

$$\psi = \left\{ \left( \frac{2}{\gamma-1} \right) \left[ \left( \frac{P}{P_1} \right)^{2/\gamma} - \left( \frac{P}{P_1} \right)^{(\gamma+1)/\gamma} \right] \right\}^{1/2} \quad (4-169)$$

Therefore, the pressure  $P_c$  causing the maximum flux can be found by differentiating  $\psi^2$  with respect to  $P$  and equating the result to zero, which gives

$$\left[ \frac{2}{\gamma} \left( \frac{P_c}{P_1} \right)^{2/\gamma} - \left( \frac{\gamma+1}{\gamma} \right) \left( \frac{P_c}{P_1} \right)^{(\gamma+1)/\gamma} \right] \frac{1}{P_c} = 0 \quad (4-170)$$

The critical pressure ratio  $P_c/P_1$  is given by

$$\frac{P_c}{P_1} = \left( \frac{2}{\gamma+1} \right)^{\frac{\gamma}{(\gamma-1)}} \quad (4-171)$$

For  $\gamma = 1.4$ ,  $P_c/P_1 = 0.528$ . For other values of  $\gamma$ , the value of the critical pressure ratio lies in the approximate range 0.5 to 0.6 where

$P_1$  = upstream pressure

$P_c$  = critical pressure

$\gamma$  = ratio of specific heat at constant pressure to specific heat at constant volume  
 $= (C_p/C_v)$ .

The maximum flux or maximum mass flow of the gas at sonic conditions is

$$\left( \frac{W}{A} \right)_{\max} = G_{\max} = P_o \left[ \left( \frac{kgM_w}{ZRT_o} \right) \left( \frac{2}{k+1} \right)^{(k+1)/k-1} \right]^{0.5} \quad (4-172)$$

where

$W$  = mass flow rate, lb/s

$A$  = cross-sectional flow area, ft<sup>2</sup>

$G$  = mass flux, lb/(ft<sup>2</sup> s)

$P_o$  = pressure at source condition, psia

$T_o$  = temperature at source condition, °R

$k$  = specific heat ratio constant.

Recently, Kumar [34] developed a method using thermodynamic principles to determine the status of flow (e.g., whether choking flow exists or not),  $(\Delta P/P_o)_{cr}$ , and the flow rate. His method removes the use of plots as generated in Crane Manual [4], which have few limitations. For an adiabatic compressible fluid flow, he showed that

$$\frac{2}{Ma_1^2(\gamma+1)} \left[ \left( \frac{(\gamma+1)Ma_1^2}{2+(\gamma-1)Ma_1^2} \right)^{\frac{(\gamma+1)}{2}} - 1 \right] + 2 \ln \sqrt{\frac{2+(\gamma-1)Ma_1^2}{(\gamma+1)Ma_1^2}} + K = 0 \quad (4-173)$$

and

$$r = \left( \frac{P_2}{P_o} \right)_{cr} = \frac{[0.5(\gamma+1)Ma_1^2]^{1/2}}{[1+0.5(\gamma-1)Ma_1^2]^{\frac{\gamma(\gamma+1)}{2(\gamma-1)}}} \quad (4-174)$$

The critical expansion factor is

$$Y_{cr} = \sqrt{\frac{K(1+r)}{2[K+2\ln(\frac{1}{r})]}} \quad (4-175)$$

and the mass flow rate at critical condition is

$$W = 0.1265 D^2 Y_{cr} \sqrt{\frac{P_o - P_2}{KV_o}} \quad (4-176)$$

where

$D$  = pipe internal diameter, mm

$L$  = pipe length, m

$P_A$  = ambient pressure, kPa

$P_o$  = stagnation upstream pressure, kPa

$P_1$  = pressure at inlet tip of the pipe, kPa

$P_2$  = pressure at outlet tip of the pipe, kPa

$Ma_1$  = Mach number at inlet tip of the pipe

$Ma_2$  = Mach number at outlet tip of the pipe

$\gamma$  = isentropic ratio of specific heat at constant pressure to specific heat at constant volume

$K$  = loss coefficient

$V_o$  = specific volume at upstream stagnation point, m<sup>3</sup>/kg

$r = (P_2/P_o)_{cr}$  = overall critical pressure ratio, dimensionless

$Y_{cr}$  = critical expansion factor, dimensionless

$W$  = mass flow rate, kg/h.

**EXAMPLE 4-9****Case Study**

The vapor ( $C_3$ ,  $C_4$  and  $C_5$ ) from the debutanizer unit C1007 operating at 17.6 bara in Figure 4-23 is cooled via an air cooler E-1031 to the accumulator V1008. The overhead gas line 8"-P10170-3101C-P is 84.7 m long, and the debutanizer boil-up rate is 17 kg/s (1738 m<sup>3</sup>/h). Calculate the pressure drop along the 8-in pipe to the air cooler condenser E-1031. The following are the other data obtained from the piping isometrics, piping data sheets, and fluid characteristics:

Operating temperature = 86 °C

Fluid density = 35.2 kg/m<sup>3</sup>

Ratio of specific heats  $\gamma = (C_p/C_v) = 1.11$

Kinematic viscosity = 0.2 cSt =  $0.2 \times 10^{-6}$  m/s<sup>2</sup>.

Fittings	Number
90° ell ( $r/R = 1.5$ )	5
Ball valve	2
Tee (straight thru)	3

*Solution*

$$\begin{aligned} \text{Dynamic viscosity} &= \text{density} \times \text{kinematic viscosity} \\ &= 35.2 \times 0.2 \times 10^{-6} \\ &= 0.00704 \times 10^{-3} \text{ kg/ms} \\ &= 0.00704 \text{ cP} \end{aligned}$$

The average molecular weight of the overhead vapor is

Composition	Molecular weight (kg/kmol)	Percentage in the vapor phase (%)	Average molecular weight
$C_3H_8$	44	18.00	7.92
$C_4H_{10}$	58	80.00	46.4
$C_5H_{12}$	72	2.00	1.44
Total		100.00	55.76

Assume compressibility factor  $Z = 0.958$

An 8-in. (203.2 mm) pipe size Sch. 40 (ID = 202.7 mm)

Friction factor  $f_T$  for 8-in. Sch. 40 CS material = 0.014 (Table 4-6)

Assuming that the flow condition of the vapor through the 8-in. pipe is isothermal.

Area of pipe

$$\begin{aligned} A &= \pi (ID)^2 / 4 \\ &= \pi (0.2027)^2 / 4 \\ &= 0.0323 \text{ m}^2 \end{aligned}$$

The velocity of gas in the line is

$$G = \rho v A$$

where

$A$  = pipe area, m<sup>2</sup>

$G$  = mass flow rate, kg/s

$v$  = fluid velocity, m/s

$\rho$  = fluid density, kg/m<sup>3</sup>.

or

$$\begin{aligned} v &= G / (\rho A) \\ &= \frac{17}{(35.2 \times 0.0323)} \\ &= 14.95 \text{ m/s} \end{aligned}$$

Sonic velocity of vapor in the line is

$$\begin{aligned} v_s &= \sqrt{Z \gamma R T} \text{ m/s} \\ &= \sqrt{(0.958) (1.11) \left( \frac{8314}{55.76} \right) (359.15)} \\ &\quad \times \sqrt{\left( \frac{J}{\text{kmol} - \text{K}} \cdot \frac{1}{\frac{\text{kg}}{\text{kmol}}} \cdot \text{K} \right)} \text{ m/s} \\ &= 238.63 \text{ m/s} \end{aligned} \quad (4-121)$$

The inlet Mach number  $Ma_1$  is

$$\begin{aligned} Ma_1 &= \frac{v}{v_s} = \frac{14.95}{238.63} \\ &= 0.0626 \end{aligned}$$

Therefore, the flow of gas through the pipe is subsonic, since the inlet Mach number is less than 1.

Gas Reynolds number is

$$\begin{aligned} Re &= 354 \frac{W}{d\mu} \\ &= \frac{(354) (17) (3600)}{(202.7) (0.00704)} \\ &= 15,181,975 \text{ (fully turbulent)} \\ &= 15.2 \times 10^6 \end{aligned} \quad (4-28)$$

Relative pipe roughness is

$$\begin{aligned} \frac{\epsilon}{D} &= \frac{0.000046}{202.7} = 0.000226 \\ \frac{1}{\sqrt{f_c}} &= -4 \log \left( \frac{\epsilon}{3.7D} - \frac{5.02}{Re} \log A \right) \end{aligned} \quad (4-35)$$

where

$$A = \frac{\epsilon/D}{3.7} + \left( \frac{6.7}{Re} \right)^{0.9}$$

$$A = \frac{0.00027}{3.7} + \left( \frac{6.7}{15,181,975} \right)^{0.9} = 6.32408 \times 10^{-5}$$

$$\begin{aligned} \frac{1}{\sqrt{f_c}} &= -4 \log \left[ \frac{0.000227}{3.7} - \frac{5.02}{15,181,975} \log (6.32408 \times 10^{-5}) \right] \\ &= 16.8098 \end{aligned}$$

$$f_c = 0.00354$$

(continued)

**EXAMPLE 4-9—(continued)**

The Darcy friction factor  $f_D = 4 f_C$

$$\begin{aligned} f_D &= 4(0.00354) \\ &= 0.0142 \end{aligned}$$

Using Darby's 3-K method

$$K_f = \frac{K_1}{Re} + K_i \left( 1 + K_d \left( \frac{25.4}{D_{n,mm}} \right)^{0.3} \right) \quad (4-74)$$

$Re = 15,181,975$  (turbulent)

Nominal pipe diameter of 8 in.,  $D_n = 203.2$  mm

Fittings	n	K <sub>1</sub>	nK <sub>1</sub>	K <sub>i</sub>	nK <sub>i</sub>	K <sub>d</sub>	K <sub>f</sub>
90° ell ( $r/D = 1.5$ )	5	800	4000	0.071	0.355	4.2	1.1543
Ball valve	2	300	600	0.017	0.034	4.0	0.1069
Tee (straight Thru)	3	800	2400	0.14	0.42	4.0	1.3204
Total							2.5816

Total loss coefficient  $K_{\text{Total}}$ :

$$K_{\text{Total}} = f_D \frac{L}{D} + \sum K_f$$

Pipe length,  $L = 84.7$  m

Diameter,  $d = 202.7$  mm.

$$\begin{aligned} K_{\text{Total}} &= 0.0142 \times \frac{84.7}{0.2027} + 2.5816 \\ &= 8.5152 \end{aligned}$$

Outlet pressure  $P_2$  is

$$P_2 = \left[ P_1^2 - \frac{P_1 G^2 K_{\text{Total}}}{\rho_1 C_1^2} \right]^{0.5} \quad (4-165)$$

where

$$C_1 = 2.484 \times 10^{-4} d^2 \quad (4-162)$$

$$\frac{G^2}{C_1^2} = \frac{17^2}{(2.484 \times 10^{-4} \times 202.7^2)^2} = 2.774 \quad (4-163)$$

$$\begin{aligned} P_2 &= \left[ P_1^2 - \frac{P_1 G^2 K_{\text{Total}}}{\rho_1 C_1^2} \right]^{0.5} \quad (4-165) \\ &= \left( 17.6^2 - \frac{(17.6)(2.774)(8.5152)}{35.2} \right)^{0.5} \\ &= 17.26 \text{ bara} \end{aligned}$$

Therefore, the pressure drop

$$\begin{aligned} \Delta P &\cong P_1 - P_2 \\ &= 0.335 \text{ bar} \end{aligned} \quad (4-166)$$

The process is assumed to be isothermal, therefore outlet temperature  $T_2$  is

$$\begin{aligned} T_2 &= T_1, \text{K} \\ T_2 &= 86^\circ \text{C} \\ &= (273.15 + 86) \text{ K} \\ &= 359.15 \text{ K} \end{aligned}$$

Density of the vapor at the exit is

$$\begin{aligned} \rho_2 &= \frac{10^5 P_2}{(R/M_w) T_2}, \text{kg/m}^3 \\ \rho_2 &= \frac{(10^5)(17.26)}{(8314/55.76)(359.15)} \\ &= 32.23 \text{ kg/m}^3 \end{aligned}$$

Flow velocity at pipe exit is

$$\begin{aligned} v &= G / (\rho A) \\ &= \frac{17}{(32.23 \times 0.0323)} \\ &= 16.33 \text{ m/s} \end{aligned}$$

$$\text{The exit Mach number } Ma_2 = 1/\sqrt{k} \quad (4-129b)$$

$$Ma_2 = 1/\sqrt{1.11} = 0.949$$

Rearranging Eq. (4-165) gives

$$\left[ P_1^2 - \frac{P_1 G^2 K_{\text{Total}}}{\rho_1 C_1^2} \right]^{0.5} - P_2 = 0 \quad (4-168)$$

Equation (4-168) involves a trial-and-error computation using a guess value for  $P_2$ . This is substituted into Eq. (4-168) until the left side gives a value of zero. The Excel spreadsheet with a *Solver* add-in tool is the most convenient computational tool for Eq. (4-168). Equation (4-168) is set to zero using a guess value of  $P_2$ . The procedure involves setting the quadratic Eq. (4-168) to zero; with a guess value of the outlet pressure and the *Solver* determine the outlet pressure after a set of iterations. The Excel spreadsheet (Example 4-9.xls) has been developed to determine the pressure drop of a compressible isothermal flow fluid using Eq. (4-168). Figure 4-36 shows the snapshots of the Excel spreadsheet calculation of Example 4-9.

Example 4-9. Pressure drop for compressible fluid flow using isothermal condition by A.K. Coker																		
3. The vapor (C <sub>3</sub> , C <sub>4</sub> and C <sub>5</sub> ) from a deutanizer C1007 is cooled via an air cooler condenser E1001.																		
4. To the accumulator vessel V1000. The overhead gas line is 64.7 m and the boil up rate is 17 kg/s.																		
5. The top of the unit operates at 17.6 bars. Calculate the pressure drop along the 6" pipe.																		
6. In the air cooler condenser. Data from piping anomalies, data sheets and fluid characteristics are:																		
9. Operating temperature, °C =	86 °C																	
10. Fluid density, kg/m <sup>3</sup> =	36.2 kg/m <sup>3</sup>																	
11. Ratio of specific heats, (Cp/Cv) =	1.11																	
12. Kinematic viscosity, cSt =	0.2 cSt																	
13. Compressibility factor, Z =	0.998																	
14. Pipe Length, m =	64.7 m																	
15. Operating pressure, P <sub>T</sub> =	17.6 bars																	
16. Vapor rate =	17 kg/s																	
17. Pipe internal diameter, 8-inch, Schedule 40 =	17.38 m <sup>3</sup> /h																	
18. Pipe nominal diameter, 8-inch, Schedule 40 =	203.2 mm																	
19. Pipe roughness, ε =	0.045 mm																	
20.																		
21.																		
22.																		
23.																		
24. Fittings Number																		
25. 90° Elbows = 5																		
26. Ball valve = 2																		
27. Tee (straight Thru) = 3																		
28.																		
29. Solution:																		
30.																		
31. Dynamic viscosity, μ = 0.00000704 kg/m·s																		
32. 0.00704 cP																		
33. 34. Average molecular weight of C <sub>3</sub> , C <sub>4</sub> and C <sub>5</sub> is:																		
35. Percent of vapor																		
36. Molecular weight of C <sub>3</sub> H <sub>8</sub> = 44 kg/mol 0.18																		
37. Molecular weight of C <sub>4</sub> H <sub>10</sub> = 58 kg/mol 0.80																		
38. Molecular weight of C <sub>5</sub> H <sub>12</sub> = 72 kg/mol 0.02																		
39.																		
40. Average molecular weight = 55.76 kg/mol																		
41. Solution:																		
42.																		
43. Absolute temperature, T = 369.15 K	Pipe Area 0.0323 m <sup>2</sup>																	
44.																		
45.																		
46. Mach Number = 14.96 m/s	Gas velocity 14.96 m/s																	
47.																		
48. Sonic velocity, vs = 290.63 m/s																		
49.																		
50. Inlet Mach number, v <sub>inlet</sub> = 0.063																		
51. Type of fluid flow flow is subsonic																		
52.																		
53. Reynold Number:																		
54.																		
55. Re = 15181975 TURBULENT																		
56.																		
57. Friction Factor, f																		
58.																		
59. Pipe roughness, ε = 0.045 mm																		
60.																		
61. ε/D = 0.00222																		
62. L/D = 5.19E+05																		
63.																		
* See equation. ** Units of p are lb/in <sup>2</sup> [Darby, Chem]																		
Type a question for help																		
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Figure 4-36 The Excel spreadsheet snapshot of Example 4-9.

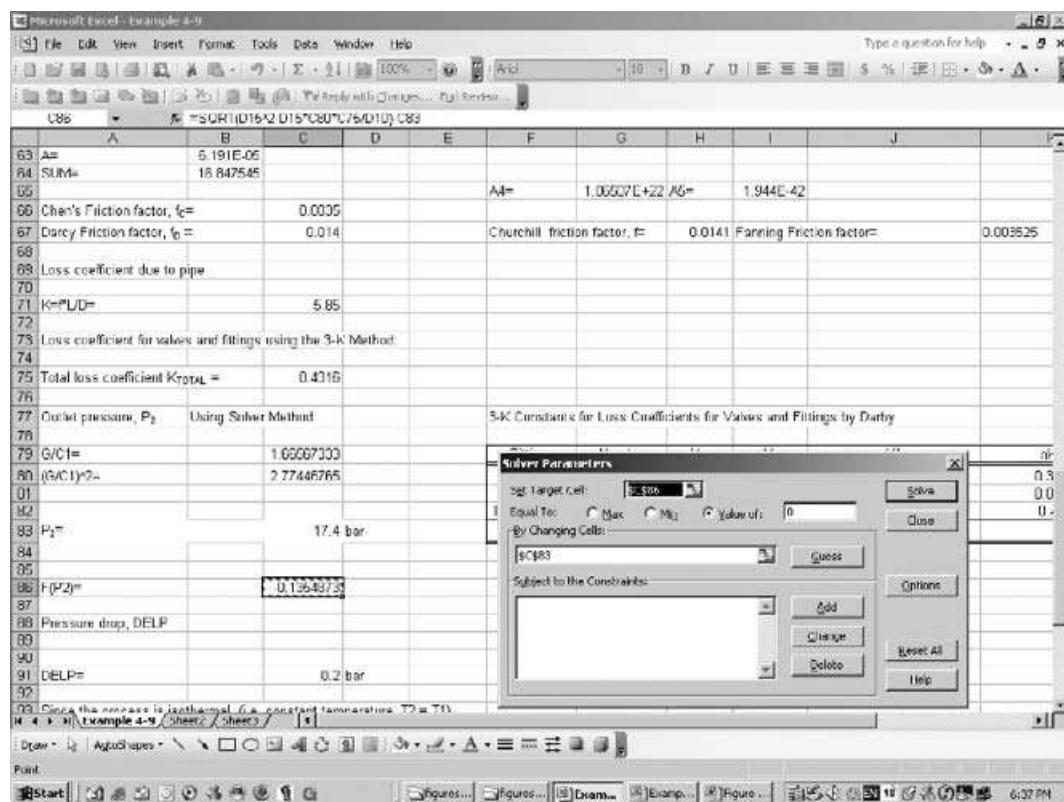
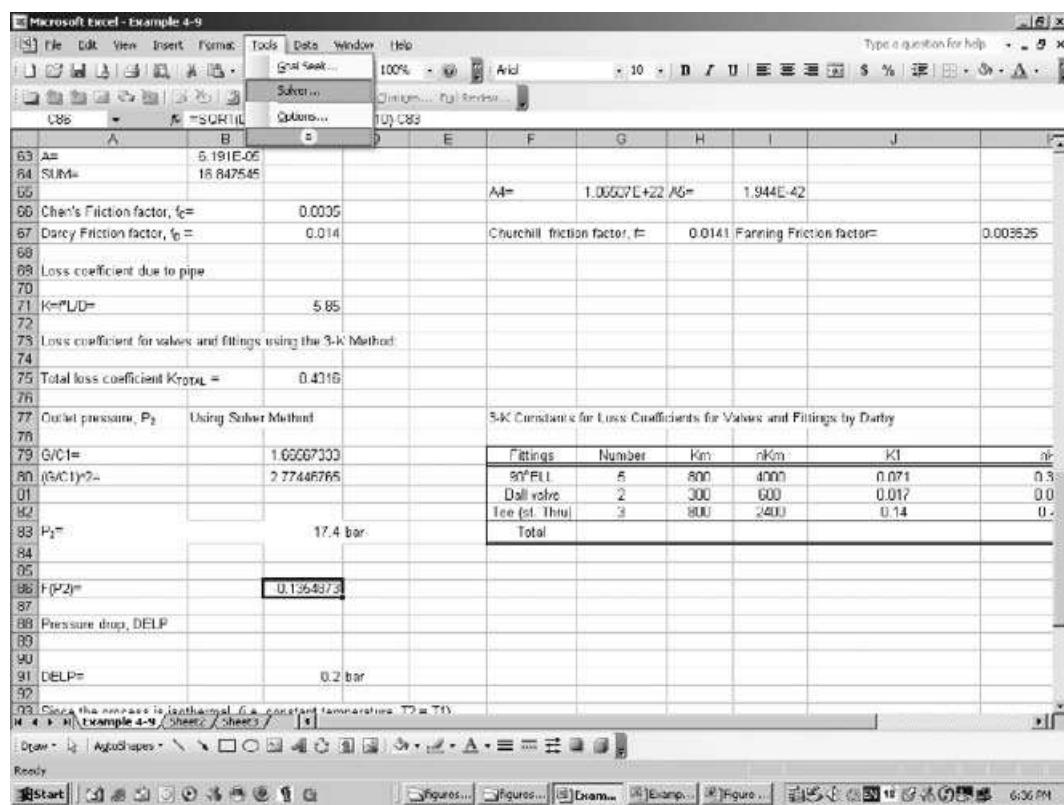
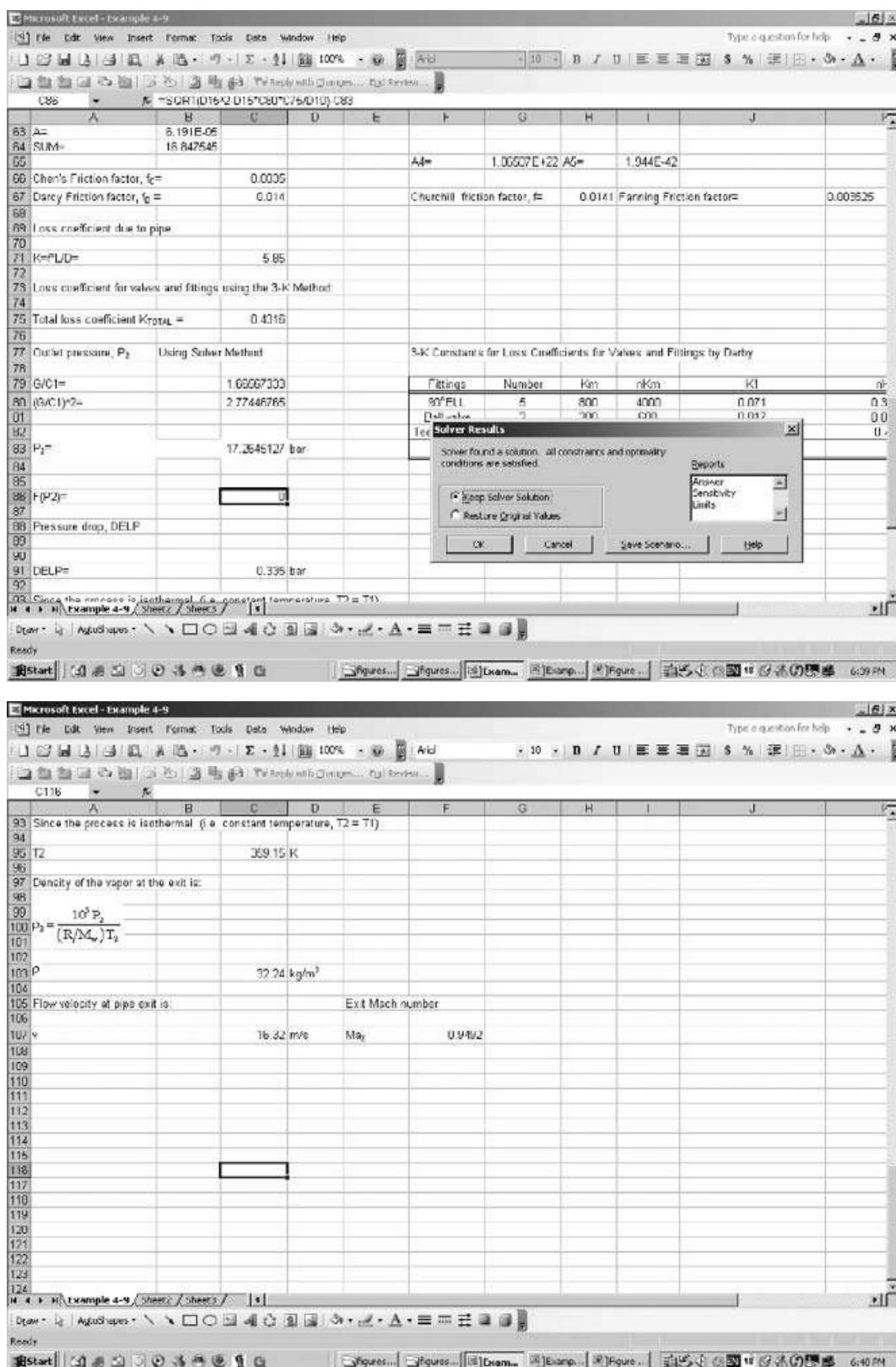


Figure 4-36—(continued)



**Figure 4-36—(continued)**

**EXAMPLE 4-10**

Propane vapor at 90°F and an upstream pressure  $P_1 = 20$  psig flows at a rate of 24,000 lb/h in an 800-ft long, 6-in. Sch. 40 horizontal carbon steel pipe. Under these conditions, the viscosity  $\mu_1 = 0.0094 \text{ cP}$  and the gas compressibility factor  $Z_1 = 0.958$ . Calculate the total pressure drop under isothermal flow conditions. Check for critical flow.

*Solution*

Since the pipe is long, assume an isothermal condition for the compressible vapor flow.

The density of propane at 90°F and pressure of 20 psig is

$$\rho = \frac{M_w P}{10.72 T} = \left( \frac{44 \times 34.7}{10.72 \times 550} \right) = 0.2589 \text{ lb/ft}^3$$

Velocity of the gas is

The 6-in. Sch. 40 pipe size (ID = 6.065 in.)

Area of pipe

$$\begin{aligned} A &= \pi (\text{ID})^2 / 4 \\ &= \pi (0.5054)^2 / 4 \\ &= 0.2006 \text{ ft}^2 \end{aligned}$$

The velocity of gas in the line is

$$G = \rho v A$$

where

$A$  = pipe area,  $\text{ft}^2$   
 $G$  = mass flow rate,  $\text{lb/s}$   
 $v$  = fluid velocity,  $\text{ft/s}$   
 $\rho$  = fluid density,  $\text{lb/ft}^3$ .

or

$$\begin{aligned} v &= G / (\rho A) \\ &= \frac{24,000}{(0.2589 \times 0.2006 \times 3600)} \\ &= 128.36 \text{ ft/s} \end{aligned}$$

Sonic velocity is

$$\begin{aligned} v_s &= \sqrt{Z k g 144 \frac{P}{\rho}}, \text{ ft/s} \\ &= \sqrt{\frac{0.958 \times 1.15 \times 32.174 \times 144 \times 34.7}{0.2589}} \\ &= 827.11 \text{ ft/s} \end{aligned} \quad (4-118)$$

Sonic flow would occur at the end of the pipe and not where the pressure is 20 psig.

The inlet Mach number  $Ma_1$  is

$$\begin{aligned} Ma_1 &= \frac{v}{v_s} \\ &= \frac{128.36}{827.11} = 0.1552 \end{aligned} \quad (4-129)$$

Since the Mach number is less than 1, the flow of propane through the pipe is subsonic.

Reynolds number  $Re$ :

$$Re = \frac{D v \rho}{\mu_e} = 123.9 \frac{d v \rho}{\mu} = 6.31 \frac{W}{d \mu} \quad (4-26)$$

$$\begin{aligned} Re &= 6.31 \frac{24,000}{6.065 \times 0.0094} \\ &= 2,656,329 \\ &= 2.66 \times 10^6 \end{aligned}$$

Friction factor,  $f$ :

$$\frac{1}{\sqrt{f_c}} = -4 \log \left\{ \frac{\varepsilon}{3.7 D} - \frac{5.02}{Re} \log A \right\} \quad (4-35)$$

where

$$A = \frac{\varepsilon/D}{3.7} + \left( \frac{6.7}{Re} \right)^{0.9}$$

$\varepsilon$  = pipe roughness, ft

D = pipe internal diameter, ft.

Substituting in Eq. (4-35)

$$\begin{aligned} A &= \frac{0.0002968}{3.7} + \left( \frac{6.7}{2,656,329} \right)^{0.9} \\ &= 8.9370 \times 10^{-5} \end{aligned}$$

$$\begin{aligned} \frac{1}{\sqrt{f_c}} &= -4 \log \left[ \frac{0.0002968}{3.7} - \frac{5.02}{26,56,329} \log (8.9370 \times 10^{-5}) \right] \\ &= 16.22468 \end{aligned}$$

$$f_c = 0.00379$$

The Darcy friction factor  $f_D = 4 f_c$

$$\begin{aligned} f_D &= 4 (0.00379) \\ &= 0.0152 \end{aligned}$$

Loss coefficient due to straight pipe:

$$\begin{aligned} K &= f_D \frac{L}{D} \\ &= \frac{0.0152 \times 800 \times 12}{6.065} \\ &= 24.06 \end{aligned}$$

(continued)

**EXAMPLE 4-10—(continued)**

Outlet pressure,  $P_2$ , is

$$P_2 = \left[ P_1^2 - \frac{P_1 G^2 K_{\text{Total}}}{\rho_1 C^2} \right]^{0.5} \quad (4-159)$$

where

$$C = 1335.6 \text{ d}^2 \quad (4-156)$$

$$\frac{G^2}{C^2} = \frac{24,000^2}{(1335.6 \times 6.065)^2} = 0.2386 \quad (4-157)$$

$$P_2 = \left( 34.7^2 - \frac{(34.7)(0.2386)(24.06)}{0.2589} \right)^{0.5}$$

$$= 20.85 \text{ psia}$$

$$\Delta P \cong P_1 - P_2$$

$$= 34.7 - 20.85$$

$$= 13.85 \text{ psi}$$

The critical pressure,  $P_c$ , is

$$P_c = \frac{G}{11,400d^2} \left( \frac{RT}{k[k+1]} \right)^{1/2}, \text{ psia} \quad (4-126)$$

$$P_c = \frac{24,000}{11,400(6.065)^2} \sqrt{\frac{34.085 \times 500}{1.15 \times 2.15}}$$

$$= 4.998 \text{ psia}$$

The outlet temperature,  $T_2$ , is the same as the inlet temperature (e.g., isothermal condition)

$$T_2 = T_1, \text{ }^\circ \text{R}$$

$$T_2 = 550, \text{ }^\circ \text{R}$$

Vapor density at the exit is

$$\rho_2 = \frac{M_w P_2}{10.72 T_2}, \text{ lb/ft}^3$$

$$\rho_2 = \frac{(44)(20.85)}{(10.72)(550)} = 0.1556 \text{ lb/ft}^3$$

Flow velocity at pipe exit is

$$v = G / (\rho A)$$

$$= \frac{24,000}{(0.1556 \times 0.2006 \times 3600)}$$

$$= 213.59 \text{ ft/s}$$

$$\text{The exit Mach number } Ma_2 = 1/\sqrt{k}$$

$$(4-129b)$$

$$Ma_2 = \frac{1}{\sqrt{1.15}} = 0.9325$$

**TABLE 4-25 Limiting Critical Values**

$K$	$\gamma = 1.2$		$\gamma = 1.3$		$\gamma = 1.4$		$\gamma = 1.5$		$\gamma = 1.6$	
	$(\Delta P/P_o)_{\text{cr}}$	$\gamma_{\text{cr}}$								
1	0.62	0.52	0.64	0.51	0.66	0.50	0.68	0.50	0.70	0.49
2	0.64	0.54	0.67	0.53	0.69	0.53	0.71	0.52	0.73	0.51
3	0.68	0.58	0.70	0.56	0.73	0.55	0.75	0.54	0.78	0.53
4	0.71	0.60	0.74	0.58	0.76	0.57	0.78	0.55	0.80	0.54
5	0.74	0.61	0.76	0.59	0.78	0.58	0.81	0.56	0.82	0.54
6	0.75	0.62	0.78	0.60	0.80	0.58	0.82	0.56	0.84	0.55
7	0.77	0.62	0.79	0.60	0.81	0.58	0.84	0.56	0.85	0.54
8	0.78	0.63	0.80	0.61	0.83	0.59	0.85	0.56	0.86	0.54
9	0.79	0.63	0.81	0.61	0.84	0.59	0.85	0.56	0.88	0.54
10	0.80	0.63	0.82	0.61	0.85	0.59	0.86	0.56	0.88	0.54
20	0.86	0.64	0.88	0.61	0.89	0.58	0.91	0.55	0.93	0.52
30	0.88	0.64	0.90	0.60	0.92	0.56	0.93	0.53	0.94	0.50
40	0.90	0.63	0.92	0.59	0.94	0.55	0.94	0.52	0.96	0.48
50	0.92	0.63	0.93	0.59	0.94	0.55	0.95	0.52	0.96	0.47
60	0.92	0.62	0.94	0.58	0.95	0.54	0.95	0.51	0.97	0.46
70	0.92	0.62	0.94	0.58	0.95	0.54	0.96	0.50	0.97	0.46
80	0.93	0.62	0.94	0.57	0.96	0.53	0.96	0.49	0.97	0.45
90	0.93	0.62	0.94	0.57	0.96	0.53	0.96	0.49	0.97	0.45
100	0.94	0.61	0.95	0.57	0.96	0.52	0.97	0.48	0.98	0.44

(Source: S. Kumar, *Chem. Eng.*, Oct 2002, p. 62.)

The simultaneous solution of Eqs (4-173) and (4-174) eliminates  $Ma$  and yields a value for  $r$ , the critical pressure ratio. Table 4-25 shows a wide variation in the critical values with respect to  $\gamma$  (i.e., ratio of specific heats,  $C_p/C_v$ ) and the loss coefficient  $K$ .

#### 4.46 ADIABATIC FLOW

If there is no heat transfer or energy dissipated in the gas when traversing from state 1 to state 2, the process is adiabatic and reversible, that is isentropic. However, the actual flow conditions are somewhere between isothermal and adiabatic and, as such, the flow behavior can be described by the isentropic equations, where the isentropic constant  $k$  replaced a polytropic constant  $\gamma$  (e.g.,  $1 < \gamma < k$ ). For isothermal condition,  $\gamma = 1$ , whereas truly isentropic flow corresponds to  $\gamma = k$ .

The density and temperature as a function of pressure are

$$\rho = \rho_1 \left( \frac{P}{P_1} \right)^{1/k}, \quad T = T_1 \left( \frac{P}{P_1} \right)^{(k-1)/k} \quad (4-177)$$

The mass flow rate,  $G$ , by using Eq. (4-177) to eliminate  $\rho$  and  $T$  and solving for  $G$  gives

$$G = \sqrt{P_1 \rho_1} \left[ \frac{2 \left( \frac{k}{k+1} \right) \left( 1 - \left( \frac{P_2}{P_1} \right)^{(k-1)/k} \right)}{\frac{4 f L}{D} - \frac{2}{k} \ln \left( \frac{P_2}{P_1} \right)} \right]^{1/2} \quad (4-178)$$

where  $f$  = Fanning friction factor.

If the system contains fittings as well as straight pipe, the term  $4 f L / D$  ( $= K_f$ , pipe) can be replaced by  $\sum K_f$ , that is the sum of all loss coefficients in the system.

#### EXAMPLE 4-11

From the table listed below determine the status of flow (e.g., whether choking flow exists or not),  $(\Delta P/P_o)_{cr}$ , and the flow rate [34].

Data	Value
Upstream pressure, $P_o$ (kPa)	6600
Downstream discharge pressure, $P_A$ (kPa)	200
Upstream specific volume, $V_o$ (m <sup>3</sup> /kg)	0.01724
Isentropic coefficient, $\gamma$	1.55
Internal pipe diameter, $D$ (mm)	52.5
Length of pipe, $m$	100
Number of elbows	4
Loss coefficient, $K$	45

#### Solution

For known isentropic coefficient  $\gamma$  and loss coefficient  $K$ , a guessed value of  $Ma_1$  is used in Eq. (4-173) until the left side of the equation approximates to a value of zero. Otherwise, a new guess value of  $Ma_1$  is used in Eq. (4-173). Once the required value is known, Eqs. (4-174), (4-175), and (4-176) are subsequently used to determine  $r$ ,  $P_2$ ,  $\Delta P$ ,  $Y_{cr}$ , and  $W$  respectively. This procedure involves the use of the Excel spreadsheet with the *Goal Seek* or *Solver* add-in from the Tools menu and is given in Example 4-11.xls.

#### 4.47 THE EXPANSION FACTOR, Y

The adiabatic flow Eq. (4-178) can be represented in a form:

$$G = Y \left( \frac{2 \rho_1 \Delta P}{\sum K_f} \right)^{1/2} = Y \sqrt{P_1 \rho_1} \left[ \frac{2 (1 - P_2/P_1)}{\sum K_f} \right]^{1/2} \quad (4-179)$$

where

$\rho_1 = P_1 M_w / RT_1$ ,  $\Delta P = P_1 - P_2$  and  $Y$  is the expansion factor. Note that Eq. (1.179) without the  $Y$  term is the Bernoulli equation for an incompressible fluid of density  $\rho_1$ . Therefore, the expansion factor  $Y = G_{\text{adiabatic}} / G_{\text{incompressible}}$  is the ratio of the adiabatic mass flux (Eq. (4-179)) to the corresponding incompressible mass flux, and is a unique function of  $P_2/P_1$ ,  $k$  and  $K_f$ . Figure 4-38a shows values of  $Y$  for  $k = 1.3$  and  $k = 1.4$  as a function of  $\Delta P/P_1$  and  $\sum K_f$  (which is denoted by simply  $K$  on these plots). Figures 4-38b and c show the expansion factor  $Y$  for compressible flow through nozzles and orifices, and plots of the critical pressure ratio for compressible flow through nozzles and venturi tubes respectively. The conditions corresponding to the lower ends of the lines on the plots (e.g., the *nought*) represent the sonic (choked flow) state where  $P_2 = P_2^*$ . These same conditions are shown in the tables accompanying the plots, thus allowing the relationships for choked flow to be determined more accurately than is possible from reading the plots.

*Note:* It is not possible to extrapolate beyond the *nought* at the end of the lines in Figures 4-38a and b as this represents the choked flow state, in which  $P_2 = P_2^*$  (inside the pipe), and is independent of the external exit pressure. Figures 4-38a

Using the Excel spreadsheet Example 4-11.xls, the calculated overall critical pressure ratio  $r$  is  $r = 0.04804$

The critical pressure  $P_2$  is

$$\begin{aligned} P_2 &= r \times P_o \\ &= 0.04804 \times 6600 \\ &= 317.06 \text{ kPa} \end{aligned}$$

#### Test for choke flow

Since  $P_2 > P_A$ , the pipe will choke.

The critical expansion factor  $Y_{cr}$  from Eq. (4-175) is

$$\begin{aligned} Y_{cr} &= \sqrt{\frac{45 (1 + 0.04804)}{2 [45 + 2 (1/0.04804)]}} \\ &= 0.6795 \end{aligned}$$

The critical mass flow rate from Eq. (4-176) is

$$\begin{aligned} W &= 0.1265 (52.5)^2 (0.6795) \sqrt{\frac{6600 - 317.06}{(45 \times 0.01724)}} \\ &= 21,320.96 \text{ kg/h} \end{aligned}$$

Figure 4-37 gives the Excel spreadsheet snapshots of Example 4-11.

Example 4-11. Critical Flow of compressible fluid calculations by A.K. Coker											
This is based upon Kumar's [68] method (Chem. Eng. Oct. 2002, p62) using thermo-dynamic principles to determine the status of flow (i.e. whether choking flow exists or not). This method removes the use of plots as generated in Crane Manual [3]. For an adiabatic compressible fluid flow:											
$\frac{2}{Ma_1^2(\gamma+1)} \left[ \left( \frac{(\gamma+1)Ma_1^2}{2+(\gamma-1)Ma_1^2} \right)^{\frac{(\gamma+1)}{2}} - 1 \right] + 2\ln \sqrt{\frac{2+(\gamma-1)Ma_1^2}{(\gamma+1)Ma_1^2}} + K = 0 \quad (4-173)$											
$\tau = \left( \frac{P_2}{P_{2c}} \right)_\sigma = \frac{[0.5(\gamma+1)Ma_1^2]^{\gamma^2}}{[1+0.5(\gamma-1)Ma_1^2]^{\frac{\gamma+1}{2}}} \quad (4-174)$											
$Y_\sigma = \sqrt{\frac{K(1+\tau)}{2(K+2\ln[1/\tau])}} \quad (4-175)$											
$W = 0.125D^2 Y_\sigma \sqrt{\frac{P_2 - P_1}{KV_u}} \quad (4-176)$											
Notations											
D = Internal pipe diameter, mm.											
K = Resistance or loss coefficient											
L = Length of pipe, m											
P <sub>A</sub> = Ambient pressure, kPa abs											
P <sub>0</sub> = Stagnation upstream pressure, kPa abs											
P <sub>1</sub> = Pressure at inlet tip of the pipe, kPa abs											
P <sub>2</sub> = Pressure at outlet tip of the pipe, kPa abs											
Ma <sub>1</sub> = Mach number at inlet tip of the pipe											
Ma <sub>2</sub> = Mach number at outlet tip of the pipe											
Y = Ratio of specific heat at constant pressure to specific heat at constant volume at upstream conditions, dimensionless											
r = overall critical pressure ratio, dimensionless ( $P_2/P_{2c}$ ) <sub>cr</sub>											
V <sub>0</sub> = Upstream specific volume, m <sup>3</sup> /kg											
W = Mass flow rate, kg/h											
D = 50.5 mm											
P <sub>A</sub> = 200 kPaabs											
P <sub>1</sub> = 6600 kPaabs											
K = 45											
Y = 1.55											
Ma <sub>1</sub> = 0.105											
V <sub>0</sub> = 0.01724 m <sup>3</sup> /kg											
L = 100 m											
Calculations											
Using Goal Seek or Solver (from the Tools menu) by setting Eq. 4-173 = 0 (i.e. cell C53 = 0).											

Figure 4-37 The Excel spreadsheet snapshot of Example 4-11.

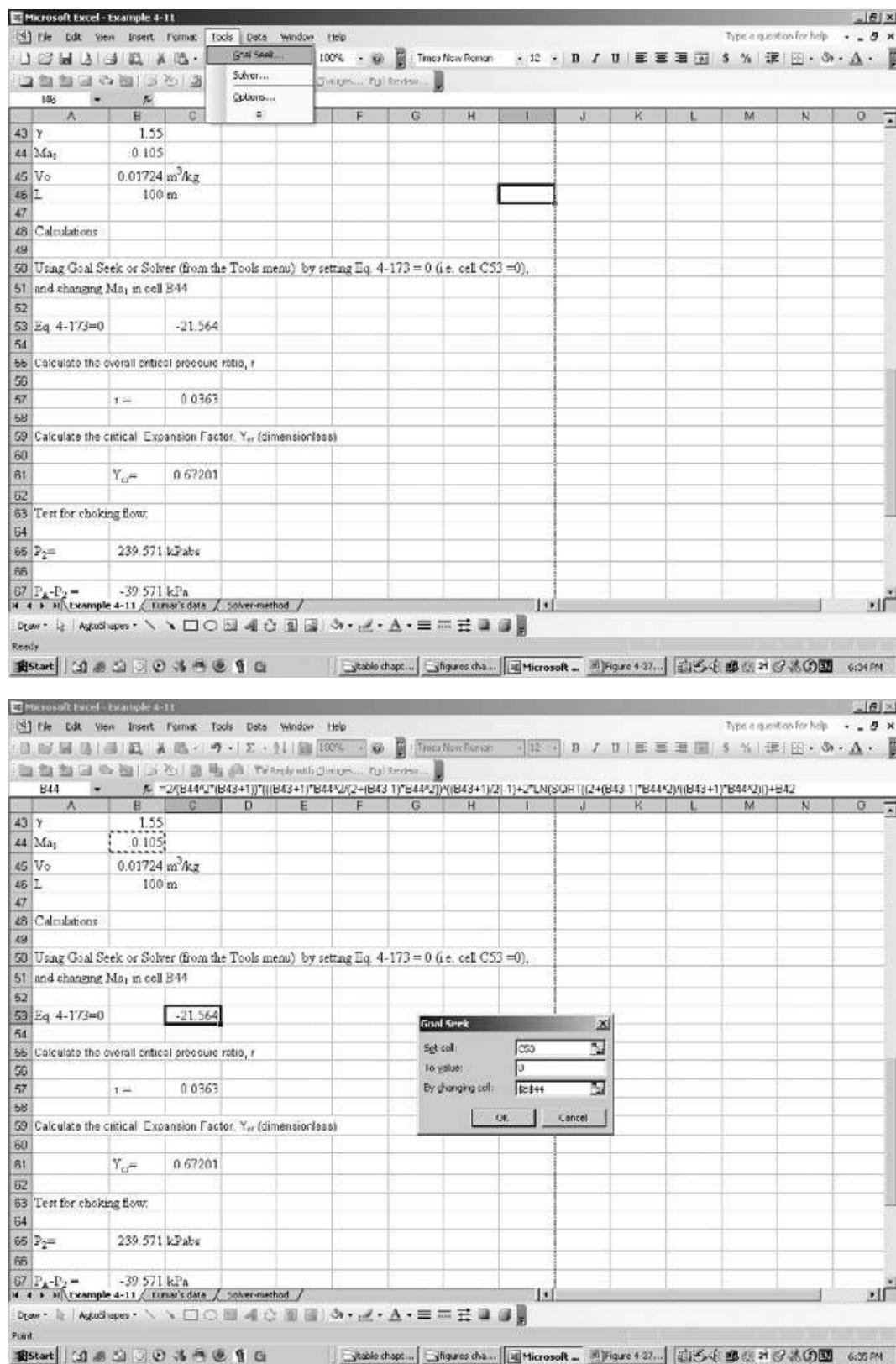
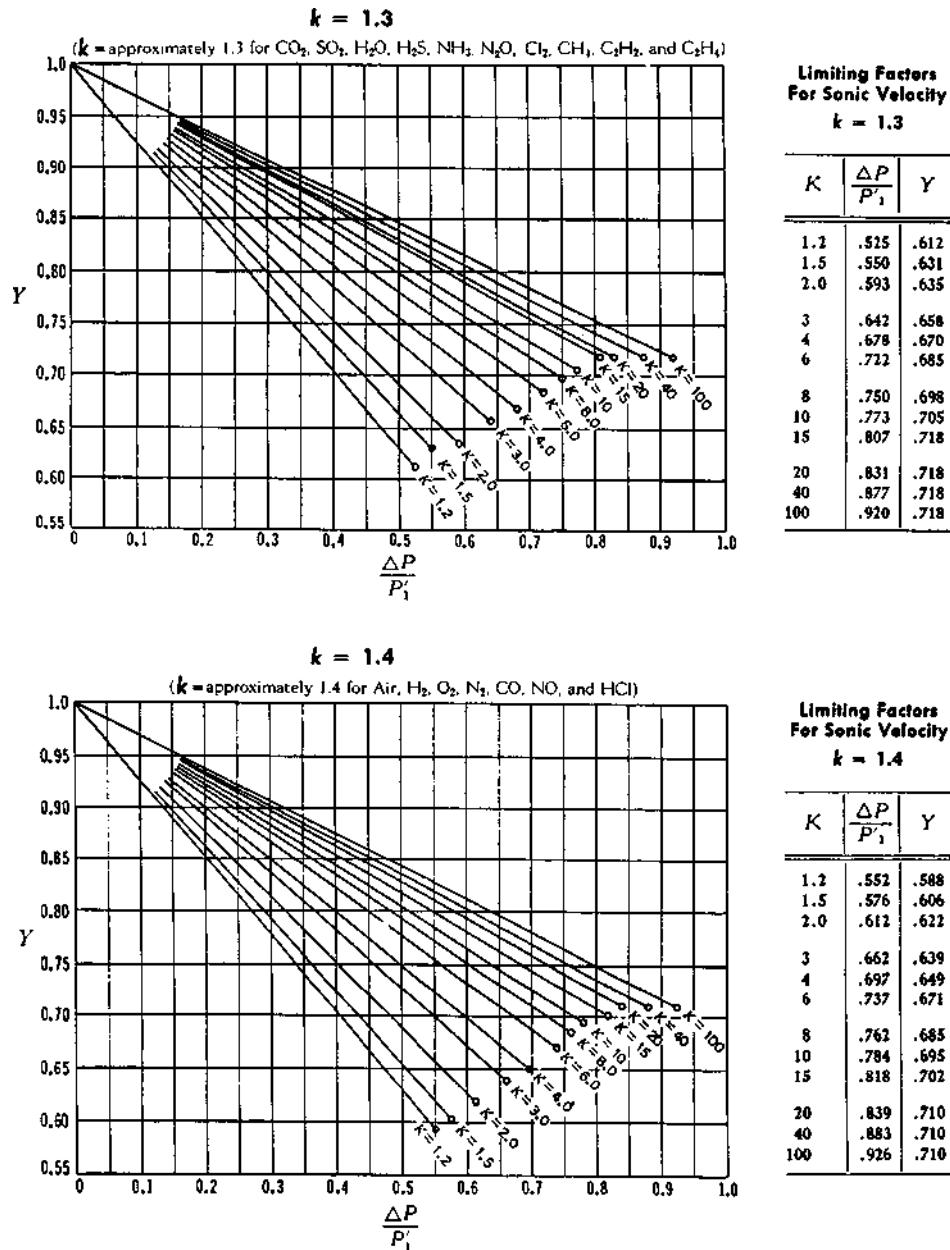


Figure 4-37—(continued)



Figure 4-37—(continued)



**Figure 4-38a** Net expansion factor,  $Y$ , for compressible flow through pipe to a larger flow area. (Reprinted/adapted with permission from "Flow of Fluids Through Valves, Fittings, and Pipe", Technical Paper No. 410, 1999, Crane Co. All rights reserved).

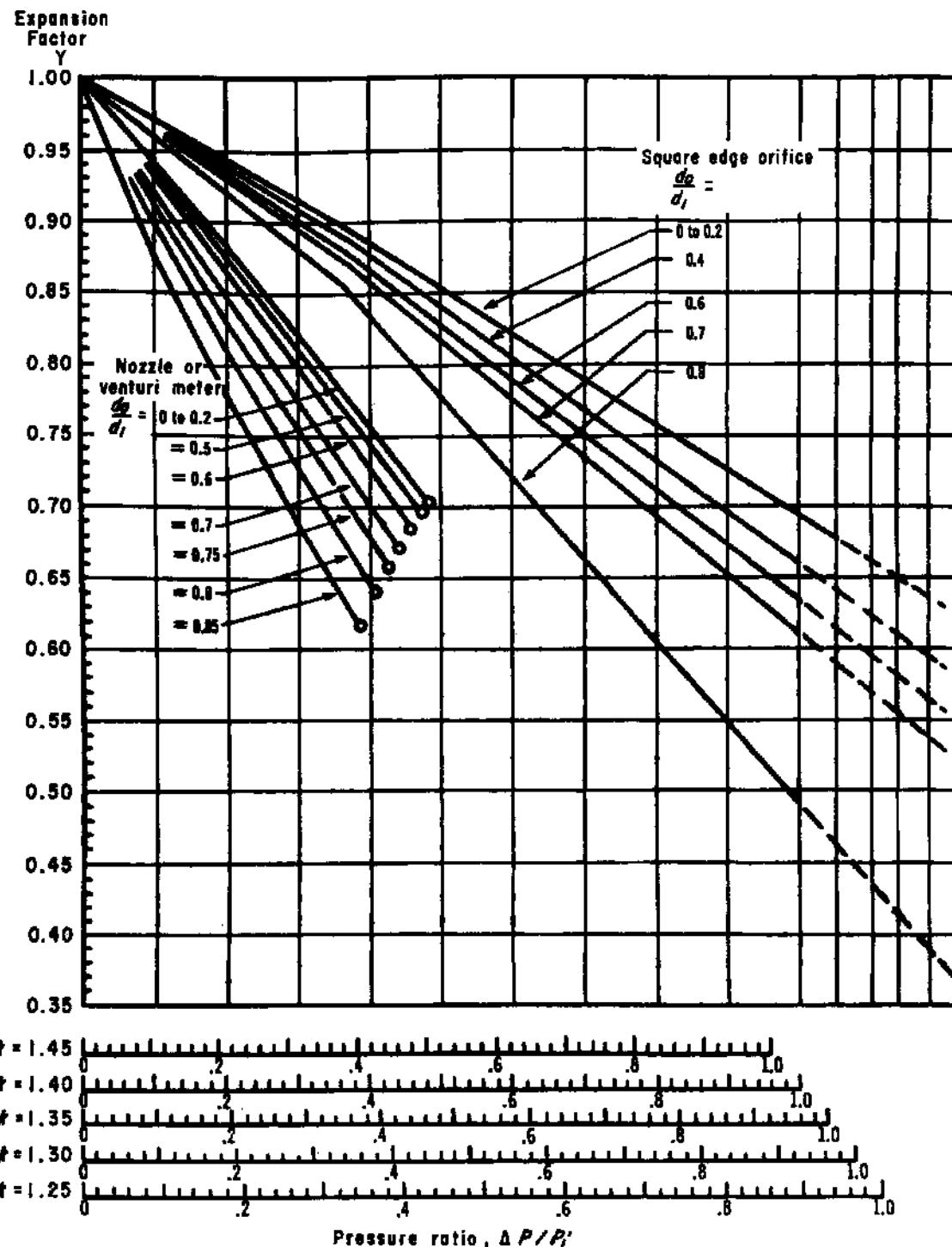
and b provide an alternative method of solving compressible adiabatic flow problems for piping systems. However, this procedure requires some iterations because the value of  $K_f$  depends on the Reynolds number which cannot be determined until  $G$  is obtained.

#### 4.48 MISLEADING RULES OF THUMB FOR COMPRESSIBLE FLUID FLOW

In general, compressible fluid flow calculations are much more complicated than incompressible fluid flow. Recently, Walters [35, 36] has shown that rules of thumb that are applied in the design

calculations for compressible fluid flow can be grossly misleading and erroneous. The common rules are

1. That adiabatic and isothermal flow bracket all flowrates. These conditions do not occur.
2. If the pipe pressure drop in a compressible flow system is less than 40% of the inlet pressure, then the incompressible flow calculation methods can be safely employed, with the average density along the pipe used in the equations. He further showed that this rule is invalid unless associated with a particular  $f L/D$  ratio.

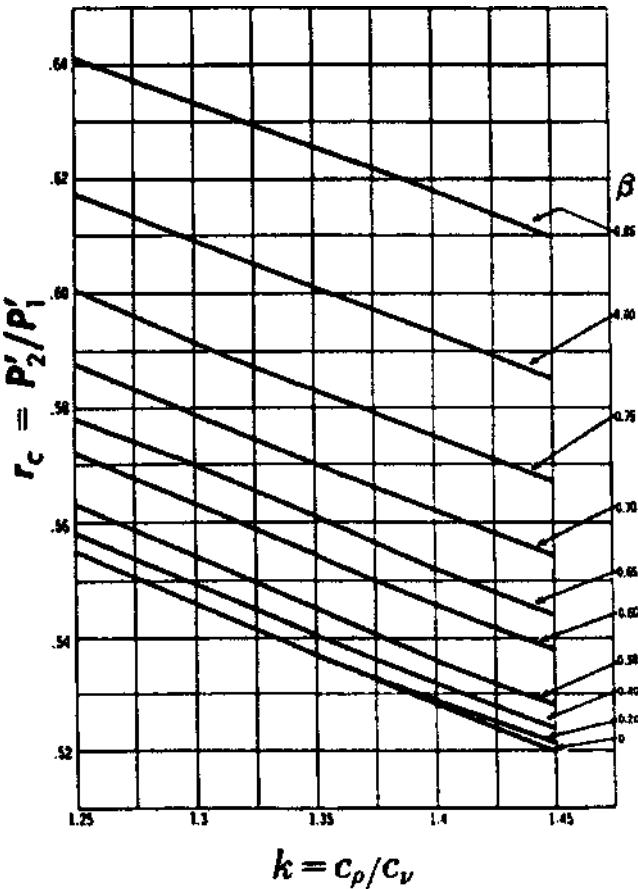


**Figure 4-38b** Net expansion factor,  $Y$ , for compressible flow through nozzles and orifices. (Reprinted/Adapted with permission from "Flow of Fluids Through Valves, Fittings, and Pipe", Technical Paper No. 410, 1999, Crane Co. All rights reserved).

3. Choked air flow at 50% pressure drop. An equation often used to determine the likelihood of sonic choking is

$$\frac{P^*}{P_o} = \left( \frac{2}{\gamma + 1} \right)^{\gamma/(\gamma-1)} \quad (4-180)$$

where  $P^*$  is the critical static pressure at sonic velocity and  $P_o$  is the local stagnation pressure at the orifice/valve. Walters indicated that using Eq. (4-180) with  $\gamma = 1.4$  results in 47% pressure drop to obtain choking. Furthermore, he stated that Eq. (4-180) cannot be used with the supply pressure if there is any significant pressure drop from the supply to the orifice/valve.



**Figure 4-38c** Critical Pressure Ratio,  $r_c$ , for compressible flow through nozzles and venturi tubes. (Reprinted/Adapted with permission from "Flow of Fluids Through Valves, Fittings, and Pipe", Technical Paper No. 410, 1999, Crane Co. All rights reserved. Note:  $P'$  = psia,  $\beta$  = ratio of small-to-large diameter in orifices and nozzles, and contractions or enlargements in pipes).

For gases with different specific heat ratios, the pressure drop ratio will differ somewhat, in accordance with Eq. (4-180). In addition, Eq. (4-180) breaks down for pipe-system analysis when pipe friction becomes a factor. This is because the stagnation pressure in the equation is the pressure at the upstream side of the shock wave. However, if there is any pressure drop in the pipe from the supply pressure to the shock wave, then the supply pressure cannot be used in Eq. (4-180). Instead, the local stagnation pressure at the shock wave must be used, which is unknown unless the pressure drop is determined from alternative means. Therefore, Eq. (4-180) cannot be used to predict the supply and discharge pressures necessary for sonic choking unless the piping has negligible friction loss.

#### 4.49 OTHER SIMPLIFIED COMPRESSIBLE FLOW METHODS

As shown earlier, most gases are not isothermal and, therefore, it is impossible to know how much error is introduced by the assumption of constant temperature.

Simplified equations typically do not address sonic-choking conditions, and cannot address the delivery temperature. These equations break down at high Mach numbers.

The entire pipe is solved in one lumped calculation instead of coupling the governing equations in marching order. It is difficult to extend the equations to pipe networks.

Walters developed compressible flow equations for single pipe [35]:

Adiabatic flow equation and integrated solution are

$$\int_0^L \frac{f}{D} dx = \int_{Ma_1}^{Ma_2} \frac{1 - Ma^2}{\gamma Ma^4 [1 + \frac{\gamma-1}{2} Ma^2]} dMa^2 \quad (4-181)$$

$$\begin{aligned} \frac{fL}{D} = & \frac{1}{\gamma} \left( \frac{1}{Ma_1^2} - \frac{1}{Ma_2^2} \right) \\ & + \frac{\gamma+1}{2\gamma} \ln \left( \frac{Ma_1^2}{Ma_2^2} \right) \left( \frac{1 + \frac{\gamma-1}{2} Ma_2^2}{1 + \frac{\gamma-1}{2} Ma_1^2} \right) \end{aligned} \quad (4-182)$$

Isothermal flow equation and integrated solution are

$$\int_0^{L_T} \frac{f}{D} dx = \int_{Ma_1}^{Ma_2} \frac{(1 - \gamma Ma^2)}{\gamma Ma^4} dMa^2 \quad (4-183)$$

$$\frac{f L_T}{D} = \frac{1 - \left( \frac{Ma_1^2}{Ma_2^2} \right)}{\gamma Ma_1^2} - \ln \left( \frac{Ma_2^2}{Ma_1^2} \right) \quad (4-184)$$

Computer software has been developed that models pipe systems of compressible fluids and this can be obtained from the website [www.aft.com](http://www.aft.com).

#### 4.50 FRICTION DROP FOR FLOW OF VAPORS, GASES, AND STEAM

See Figures 4-39a and 4-39b

##### A. DARCY RATIONAL RELATION FOR COMPRESSIBLE FLOW FOR ISOTHERMAL PROCESS [4]

$$\frac{\Delta P}{100 \text{ ft}} = \frac{0.000336 f W^2 \bar{V}}{d^5} = \frac{0.000336 f W^2}{d^5 \rho} \quad (4-65)$$

or

$$\frac{\Delta P}{100 \text{ ft}} = \frac{0.000001959 f (q'_h)^2 S_g^2}{d^5 \rho} \quad (4-185)$$

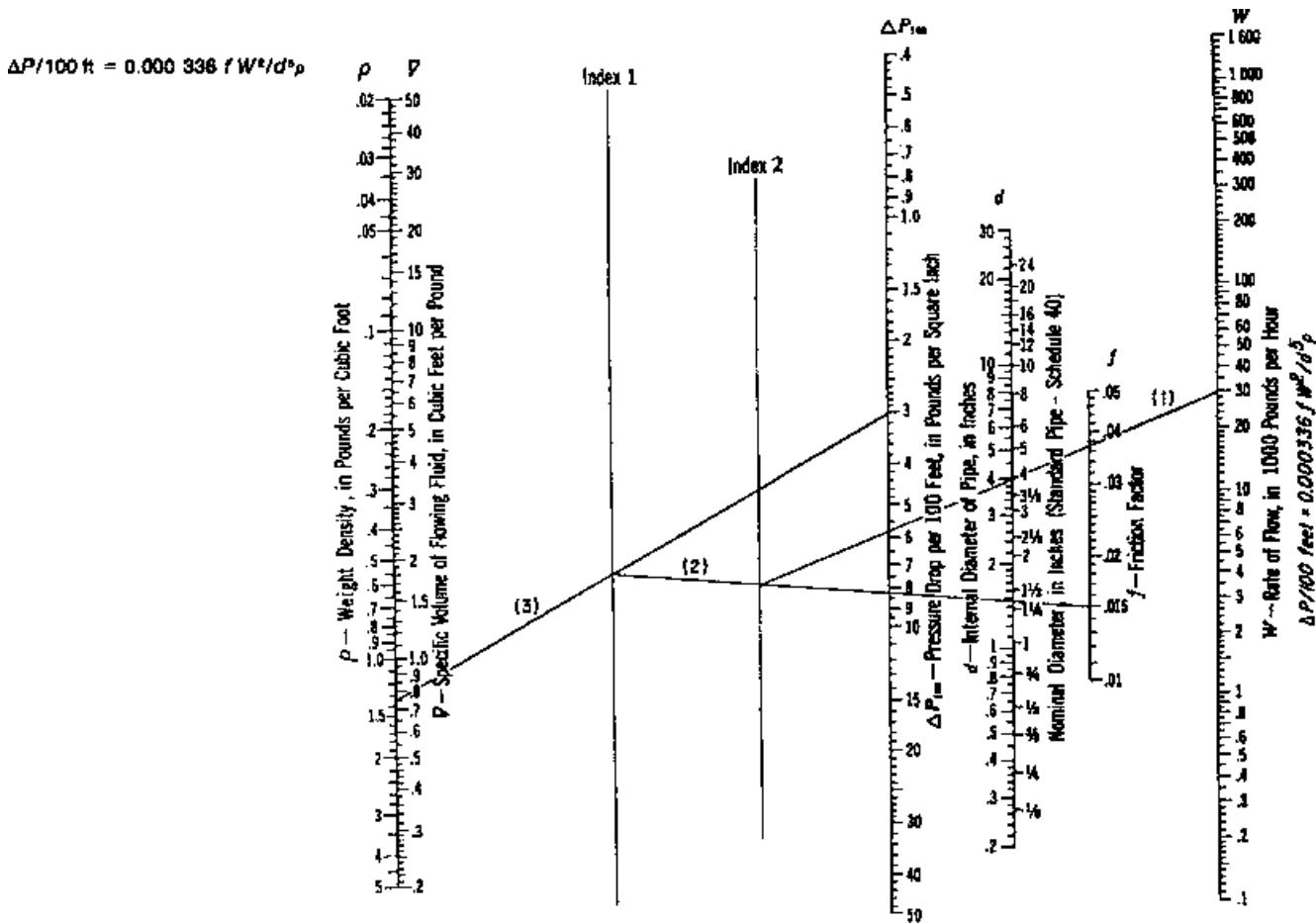
In SI units,

$$\Delta P / 100 \text{ m} = \frac{62,530 f W^2 \bar{V}}{d^5} = \frac{62,530 f W^2}{d^5 \rho} \quad (4-66)$$

or

$$\Delta P / 100 \text{ m} = \frac{93,650 f (q'_h)^2 S_g}{d^5 \rho} \quad (4-186)$$

The general procedures outlined previously for handling fluids involving the friction factor,  $f$ , and the  $Re$  chart are used with the above relations. This is applicable to compressible flow systems under the following conditions [4].



**Figure 4-39a** Pressure drop in compressible flow lines. (Reprinted/Adapted with permission from "Flow of Fluids Through Valves, Fittings, and Pipe", Technical Paper No. 410, 1999, Crane Co. All rights reserved.)

where

$W$  = rate of flow, lb/h (kg/h)

$\bar{V}$  = specific volume of fluid, lb/ft<sup>3</sup> (m<sup>3</sup>/kg)

$f$  = friction factor

$d$  = internal pipe diameter, in. (mm)

$\rho$  = fluid density, lb/ft<sup>3</sup> (kg/m<sup>3</sup>)

$S_g$  = specific gravity of gas relative to air = the ratio of molecular weight of the gas to that of air

$q'_h$  = rate of flow, ft<sup>3</sup>/h (m<sup>3</sup>/h) at standard conditions (14.7 psia and 60° F), SCFH (m<sup>3</sup>/s) at metric standard conditions (MSC) – 1.01325 bara and 15° C.

Babcock formula for steam flow at isothermal condition is

$$q'_h = 24,700 \left[ Yd^2 / S_g \right] (\Delta P \rho_1 / K)^{1/2}, \text{ CFH at 14.7 psia and } 60^\circ \text{ F} \quad (4-187)$$

or

$$q'_h = 40,700 Yd^2 \left[ (\Delta P) (P'_1) / (KT_1 S_g) \right]^{1/2} \quad (4-188)$$

where

$d$  = internal pipe diameter, in.

$Y$  = net expansion factor for compressible flow through orifices, nozzles, or pipe (see Figures 4-38a–c)

$K$  = loss coefficient for all valves, fittings, and pipe (resistance coefficient)

$$\left( K = fL / D + \sum_{i=\text{fittings+valves}} K_i \right)$$

$q'_h$  = flow rate, ft<sup>3</sup>/h at 14.7 psia and 60° F

$S_g$  = specific gravity of a gas relative to air = the ratio of the molecular weight of the gas to that of air

$P'$  = pressure, lb<sub>f</sub>/in.<sup>2</sup> abs

$\Delta P$  = pressure drop, psi

$T_1$  = inlet temperature, abs (°R = °F + 460)

$\rho_1$  = upstream density of steam, lb/ft<sup>3</sup>.

In SI units,

$$q'_h = 1.0312 \left[ Yd^2 / S_g \right] (\Delta p \rho_1 / K)^{1/2} \quad (4-189)$$

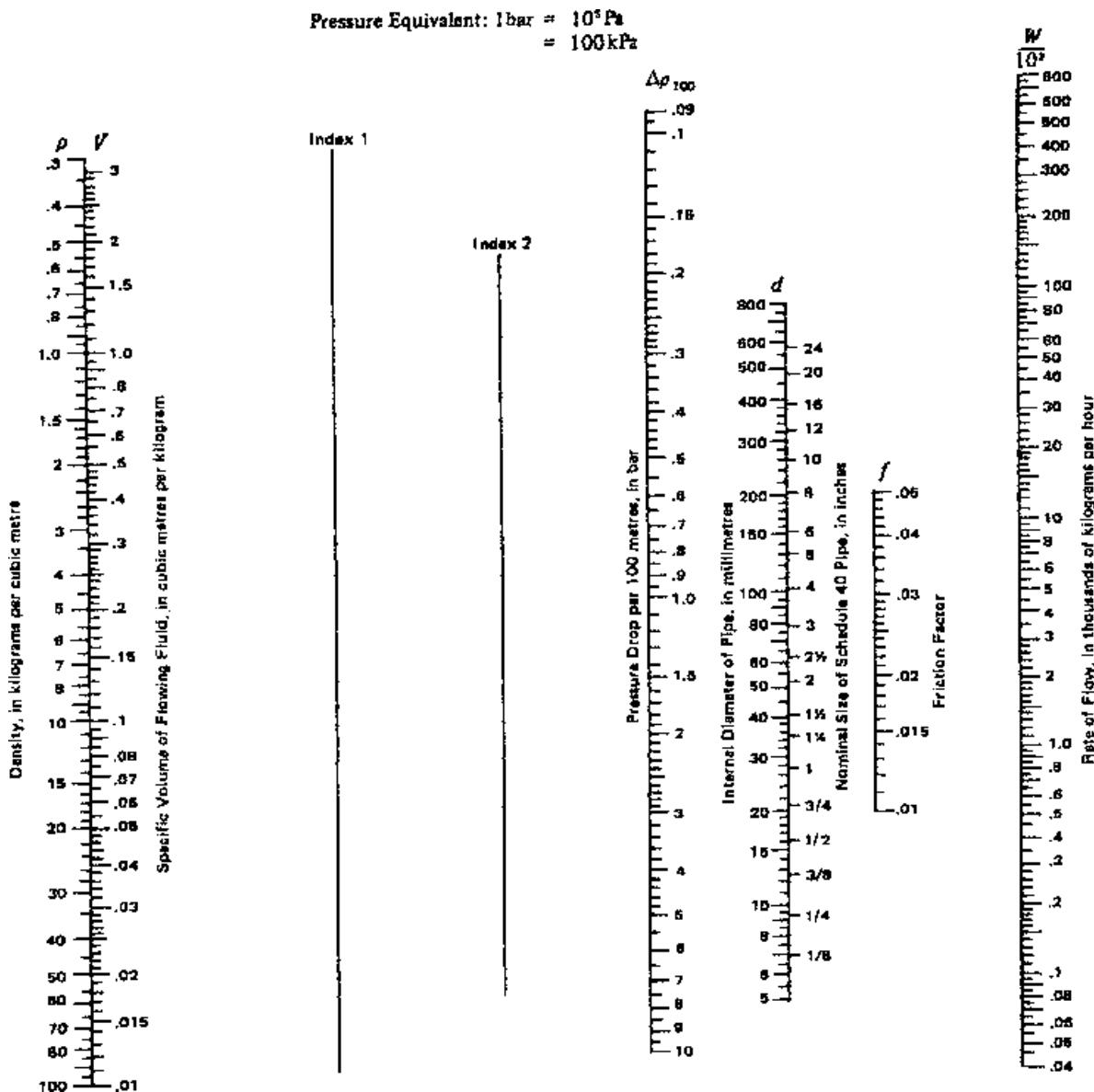
or

$$q'_h = 19.31 Yd^2 \left[ (\Delta p) (p'_1) / (KT_1 S_g) \right]^{1/2} \quad (4-190)$$

where

$d$  = internal pipe diameter, mm

$Y$  = net expansion factor for compressible flow through orifices, nozzles, or pipe



**Figure 4-39b** Pressure drop in compressible flow lines (Metric units). (Reprinted/Adapted with permission from "Flow of Fluids Through Valves, Fittings, and Pipe", Technical Paper No. 410, 1999, Crane Co. All rights reserved.)

$K$  = loss coefficient (resistance coefficient)

$p'$  = pressure, bara

$q'_h$  = flow rate,  $\text{m}^3/\text{h}$  at MSC (metric standard conditions – 1.01325 bar at  $15^\circ\text{C}$ )

$S_g$  = specific gravity of a gas relative to air = the ratio of the molecular weight of the gas to that of air

$\Delta p$  = pressure drop, bar

$T_1$  = inlet temperature, abs ( $K = {}^\circ\text{C} + 273$ )

$\rho_1$  = upstream density of steam,  $\text{kg}/\text{m}^3$

## B. ALTERNATE VAPOR/GAS FLOW METHODS

Note that all specialized or alternate methods for solving are based on simplified assumptions or empirical procedures presented earlier. They are not presented as better approaches to solving the specific problem.

Figures 4-40a and b (SI) are useful in solving the usual steam or any vapor flow problem for turbulent flow based on the modified Darcy friction factors. At low vapor velocities the results may be low; then use Figure 4-39a or b (SI). For steel pipe the limitations listed in (A) above apply.

- Determine  $C_1$  and  $C_2$  from Figure 4-40a or b (SI units) and Table 4-26 for the steam flow rate and assumed pipe size respectively. Use Table 4-6 or 4-19 to select steam velocity for line size estimate.
- Read the specific volume of steam at known temperature and pressure from steam tables.
- Calculate pressure drop (Figure 4-40a or b) per 100 ft of pipe from

$$\Delta P/100 \text{ ft} = C_1 C_2 \bar{V} \quad (4-191)$$

The Darcy formula can be written in the following form:

$$\Delta P_{100} = W^2 \left( \frac{0.000336 f}{d^5} \right) \bar{V} = (W^2 10^{-8}) \left( \frac{336,000 f}{d^5} \right) \bar{V}$$

$$C_1 = W^2 10^{-8} \quad C_2 = \frac{336,000 f}{d^5}$$

The simplified flow formula can then be written:

$$\Delta P_{100} = C_1 C_2 \bar{V} = \frac{C_1 C_2}{\rho}$$

$$C_1 = \frac{\Delta P_{100}}{C_2 \bar{V}} = \frac{\Delta P_{100} \rho}{C_2} \quad C_2 = \frac{\Delta P_{100}}{C_1 \bar{V}} = \frac{\Delta P_{100} \rho}{C_1}$$

$C_1$  = discharge factor from chart at right.

$C_2$  = size factor, from table on next page.

The limitations of the Darcy formula for compressible flow, as outlined on page 3-3, apply also to the simplified flow formula.

### Example 1

*Given:* Steam at 345 psig and 500 F flows through 8-inch Schedule 40 pipe at a rate of 240,000 pounds per hour.

*Find:* The pressure drop per 100 feet of pipe.

*Solution:*  $C_1 = .57$

$C_2 = 0.146$

$\bar{V} = 1.45$  .....page 3-17 or A-16

$$\Delta P_{100} = .57 \times 0.146 \times 1.45 = 12$$

### Example 2

*Given:* Pressure drop is 5 psi with 100 psig air at 90 F flowing through 100 feet of 4-inch Schedule 40 pipe.

*Find:* The flow rate in standard cubic feet per minute.

*Solution:*  $\Delta P_{100} = 5.0$

$C_2 = 5.17$

$\rho = 0.564$  .....page A-10

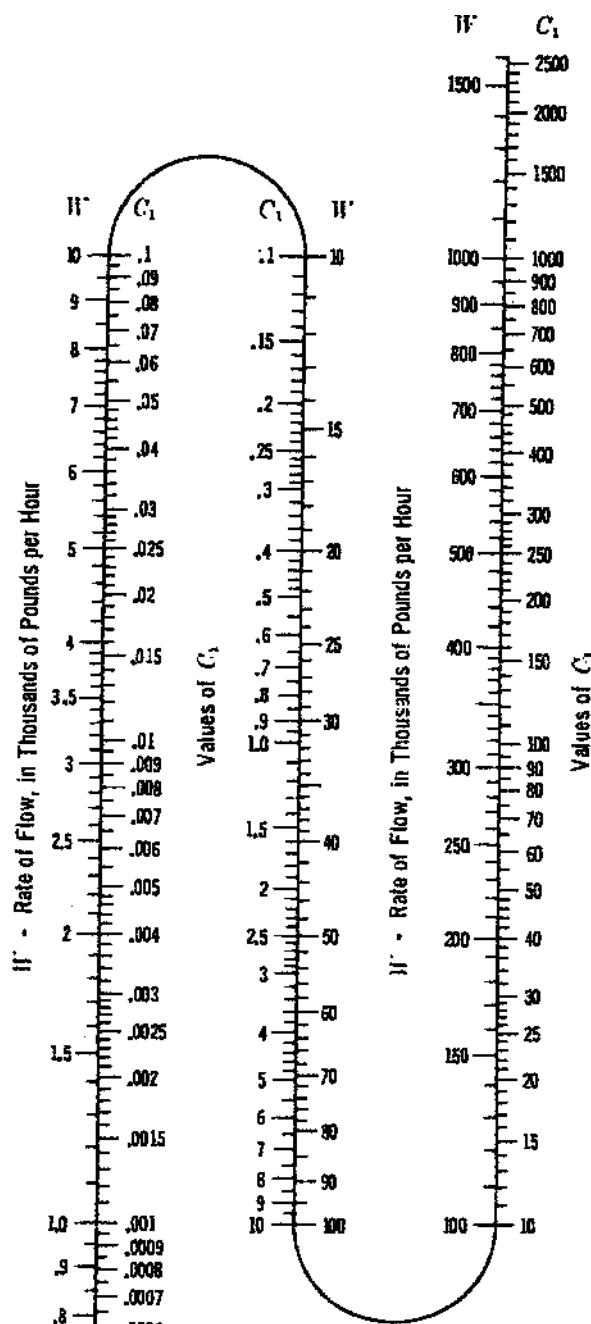
$$C_1 = (5.0 \times 0.564) \div 5.17 = 0.545$$

$$W = 23,000$$

$$q'_{in} = W \div (4.58 S_g) \quad \dots \text{page B-2}$$

$$q'_{in} = 23,000 \div (4.58 \times 1.0) = 5000 \text{ scfm}$$

Figure 4-40a Simplified flow formula for compressible fluids. (Reprinted/adapted with permission from "Flow of Fluids Through Valves, Fittings, and Pipe", Technical Paper No. 410, 1999, Crane Co. All rights reserved.)



For  $C_2$  values and an example on "determining pipe size", see the opposite page.

The Darcy formula can be written in the following form;

$$\Delta p_{100} = \frac{62.530 f W^2 V}{d^4} = \left(\frac{W^2}{10^4}\right) \left(\frac{62.530 \times 10^4 f}{d^4} V\right)^2$$

$$\text{Let } C_1 = \frac{W^2}{10^4} \text{ and } C_2 = \frac{62.530 \times 10^4 f}{d^4}$$

The simplified flow formula can then be written:

$$\Delta p_{100} = C_1 C_2 V = \frac{C_1 C_2}{\rho}$$

$$C_1 = \frac{\Delta p_{100}}{C_2 V} = \frac{\Delta p_{100} \rho}{C_2} \quad C_2 = \frac{\Delta p_{100}}{C_1 V} = \frac{\Delta p_{100} \rho}{C_1}$$

$C_1$  = discharge factor, from chart at right

$C_2$  = size factor from tables on pages 3-23 to 3-25

The limitations of the Darcy formula for compressible flow, as outlined on page 3-3 apply also to the simplified flow formula.

#### Example 1

**Given:** Steam at 24 bar absolute and  $250^\circ\text{C}$  flows through an 8-inch Schedule 40 pipe at a rate of 100 000 kilograms per hour.

**Find:** The pressure drop per 100 metres of pipe.

**Solution:**  $C_1 = 100$

$C_2 = 0.257$  ..... facing page

$V = 0.091 \text{ m}^3/\text{kg}$ , ..... page 3-17 or A-15

$$\Delta p_{100} = 100 \times 0.257 \times 0.091$$

$$\Delta p_{100} = 2.34 \text{ bar}$$

#### Example 2

**Given:** Pressure drop is 1 bar with 7 bar gauge air at  $30^\circ\text{C}$  flowing through 100 metres of 4 inch nominal size ISO steel pipe, 6.3 mm wall thickness.

**Find:** The flow rate in cubic metres per minute at metric standard conditions (1.013 25 bar and  $15^\circ\text{C}$ ).

**Solution:**  $\Delta p_{100} = 1$

$C_2 = 9.42$  ..... page 3-24

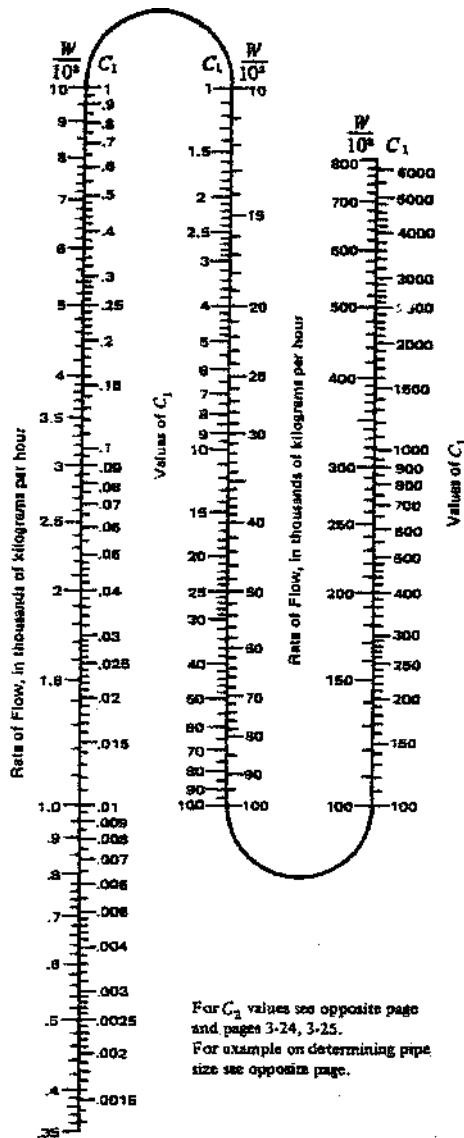
$\rho = 9.21$  ..... page A-10

$$C_1 = \frac{1 \times 9.21}{9.42} = 0.978$$

$$W = 9900$$

$$q'_m = W \div (73.5 S_g)$$
 ..... page B-2

$$q'_m = 9900 \div (73.5 \times 1) = 134.7 \text{ m}^3/\text{min}$$



**Figure 4-40b** Simplified flow formula for compressible fluids (Metric units). (Reprinted/adapted with permission from "Flow of Fluids Through Valves, Fittings, and Pipe", Technical Paper No. 410, 1999, Crane Co. All rights reserved.)

4. Determine the loss coefficient  $K$  of all fittings, valves, and so on, and hence the equivalent length ( $K = f L_{eq}/D$ ) or, alternatively, from Figure 4-24 or 4-25.
5. Determine expansion and contraction losses, fittings and at vessel connections.
6. Determine pressure drops through orifices and control valves.
7. Total system pressure drop

$$\Delta P_{TOTAL} = (L + L_{eq}) (\Delta P/100) + \text{Item5} + \text{Item6} \quad (4-192)$$

8. If pressure drop is too large (or greater than a percentage of the inlet system pressure), re-estimate line size and repeat calculations (see paragraph (A) above) and also examine pressure drop assumption for orifices and control valves.

#### C. AIR

For quick estimates for air line pressure drop and through an orifice, see Tables 4-27a and b.

#### D. BABCOCK EMPIRICAL FORMULA FOR STEAM

Comparison of results between the various empirical steam flow formulas suggests the Babcock equation is a good average for most design purposes at pressure 500 psia and below. For pipe lines smaller than 4 in., this relation may be 0–40% high [38].

$$p_1 - p_2 = \Delta P = 0.000131 (1 + 3.6/d) \frac{w^2 L}{\rho d^5} \quad (4-193)$$

$$\Delta P/100 \text{ ft} = w^2 F/\rho \quad (4-194)$$

**TABLE 4-26 Simplified Flow Formula For Compressible Fluids Pressure Drop, Rate of flow, and pipe sizes  
(use with figure 4-40a)**

Values of $C_2$								
Nominal Pipe Size Inches	Schedule Number	Value of $C_2$	Nominal Pipe Size Inches	Schedule Number	Value of $C_2$	Nominal Pipe Size Inches	Schedule Number	Value of $C_2$
$\frac{1}{8}$	40 s	7,920,000.00	5	40 s	1.59	16	10	0.00463
	80 x	26,200,000.00		80 x	2.04		20	0.00421
				120	2.69		30 s	0.00504
$\frac{1}{4}$	40 s	1,590,000.00		160	3.59		40 x	0.00549
	80 x	4,290,000.00		... xx	4.93		60	0.00612
$\frac{3}{8}$	40 s	319,000.00	6	40 s	0.610		80	0.00700
	80 x	718,000.00		80 x	0.798		100	0.00804
				120	1.015		120	0.00926
$\frac{1}{2}$	40 s	93,500.00		160	1.376		140	0.01099
	80 x	186,100.00		... xx	1.861		160	0.01244
	160	4,300,000.00						
	... xx	11,180,000.00	8	20	0.133			
				30	0.135	18	10	0.00247
				40 s	0.146		20	0.00256
$\frac{3}{4}$	40 s	21,200.00		60	0.163		... s	0.00266
	80 x	36,900.00		80 x	0.185		30	0.00276
	160	100,100.00		100	0.211		... x	0.00287
	... xx	627,000.00		120	0.252		40	0.00298
1	40 s	5,950.00		140	0.289		60	0.00335
	80 x	9,640.00		... xx	0.317		80	0.00376
	160	22,500.00		160	0.333		100	0.00435
	... xx	114,100.00					120	0.00504
$\frac{11}{4}$	40 s	1,408.00	10	20	0.0397		140	0.00573
	80 x	2,110.00		30	0.0421		160	0.00669
	160	3,490.00		40 s	0.0447			
	... xx	13,640.00		60 x	0.0514			
				80	0.0569	20	10	0.00141
							20 s	0.00150
$\frac{11}{2}$	40 s	627.00		100	0.0661		30 x	0.00161
	80 x	904.00		120	0.0753		40	0.00169
	160	1,656.00		140	0.0905		60	0.00191
	... xx	4,630.00		160	0.1052			
2	40 s	169.00	12	20	0.0157		80	0.00217
	80 x	236.00		30	0.0168		100	0.00251
	160	488.00		... s	0.0175		120	0.00287
	... xx	899.00		40	0.0180		140	0.00335
				... x	0.0195		160	0.00385
				60	0.0206			
$\frac{21}{2}$	40 s	66.70				24	10	0.000534
	80 x	91.80		80	0.0231		20 s	0.000565
	160	146.30		100	0.0267		... x	0.000597
	... xx	380.00		120	0.0310		30	0.000614
				140	0.0350		40	0.000651
				160	0.0423		60	0.000741
3	40 s	21.40	14	10	0.00949			
	80 x	28.70		20	0.00996		80	0.000835
	160	48.30		30 s	0.01046		100	0.000972
	... xx	96.60		40	0.01099		120	0.001119
				... x	0.01155		140	0.001274
				60	0.01244		160	0.001478
$\frac{31}{2}$	40 s	10.00		80	0.01416			
	80 x	37.70		100	0.01657			
4	40 s	5.17		120	0.01898			
	80 x	6.75		140	0.0218			
	120	8.94		160	0.0252			
	160	11.80						
	... xx	18.59						

*Note:*  
The letters s, x, and xx in the columns of Schedule Numbers indicate Standard, Extra Strong, and Double Extra Strong pipe respectively.

Source: By permission from Crane Co., Technical Paper No. 410, Engineering Div., 1957. See author's note at Figure 2-31.

Figure 4-41 is a convenient chart for handling most in-plant steam line problems. For long transmission lines over 200 ft, the line should be calculated in sections in order to re-establish the steam-specific density. Normally an estimated average  $\rho$  should be selected for each line increment to obtain good results.

Table 4-28 is used to obtain the value for "F" in Eq. (4-194).

#### 4.51 DARCY RATIONAL RELATION FOR COMPRESSIBLE VAPORS AND GASES

- Determine first estimate of line size by using suggested velocity from Table 4-6.
- Calculate Reynolds number  $Re$  and determine the friction factor,  $f$ , using Figure 4-5, 4-42a, or b (SI) (for steel pipe).

TABLE 4-27a Flow of Air Through Sch. 40 Pipe\* (use for estimating; for detailed calculations use friction factors  $f$ )

For lengths of pipe other than 100 feet, the pressure drop is proportional to the length. Thus, for 50 feet of pipe, the pressure drop is approximately one-half the value given in the table . . . for 300 feet, three times the given value, etc.

The pressure drop is also inversely proportional to the absolute pressure and directly proportional to the absolute temperature.

Therefore, to determine the pressure drop for inlet or average pressures other than 100 psi and at temperatures other than 60°F, multiply the values given in the table by the ratio:

$$\left( \frac{100 + 14.7}{P + 14.7} \right) \left( \frac{460 + t}{520} \right)$$

where:

"P" is the inlet or average gauge pressure in pounds per square inch, and,

"t" is the temperature in degrees Fahrenheit under consideration.

The cubic feet per minute of compressed air at any pressure is inversely proportional to the absolute pressure and directly proportional to the absolute temperature.

To determine the cubic feet per minute of compressed air at any temperature and pressure other than standard conditions, multiply the value of cubic feet per minute of free air by the ratio:

$$\left( \frac{14.7}{14.7 + P} \right) \left( \frac{460 + t}{520} \right)$$

#### Calculations for Pipe Other than Schedule 40

To determine the velocity of water, or the pressure drop of water or air, through pipe other than Schedule 40, use the following formulas:

$$v_a = v_{40} \left( \frac{d_{40}}{d_a} \right)^2$$

$$\Delta P_a = \Delta P_{40} \left( \frac{d_{40}}{d_a} \right)^4$$

Subscript "a" refers to the Schedule of pipe through which velocity or pressure drop is desired.

Subscript "40" refers to the velocity or pressure drop through Schedule 40 pipe, as given in the tables on these facing pages.

Free Air $q'$	Com- pressed Air	Pressure Drop of Air In Pounds per Square Inch Per 100 Feet of Schedule 40 Pipe							
		For Air at 100 Pounds per Square Inch Gauge Pressure and 60°F Temperature							
Cubic Feet Per Minute at 60°F and 14.7 psia	Cubic Feet Per Minute at 60°F and 100 psig	$\frac{1}{8}$ "	$\frac{1}{4}$ "	$\frac{3}{8}$ "	$\frac{1}{2}$ "	$\frac{5}{8}$ "	$\frac{3}{4}$ "	$\frac{7}{8}$ "	1"
1	0.128	0.261	0.083	0.018	0.020	0.042	0.062	0.094	0.037
2	0.256	1.31	0.285	0.064	0.020	0.042	0.062	0.094	0.037
3	0.384	3.06	0.605	0.133	0.042	0.071	0.094	0.129	0.055
4	0.513	4.83	1.04	0.226	0.071	0.106	0.133	0.162	0.062
5	0.641	7.45	1.58	0.343	0.106	0.207	0.236	0.266	0.071
6	0.769	10.6	2.23	0.406	0.148	0.276	0.316	0.359	0.081
8	1.025	18.9	3.89	0.848	0.285	0.662	0.748	0.819	0.200
10	1.282	28.7	5.96	1.26	0.356	0.944	1.00	1.09	0.270
15	1.912	43.8	2.23	0.834	0.201	0.302	0.359	0.444	0.044
20	2.563	72.8	4.76	1.43	0.345	0.102	0.126	0.155	0.055
25	3.204	... 35.6	7.34	2.21	0.526	0.156	0.189	0.220	0.071
30	3.845	... 40.7	10.6	3.15	0.748	0.219	0.255	0.286	0.071
40	4.886	... 44.2	14.2	4.24	1.00	0.293	0.373	0.388	0.081
45	5.726	... 46.4	16.4	5.49	1.30	0.379	0.495	0.444	0.081
50	6.767	... 49.1	23.1	6.90	1.62	0.474	0.116	0.155	0.081
50	6.406	28.5	8.49	1.99	0.578	0.149	0.167	0.199	0.071
60	7.690	21.4"	40.7	12.2	0.819	0.200	0.249	0.277	0.071
70	8.971	16.5	3.83	1.10	0.270	0.136	0.166	0.196	0.071
80	10.255	0.019	21.4	4.96	1.43	0.350	0.163	0.166	0.071
90	11.533	0.073	27.0	6.25	1.80	0.437	0.303	0.366	0.071
100	12.822	0.029	37.9	7.69	2.21	0.534	0.242	0.270	0.071
125	16.02	0.044	... 11.9	3.39	0.825	0.380	0.107	0.146	0.071
150	19.32	0.063	0.021	17.0	4.87	1.17	0.537	0.151	0.071
175	22.43	0.083	0.028	23.1	6.50	1.58	0.727	0.205	0.071
200	25.63	0.107	0.036	30.0	8.54	2.05	0.937	0.264	0.071
225	28.84	0.134	0.048	37.9	10.8	2.59	1.19	0.331	0.071
250	32.04	0.164	0.066	40.7	13.3	3.18	1.48	0.404	0.071
275	35.24	0.191	0.066	... 16.0	3.83	1.75	0.484	0.567	0.071
300	38.45	0.232	0.078	0.037	19.0	4.56	2.07	0.573	0.673
325	41.65	0.270	0.090	0.043	22.3	5.32	1.42	0.673	0.776
350	44.87	0.313	0.104	0.050	25.8	6.17	2.89	0.776	0.887
375	48.06	0.356	0.119	0.057	29.6	7.06	3.20	0.887	0.987
400	51.26	0.402	0.134	0.064	33.5	8.02	3.64	1.00	1.00
425	54.47	0.452	0.151	0.072	37.9	9.01	4.09	1.13	1.13
450	57.67	0.507	0.168	0.081	40.2	4.59	1.26	1.26	1.26
475	60.88	0.562	0.187	0.089	0.047	11.3	5.09	1.40	1.40
500	64.08	0.623	0.206	0.099	0.052	12.5	5.61	1.55	1.55
550	70.49	0.749	0.248	0.118	0.062	15.1	6.79	1.87	1.87
600	76.90	0.867	0.293	0.139	0.073	18.0	8.04	2.21	2.21
650	83.30	1.04	0.342	0.163	0.086	21.1	9.43	2.60	2.60
700	89.71	1.19	0.395	0.188	0.099	24.3	10.9	3.00	3.00
750	96.12	1.36	0.451	0.214	0.113	27.9	12.6	3.44	3.44
800	102.5	1.55	0.513	0.244	0.127	31.8	14.2	3.90	3.90
850	108.9	1.74	0.576	0.274	0.144	35.9	16.0	4.40	4.40
900	115.3	1.95	0.642	0.305	0.160	40.2	18.0	4.91	4.91
950	121.8	2.18	0.715	0.340	0.178	0.023	20.0	5.47	5.47
1000	128.2	2.40	0.788	0.375	0.197	0.025	22.1	6.06	6.06
1100	141.0	2.59	0.948	0.451	0.236	0.025	26.7	7.29	7.29
1200	163.8	3.14	1.13	0.833	0.279	0.036	31.8	8.63	8.63
1300	166.6	4.01	1.32	0.626	0.327	0.103	0.041	37.3	10.1
1400	179.4	4.66	1.52	0.718	0.377	0.119	0.047	11.8	11.8
1500	192.2	5.31	1.74	0.824	0.431	0.136	0.064	13.5	13.5
1600	205.1	6.04	1.97	0.932	0.490	0.154	0.061	15.3	15.3
1800	230.7	7.68	2.50	1.16	0.616	0.193	0.076	19.3	19.3
2000	256.3	9.44	3.06	1.46	0.757	0.237	0.094	0.023	23.9
2500	320.4	14.7	4.76	2.26	1.17	0.366	0.143	0.035	37.3
3000	384.5	21.1	6.82	3.20	1.67	0.524	0.204	0.051	0.016
3500	448.6	28.8	9.23	4.33	2.16	0.709	0.276	0.068	0.022
4000	512.6	37.6	12.1	5.66	2.84	0.919	0.358	0.088	0.028
4500	576.7	47.6	15.3	7.16	3.59	1.18	0.450	0.111	0.038
5000	640.8	... 18.8	8.85	4.56	1.43	0.551	0.136	0.043	0.018
6000	769.0	27.1	12.7	6.87	2.03	0.794	0.195	0.061	0.025
7000	897.1	36.9	17.2	8.34	2.76	1.07	0.262	0.082	0.034
8000	1025	... 22.5	11.7	3.89	1.39	0.339	0.107	0.044	0.022
9000	1153	... 28.5	14.9	4.54	1.76	0.427	0.134	0.053	0.033
10000	1182	... 35.2	18.4	5.60	2.16	0.526	0.164	0.067	0.033
11000	1210	... 22.2	6.78	2.60	0.633	0.197	0.061	0.031	0.031
12000	1238	... 26.4	8.07	3.09	0.783	0.234	0.069	0.036	0.036
13000	1266	... 31.0	9.47	3.53	0.884	0.273	0.073	0.042	0.042
14000	1294	... 36.0	11.0	4.21	1.02	0.316	0.139	0.053	0.044
15000	1322	... 12.6	4.84	1.17	0.364	0.148	0.044	0.022	0.022
16000	1351	... 14.3	5.80	1.33	0.411	0.167	0.044	0.022	0.022
18000	1427	... 18.2	6.96	1.56	0.520	0.213	0.053	0.031	0.031
20000	1563	... 23.4	8.60	2.01	0.642	0.260	0.062	0.040	0.040
22000	1620	... 27.1	10.4	2.50	0.771	0.314	0.071	0.044	0.044
24000	2076	... 32.3	12.4	2.57	0.918	0.371	0.121	0.053	0.053
26000	3332	... 37.9	14.5	3.49	1.12	0.433	0.141	0.062	0.062
28000	3688	... 16.9	4.04	1.25	0.506	0.314	0.121	0.053	0.053
30000	3945	... 19.3	4.64	1.42	0.530	0.314	0.139	0.062	0.062

\*By permission Technical Paper No. 410, Crane Co., Engineering Div., Chicago, 1957.

**TABLE 4-27b Discharge of Air Through an Orifice\* in Cubic Feet of Free Air per minute at Standard Atmospheric Pressure of 14.7 lb in.<sup>2</sup> abs and 70° F**

Gauge Pressure Before Orifice in Pounds Per square inch	Diameter of Orifice (in.)										
	$\frac{1}{64}$	$\frac{1}{32}$	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1
1	0.028	0.112	0.450	1.80	7.18	16.2	28.7	45.0	64.7	88.1	115
2	0.040	0.158	0.633	2.53	10.1	22.8	40.5	63.3	91.2	124	162
3	0.048	0.194	0.775	3.10	12.4	27.8	49.5	77.5	111	152	198
4	0.056	0.223	0.892	3.56	14.3	32.1	57.0	89.2	128	175	228
5	0.062	0.248	0.993	3.97	15.9	35.7	63.5	99.3	143	195	254
6	0.068	0.272	1.09	4.34	17.4	39.1	69.5	109	156	213	278
7	0.073	0.293	1.17	4.68	18.7	42.2	75.0	117	168	230	300
9	0.083	0.331	1.32	5.30	21.2	47.7	84.7	132	191	260	339
12	0.095	0.379	1.52	6.07	24.3	54.6	97.0	152	218	297	388
15	0.105	0.420	1.68	6.72	26.9	60.5	108	168	242	329	430
20	0.123	0.491	1.96	7.86	31.4	70.7	126	196	283	385	503
25	0.140	0.562	2.25	8.98	35.9	80.9	144	225	323	440	575
30	0.158	0.633	2.53	10.1	40.5	91.1	162	253	368	496	648
35	0.176	0.703	2.81	11.3	45.0	101	180	281	405	551	720
40	0.194	0.774	3.10	12.4	49.6	112	198	310	446	607	793
45	0.211	0.845	3.38	13.5	54.1	122	216	338	487	662	865
50	0.229	0.916	3.66	14.7	58.6	132	235	366	528	718	938
60	0.264	1.06	4.23	16.9	67.6	152	271	423	609	828	1082
70	0.300	1.20	4.79	19.2	76.7	173	307	479	690	939	1227
80	0.335	1.34	5.36	21.4	85.7	193	343	536	771	1050	1371
90	0.370	1.48	5.92	23.7	94.8	213	379	592	853	1161	1516
100	0.406	1.62	6.49	26.0	104	234	415	649	934	1272	1661
110	0.441	1.76	7.05	28.2	113	254	452	705	1016	1383	1806
120	0.476	1.91	7.62	30.5	122	274	488	762	1097	1494	1951
125	0.494	1.98	7.90	31.6	126	284	506	790	1138	1549	2023

*Notes:* Table is based on 100% coefficient of flow. For well-rounded entrance multiply values by 0.97. For sharp-edged orifices a multiplier of 0.65 may be used for approximate results.

Values for pressures from 1 to 15 lbs gauge calculated by standard adiabatic formula.

Values for pressures above 15 lb. gauge calculated by approximate formula proposed by S.A. Moss.

$$W_a = 0.5303 \frac{aCP_1}{\sqrt{T_1}}$$

where

$W_a$  = discharge in lb/s

$a$  = area of orifice in.<sup>2</sup>

$C$  = Coefficient of flow

$P_1$  = Upstream total pressure in lb/in.<sup>2</sup> abs

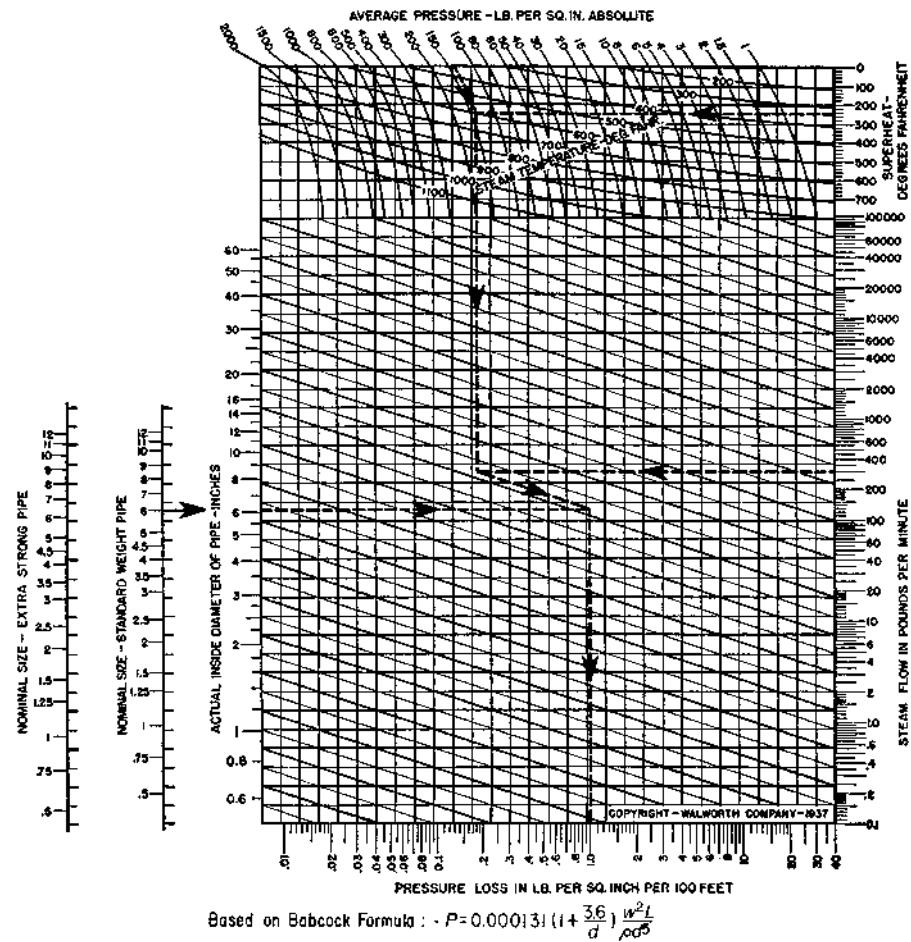
$T_1$  = Upstream temperature in °F, abs

Values used in calculating above table were:  $C = 1.0$ ,  $P_1$  = gauge pressure + 14.7 lb/in.<sup>2</sup>.

Weights ( $W$ ) were converted to volume using density of factor of 0.07494 lb/ft. This is correct for dry air at 14.7 lb/in.<sup>2</sup> absolute pressure and 70°F.

Formula cannot be used where  $P_1$  is less than two times the barometric pressure.

(Source: By permission from F.W. O'Neil, ed., "Compressed Air Data", Compressed Air Magazine, 5th Ed. New York, 1939 [37].)



**Figure 4-41** Steam flow chart. (By permission from Walworth Co. Note: Use for estimating only (Ludwig [19].)

3. Determine total straight pipe length,  $L$ .
4. Determine equivalent pipe length for fittings and valves,  $L_{eq}$ .
5. Determine or assume losses through orifice plates, control valves, equipment, contraction and expansion, and so on.
6. Calculate pressure drop,  $\Delta P/100$  ft (or use Figure 4-39a or b (SI)).

$$\Delta P/100 \text{ ft} = \frac{0.000336 f W^2}{\rho d^5} \quad (4-65)$$

$$= \frac{0.000000726 f T S_g (q'_h)^2}{P' d^5} \quad (4-65a)$$

7. Total pressure drop,  $\Delta P$  total

$$= (L + L_{eq}) (\Delta P/100) + \text{Item 5} \quad (4-195)$$

8. If total line or system pressure drop is excessive (or greater than a percentage of the inlet system pressure), examine the portion of pressure drop due to pipe friction and that due to other factors in the system. If the line pressure drop is a small portion of the total, little will be gained by increasing pipe size. Consider reducing losses through items in step 5 above. Recheck other pipes sizes as may be indicated.

## 4.52 VELOCITY OF COMPRESSIBLE FLUIDS IN PIPE

See Figures 4-43a and 4-43b

$$v_m = \frac{2.40 W \bar{V}}{a} = \frac{3.06 W \bar{V}}{d^2} = \frac{3.06 W}{d^2 \rho} \quad (4-196)$$

In SI units,

$$v_m = \frac{16,670 W \bar{V}}{a} = \frac{21,220 W \bar{V}}{d^2} = \frac{21,220 W}{d^2 \rho} \quad (4-196a)$$

where

$v_m$  = mean velocity in pipe, at conditions stated for  $\bar{V}$ , ft/min (m/s)

$a$  = cross-sectional area of pipe, in.<sup>2</sup> (mm<sup>2</sup>)

$W$  = flow rate, lb/h (kg/h)

$\bar{V}$  = fluid specific volume, ft<sup>3</sup>/lb (m<sup>3</sup>/kg)

$d$  = inside pipe diameter, in. (mm)

$\rho$  = fluid density, lb/ft<sup>3</sup> (kg/m<sup>3</sup>) at  $T$  and  $P$ .

Note that determining the velocity at the inlet conditions to a pipe may create significant error when results are concerned with the outlet conditions, particularly if the pressure drop is high. Even the average of inlet and outlet conditions is not sufficiently accurate for some systems; therefore conditions influenced by pressure drop

**TABLE 4-28 Factor "F" For Babcock Steam Formula**

Nominal Pipe Size (in.)	Standard Weight Pipe*	Extra Strong Pipe†
1/2	$955.1 \times 10^{-3}$	$2.051 \times 10^{-3}$
3/4	$184.7 \times 10^{-3}$	$340.8 \times 10^{-3}$
1	$45.7 \times 10^{-3}$	$77.71 \times 10^{-3}$
1 1/4	$9.432 \times 10^{-3}$	$14.67 \times 10^{-3}$
1 1/2	$3.914 \times 10^{-3}$	$5.865 \times 10^{-3}$
2	$951.9 \times 10^{-6}$	$1.365 \times 10^{-3}$
2 1/2	$351.0 \times 10^{-6}$	$493.8 \times 10^{-6}$
3	$104.7 \times 10^{-6}$	$143.2 \times 10^{-6}$
3 1/2	$46.94 \times 10^{-6}$	$62.95 \times 10^{-6}$
4	$23.46 \times 10^{-6}$	$31.01 \times 10^{-6}$
5	$6.854 \times 10^{-6}$	$8.866 \times 10^{-6}$
6	$2.544 \times 10^{-6}$	$3.354 \times 10^{-6}$
8	$587.1 \times 10^{-9}$	$748.2 \times 10^{-9}$
10	$176.3 \times 10^{-9}$	$225.3 \times 10^{-9}$
12	$70.32 \times 10^{-9}$	$90.52 \times 10^{-9}$
14 OD	$42.84 \times 10^{-9}$	$55.29 \times 10^{-9}$
16 OD	$21.39 \times 10^{-9}$	$27.28 \times 10^{-9}$
18 OD	$11.61 \times 10^{-9}$	$14.69 \times 10^{-9}$
20 OD	$6.621 \times 10^{-9}$	$8.469 \times 10^{-9}$
24 OD	$2.561 \times 10^{-9}$	$3.278 \times 10^{-9}$

(Source: By permission from The Walworth Co.)

\* Factors are based upon ID listed as Schedule 40

† Factors are based upon ID listed as Schedule 80

can produce more accurate results when calculations are prepared for successive sections of the pipe system (long or high pressure).

**EXAMPLE 4-12****Pressure Drop for Vapor System**

The calculations are presented in Figure 4-44, Line Size Specification Sheet.

Figure 4-45 is convenient when using Dowtherm vapor.

**4.53 ALTERNATE SOLUTION TO COMPRESSIBLE FLOW PROBLEMS**

There are several good approaches to recognizing the effects of changing conditions on compressible flow [39, 40].

**FRICTION DROP FOR AIR**

Table 4-27a is convenient for most air problems, noting that both free air ( $60^{\circ}\text{F}$  and 14.7 psia) and compressed air at 100 psig and  $60^{\circ}\text{F}$  are indicated. The corrections for other temperatures and pressures are also indicated. Figure 4-46 is useful for quick checking. However, its values are slightly higher (about 10%) than the rational values of Table 4-26, above about 1000 cfm of free air. Use for estimating only.

**EXAMPLE 4-13**  
**Steam Flow Using Babcock Formula**

Determine the pressure loss in 138 ft of 8-in. Sch. 40 steel pipe, flowing 86,000 lb/h of 150 psig steam (saturated).

Use Figure 4-41,  $w = 86,000/60 = 1432 \text{ lb/min}$

Reading from top at 150 psig, no superheat, down vertically to intersect the horizontal steam flow of 1432 lb/min, follow diagonal line to the horizontal pipe size of 8 in., and then vertically down to the pressure drop loss of 3.5 psi/100 ft.

For 138 ft (no fittings or valves), total  $\Delta P$  is  $138(3.5/100) = 4.82 \text{ psi}$ .

For comparison, solve by equation, using value of  $F = 587.1 \times 10^{-9}$  from Table 4-28.

$$\Delta P/100 \text{ ft} = (1432)^2 (587.1 \times 10^{-9})/0.364$$

$$= 3.31 \text{ psi}/100 \text{ ft}$$

$$\Delta P_{\text{total}} = (3.31/100)(138) = 4.57 \text{ psi}$$

These values are within graphical accuracy.

For the discharge of compressible fluids from the end of a short piping length into a larger cross section, such as a larger pipe, vessel, or atmosphere, the flow is considered adiabatic. Corrections are applied to the Darcy equation to compensate for fluid property changes due to the expansion of the fluid, and these are known as  $Y$  net expansion factors [4]. The corrected Darcy equation is:

For valves, fittings, and pipe (vapors/gases):

$$w = 0.525 Y d_i^2 \sqrt{\Delta P \rho_1 / K}, \text{ lb/s} \quad (4-197)$$

$$w = 1891 Y d_i^2 \sqrt{\Delta P \rho_1 / K}, \text{ lb/h} \quad (4-198)$$

In SI units,

$$w = 1.111 \times 10^{-6} Y d_i^2 \sqrt{\frac{\Delta P}{(K \bar{V}_1)}}, \text{ kg/s} \quad (4-199)$$

$$w = 1.265 Y d_i^2 \sqrt{\frac{\Delta P \rho_1}{K}}, \text{ kg/h} \quad (4-200)$$

For nozzles and orifices (vapors/gases):

$$w = 0.525 Y d_i^2 C' \sqrt{\frac{\Delta P}{\bar{V}_1}}, \text{ lb/s} \quad (4-201)$$

In SI units,

$$w = 1.111 \times 10^{-6} Y d_i^2 C' \sqrt{\frac{\Delta P}{(K \bar{V}_1)}}, \text{ kg/s} \quad (4-202)$$

For valves, fittings, and pipe (liquids):

English Engineering units

$$w = 0.525 d_i^2 \sqrt{\frac{\Delta P (\rho_1)}{K}}, \text{ lb/s} \quad (4-203)$$

(continued)

**EXAMPLE 4-13—(continued)**

In SI units,

$$w = 1.111 \times 10^{-6} d_i^2 \sqrt{\frac{\Delta P(\rho_1)}{K}}, \text{ kg/s}$$
(4-204)

English Engineering units

For nozzles and orifices (liquids):

$$w = 0.525 d_i^2 C' \sqrt{\Delta P(\rho_1)}, \text{ lb/s}$$
(4-205)

In SI units,

$$w = 1.111 \times 10^{-6} d_i^2 C' \sqrt{\Delta P(\rho_1)}, \text{ kg/s}$$
(4-206)

where

$d_i$  = pipe inside diameter, in. (mm)  
 $\rho_1$  = upstream fluid density, lb/ft<sup>3</sup> (kg/m<sup>3</sup>)  
 $w$  = rate of flow, lb/s (kg/s)

$\Delta P$  = pressure drop across the system, psi (bar) (inlet-discharge)  
 $K$  = total resistance (loss) coefficient of pipe, valves, fittings, and entrance and exit losses in the line

$Y$  = net expansion factor for compressible flow through orifices, nozzles, and pipes [4] (see Figure 4-38a, b, or c)

$\Delta P$  = pressure drop ratio in  $\Delta P/P'$ , used to determine  $Y$  from Figure 4-38a and b. The  $\Delta P$  is the difference between the inlet pressure and the pressure in the area of larger cross section

$C'$  = flow coefficient for orifices and nozzles (Figures 4-19 and 4-20).

For example, for a line discharging a compressible fluid to atmosphere, the  $\Delta P$  is the inlet gauge pressure or the difference between the absolute inlet pressure and the atmospheric pressure absolute. When  $\Delta P/P'$  falls outside the limits of the  $K$  curves on the charts, sonic velocity occurs at the point of discharge or at some restriction within the pipe, and the limiting value for  $Y$  and  $\Delta P$  must be determined from the tables in Figures 4-38a and b and used in the velocity equation,  $v_s$ , above [4].

Figures 4-38a and b are based on the perfect gas laws and for sonic conditions at the outlet end of a pipe. For gases/vapors that deviate from these laws, such as steam, the same application will yield about 5% greater flow rate. For improved accuracy, use the charts in Figures 4-38a and b (SI) to determine the downstream pressure when sonic velocity occurs. Then use the fluid properties at this condition of pressure and temperature in:

$$v_s = \sqrt{kgRT} = \sqrt{kg144P'\bar{V}}, \text{ ft/s}$$
(4-207)

$$\text{or } v_s = \sqrt{\gamma RT} = \sqrt{\gamma P'\bar{V}}, \text{ m/s}$$
(4-208)

to determine the flow rate at this condition from

$$v = q/A = 183.3 q/d^2 = 0.0509 W/(\rho)(d^2), \text{ ft/s}$$
(4-30)

In SI units,

$$v = q/A = 1.273 \times 10^6 q/d^2 = 21.22 \frac{Q}{d^2} = 354 \frac{W}{\rho d^2}, \text{ m/s}$$
(4-31)

where

$d$  = internal diameter of pipe, in. (mm)

$A$  = cross section of pipe, ft<sup>2</sup> (m<sup>2</sup>)

$q$  = ft<sup>3</sup>/sc (m<sup>3</sup>/s) at flowing conditions

$T$  = temperature, °R ( $K = 273 + t$ )

$t$  = fluid temperature, °C

$k$  =  $\gamma$  = ratio of specific heats ( $C_p/C_v$ )

$P'$  = pressure, psia (N/m<sup>2</sup> abs)

$W$  = flow, lb/h (kg/h)

$v$  = velocity, mean or average, ft/s (m/s).

These conditions are similar to flow through orifices, nozzles, and venturi tubes. Flow through nozzles and venturi devices is limited by the critical pressure ratio,  $r_c$  = downstream pressure/upstream pressure, at sonic conditions (see Figure 4-38c).

For nozzles and venturi meters, the flow is limited by critical pressure ratio and the minimum value of  $Y$  is to be used. For flow of gases and vapors through nozzles and orifices:

$$q = Y C' A \sqrt{\frac{2g(144)\Delta P}{\rho}}, \text{ ft}^3/\text{s flow}$$
(4-209)

$$\text{or } q = Y C' A \sqrt{\frac{2\Delta P}{\rho}}, \text{ m}^3/\text{s}$$
(4-210)

where

$\beta$  = ratio of orifice throat diameter to inlet diameter

$C'$  = flow coefficient for nozzles and orifices (see Figures 4-19 and 4-20), when used as per ASME specification for differential pressure

$\rho$  = fluid density, lb/ft<sup>3</sup> (kg/m<sup>3</sup>)

$A$  = cross-sectional flow area, ft<sup>2</sup> (m<sup>2</sup>).

*Note:* The use of  $C'$  eliminates the calculation of velocity of approach. The flow coefficient  $C'$  is  $C' = C_d / \sqrt{1 - \beta^4}$ ,  $C_d$  = discharge coefficient for orifices and nozzles [4].

For compressible fluids flowing through nozzles and orifices use Figures 4-19 and 4-20, using  $h_L$  or  $\Delta P$  as differential static head or pressure differential across taps located one diameter upstream, and 0.5 diameters downstream from the inlet face of orifice plate or nozzles, when values of  $C$  are taken from Figures 4-19 and 4-20 [4]. For any fluid:

$$q = C' A \left[ \left( \frac{2g(144)\Delta P}{\rho} \right) \right]^{1/2}, \text{ ft}^3/\text{s}$$
(4-211)

In SI units,

$$q = C' A \sqrt{\frac{2\Delta P}{\rho}}, \text{ m}^3/\text{s flow}$$
(4-212)

(continued)

**EXAMPLE 4-13—(continued)**

For estimating purposes in liquid flow with viscosity similar to water through orifices and nozzles, the following can be used [7]:

$$Q = 19.636 C' d_1^2 \sqrt{h} \sqrt{\frac{1}{1 - \left(\frac{d_o}{d_i}\right)^4}}, \text{ gpm} \quad (4-213)$$

In SI units,

$$Q = 0.2087 C' d_1^2 \sqrt{h} \sqrt{\frac{1}{1 - \left(\frac{d_o}{d_i}\right)^4}}, \text{ L/min} \quad (4-214)$$

$$\text{where } \frac{d_o}{d_i} \text{ is greater than 0.3} \quad (4-215)$$

$$Q = 19.636 C' d_o^2 \sqrt{h} \text{ where } \frac{d_o}{d_i} \text{ is less than 0.3} \quad (4-216)$$

In SI units,

$$Q = 0.2087 C' d_o^2 \sqrt{h}, \text{ L/min} \quad (4-217)$$

$$\text{or [4], } W = 157.6 d_o^2 C' \sqrt{h_L \rho^2} \quad (4-218)$$

$$= 1891 d_o^2 C' \sqrt{\Delta P \rho} \quad (4-219)$$

In SI units,

$$W = 0.01252 d_o^2 C \sqrt{h_L \rho^2} = 1.265 d_o^2 C \sqrt{\Delta p \rho} \quad (4-220)$$

where

$Q$  = liquid flow, gpm (l/min)

$C$  = flow coefficient for orifices and nozzles = discharge coefficient corrected for velocity of approach =  $C_d / \sqrt{1 - \beta^4}$

$d_o$  = diameter of orifice or nozzle opening, in. (mm)

$d_i$  = pipe inside diameter in which orifice or nozzle is installed, in. (mm)

$h$  = static pressure head existing at a point, in. (m) of fluid.

$h_L$  = loss of static pressure head due to fluid flow, m of fluid.

$C'$  = flow coefficient (see Figure 4-47 for water and Figures 4-20 and 4-21 for vapors or liquids)

$q$  =  $\text{ft}^3/\text{s}$  ( $\text{m}^3/\text{s}$ ) at flowing conditions

$r_c$  = critical pressure ratio for compressible flow,  $= P'_2 / P'_1$

$\Delta P$  = pressure drop, psi

$\Delta p$  = pressure drop, bar ( $h_L$  and  $\Delta p$  measured across taps at 1 diameter and 0.5 diameter)

$W$  = flow rate, lb/h (kg/h).

Flow of gases and vapors (compressible fluids) through nozzles and orifices (for flow field importance, see [41]). From [4]:

$$q = Y C' A \left[ \left( \frac{2g(144) \Delta P}{\rho} \right) \right]^{1/2}, \text{ ft}^3/\text{s} \quad (4-221)$$

(at flowing conditions)

In SI units,

$$q = Y C' A \sqrt{2 \Delta p / \rho}, \text{ m}^3/\text{s} \quad (4-222)$$

where

$Y$  = net expansion factor from Figure 4-38a or b

$\Delta P$  = differential pressure (equal to inlet gauge pressure when discharging to atmosphere)

$\rho$  = weight density of fluid,  $\text{lb}/\text{ft}^3$  ( $\text{kg}/\text{m}^3$ ) at flowing conditions

$A$  = cross-sectional area of orifice or nozzle,  $\text{ft}^2$  ( $\text{m}^2$ )

$C'$  = flow coefficient from Figure 4-38a or 4-38b.

$$W = 1891 Y d_o^2 C' \sqrt{\frac{\Delta P}{V_1}}, \text{ lb/h} \quad (4-223)$$

$$\text{or } W = 1.265 d_o^2 C' \sqrt{\Delta p \rho_1}, \text{ kg/h} \quad (4-224)$$

where

$d_o$  = internal diameter of orifice, in. (mm)

$V_1$  = specific volume of fluid,  $\text{ft}^3/\text{lb}$  ( $\text{m}^3/\text{kg}$ )

$\rho_1$  = density of fluid,  $\text{lb}/\text{ft}^3$  ( $\text{kg}/\text{m}^3$ )

$\Delta p$  = pressure drop, psi (bar).

$$q' = 11.30 Y d_o^2 C' \sqrt{\frac{\Delta P P'_1}{T_1 S_g}}, \text{ ft}^3/\text{s}, \text{ at } 14.7 \text{ psia and } 60^\circ \text{F} \quad (4-225)$$

where

$S_g$  = sp gr of gas relative to air = mol wt of gas/29

$T_1$  = absolute temperature,  $^\circ \text{R} = (460 + ^\circ \text{F})$

$P'_1$  = pressure, psia.

In SI units,

$$q = 0.005363 Y d_o^2 C' \sqrt{\frac{\Delta p P'_1}{T_1 S_g}}, \text{ m}^3/\text{s} \quad (4-226)$$

where

$T_1 = 273.15 + t$

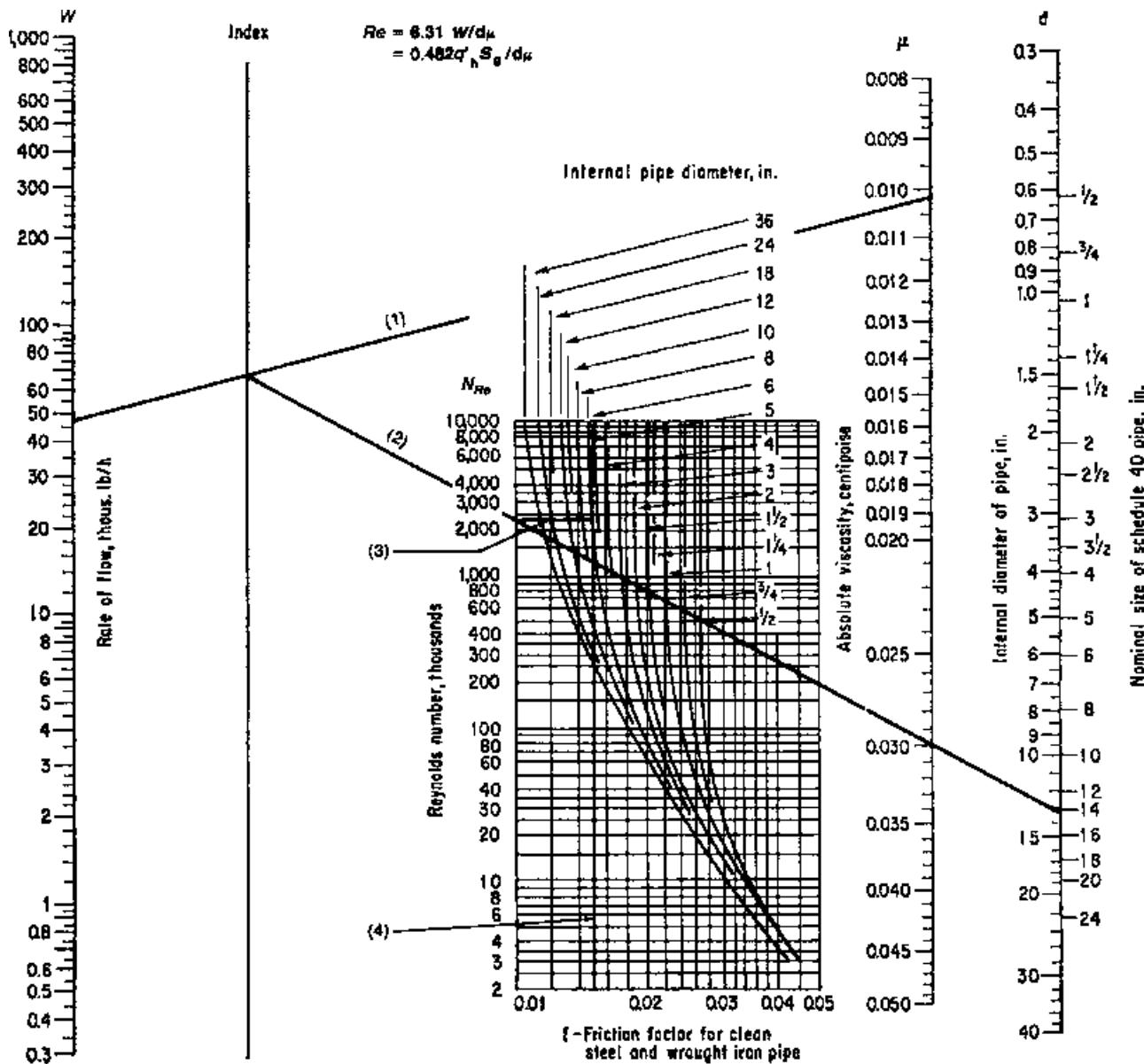
$t$  = fluid temperature,  $^\circ \text{C}$

$P'_1$  = pressure, bara

$\Delta p$  = pressure drop, bar

$S_g$  = sp gr of gas relative to air = mol wt of gas/29

$Y$  = net expansion factor compressibility flow through orifices, nozzles, or pipe.



**Figure 4-42a** Reynolds number of compressible flow, steel pipe. (Reprinted/adapted with permission from "Flow of Fluids Through Valves, Fittings, and Pipe", Technical Paper No. 410, 1999, Crane Co. All rights reserved.)

#### 4.54 PROCEDURE

A. How to determine pipe size for given capacity and pressure drop

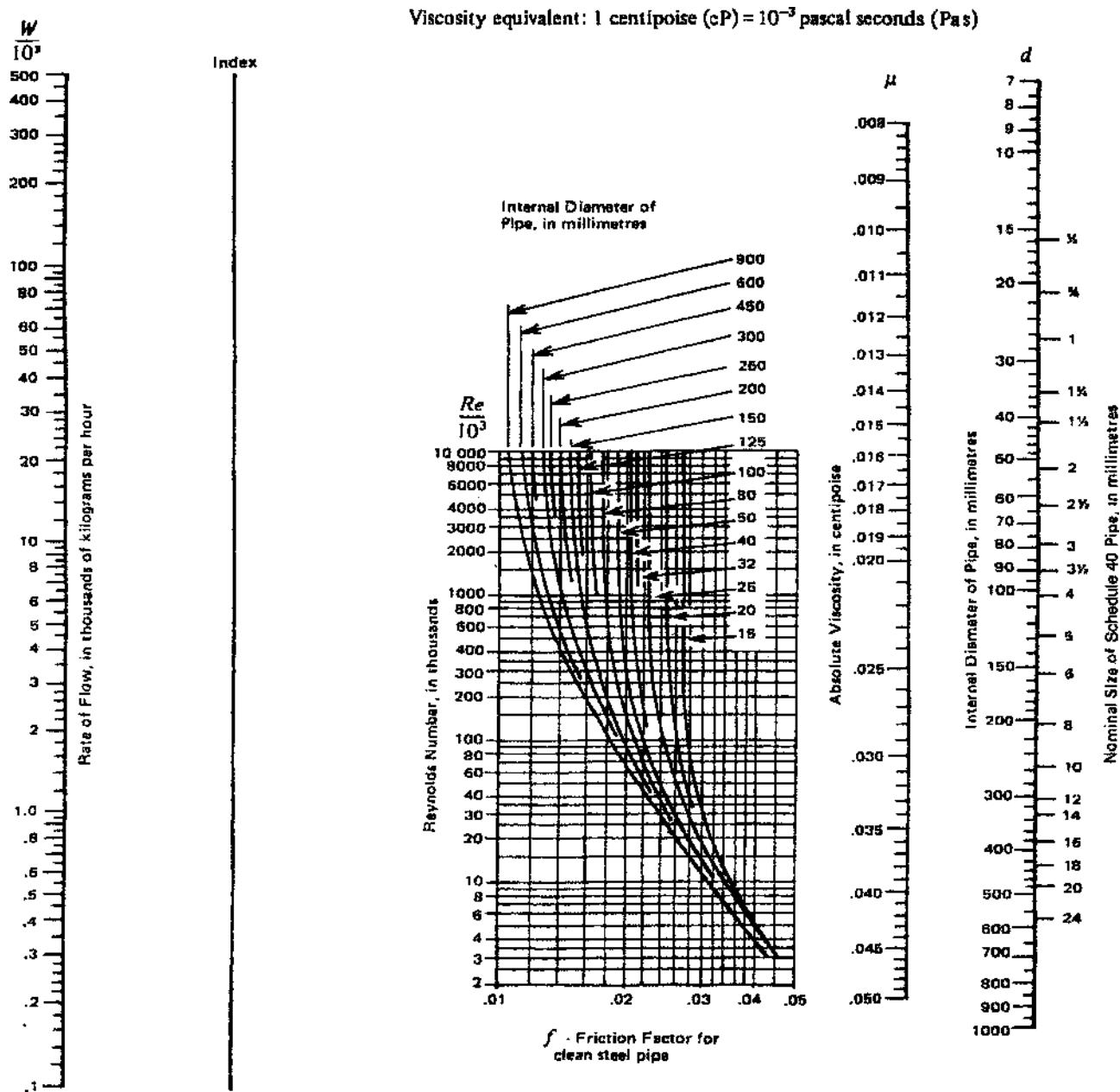
1. Assume a pipe diameter, and calculate velocity in feet per second using the given flow.
2. Calculate sonic velocity for fluid using Eqs (4-118)–(4-123).
3. If sonic velocity of step 2 is greater than calculated velocity of step 1, calculate line pressure drop using usual flow equations. If these velocities are equal, then the pressure drop calculated will be the maximum for the line, using usual flow equations. If sonic velocity is less than the velocity of step 1, reassume line size and repeat calculations.

B. How to determine flow rate (capacity) for a given line size and fixed pressure drop

This is also a trial-and-error solution following the pattern of (A), except capacities are assumed and the pressure drops are calculated to find a match for the given conditions of inlet pressure, calculating back from the outlet pressure.

C. How to determine pressure at inlet of pipe system for fixed pipe size and flow rate

Determining the sonic flow rate involves knowledge of the local conditions at the exit. However, this is difficult to establish and highly complicated in practice as it requires extensive iterative computations.



**Figure 4-42b** Reynolds number of compressible flow, steel pipe (Metric units). (Reprinted/adapted with permission from "Flow of Fluids Through Values, Fittings, and Pipe", Technical Paper No. 410, 1999, Crane Co. All rights reserved.)

#### 4.55 FRICTION DROP FOR COMPRESSIBLE NATURAL GAS IN LONG PIPE LINES

Tests of the US Department of the Interior, Bureau of Mines, reported in Monograph 6 *Flow of Natural Gas Through High-Pressure Transmission Lines* [42] indicate that the Weymouth formula gives good results on flow measurements on lines 6 in. in diameter and larger when operating under steady flow conditions of 30–600 psig.

Long gas transmission lines of several miles length are not considered the same as process lines inside plant connecting process equipment where the lengths usually are measured in feet (metres) or hundreds of feet (metres). Some plants will transfer a

manufactured gas, such as oxygen, carbon dioxide, or hydrogen, from one plant to an adjacent plant. Here the distance can be from 1 to 15 miles (kilometres). In such cases, the previously discussed flow relations for compressible gases can be applied in incremental segments, recalculating each segment, and then the results can be checked using one of the formulas that follow. However, there are many variables to evaluate and understand in the Weymouth, Panhandle, Panhandle-A and modifications as well as other flow relationships. Therefore, they will be presented for reference. However, the engineer should seek out the specialized flow discussions on this type of flow condition. The above mentioned equations are derived somewhat empirically for the flow of a natural gas containing some entrained liquid (perhaps 5–12%), and the

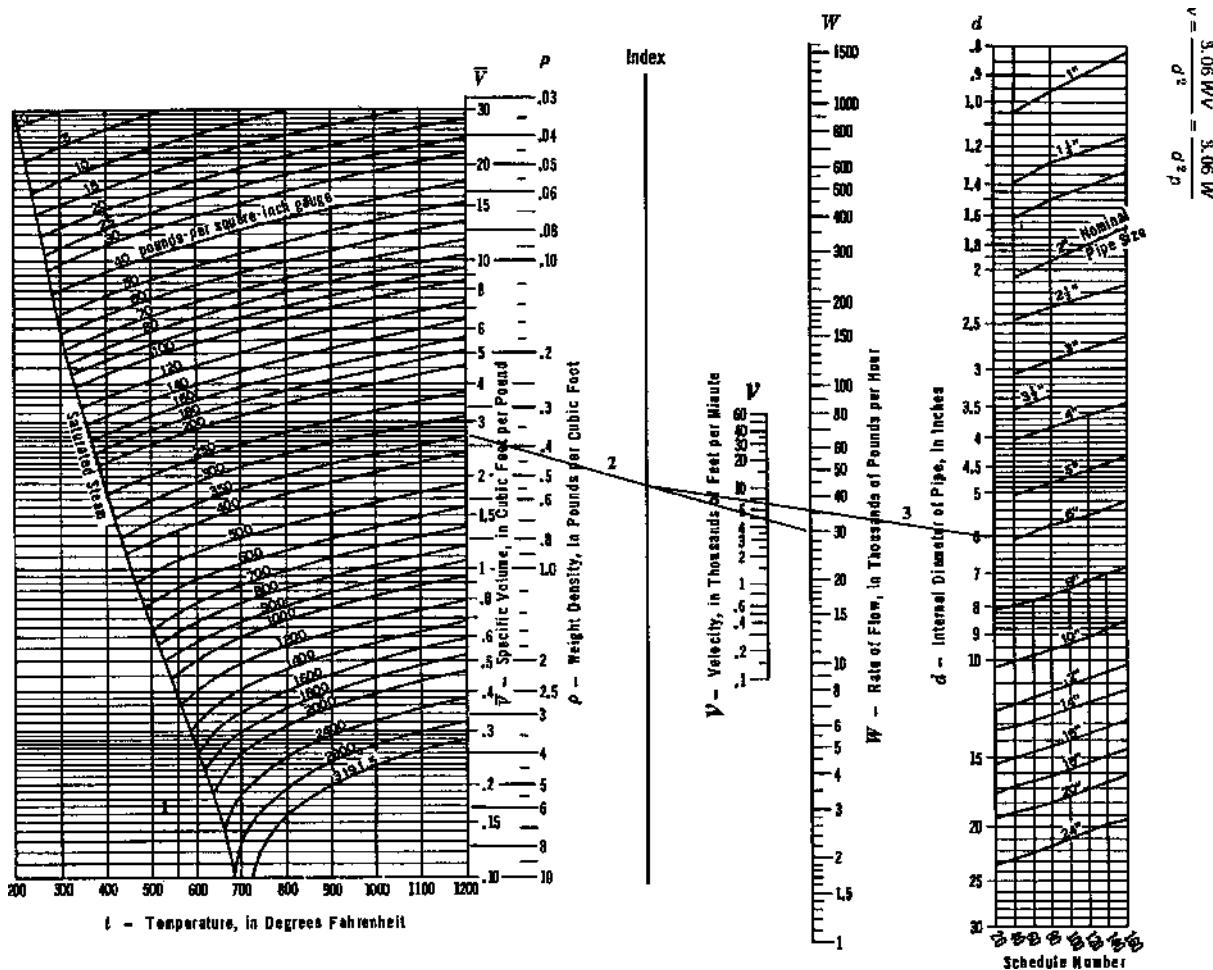


Figure 4-43a Velocity in compressible flow lines. (Reprinted/adapted with permission from "Flow of Fluids Through Valves, Fittings, and Pipe", Technical Paper No. 410, 1999, Crane Co. All rights reserved).

#### EXAMPLE 4-14

What is the flow rate of natural gas through a ruptured exchanger tube assuming (1) a complete break near the tube sheet (see Figure 4-34), and (2) isothermal flow? Compare the flow rate to adiabatic condition. The following data are [39]

Pressure in exchanger tubes, $P_1$ , psig	1110
Relief valve set pressure, $P_2$ , psig	400
Gas temperature, °F	100
Compressibility factor, Z	0.9
Molecular weight	18.7
Ratio of specific heats $k = C_p/C_v$	1.3
Gas viscosity, $\mu$ , cP	0.1
Exchanger tubes, $3/4$ in. Sch. 160, ID in.	0.614
Tube length, ft	20
Friction factor for complete turbulence, $f$	0.026

#### Solution

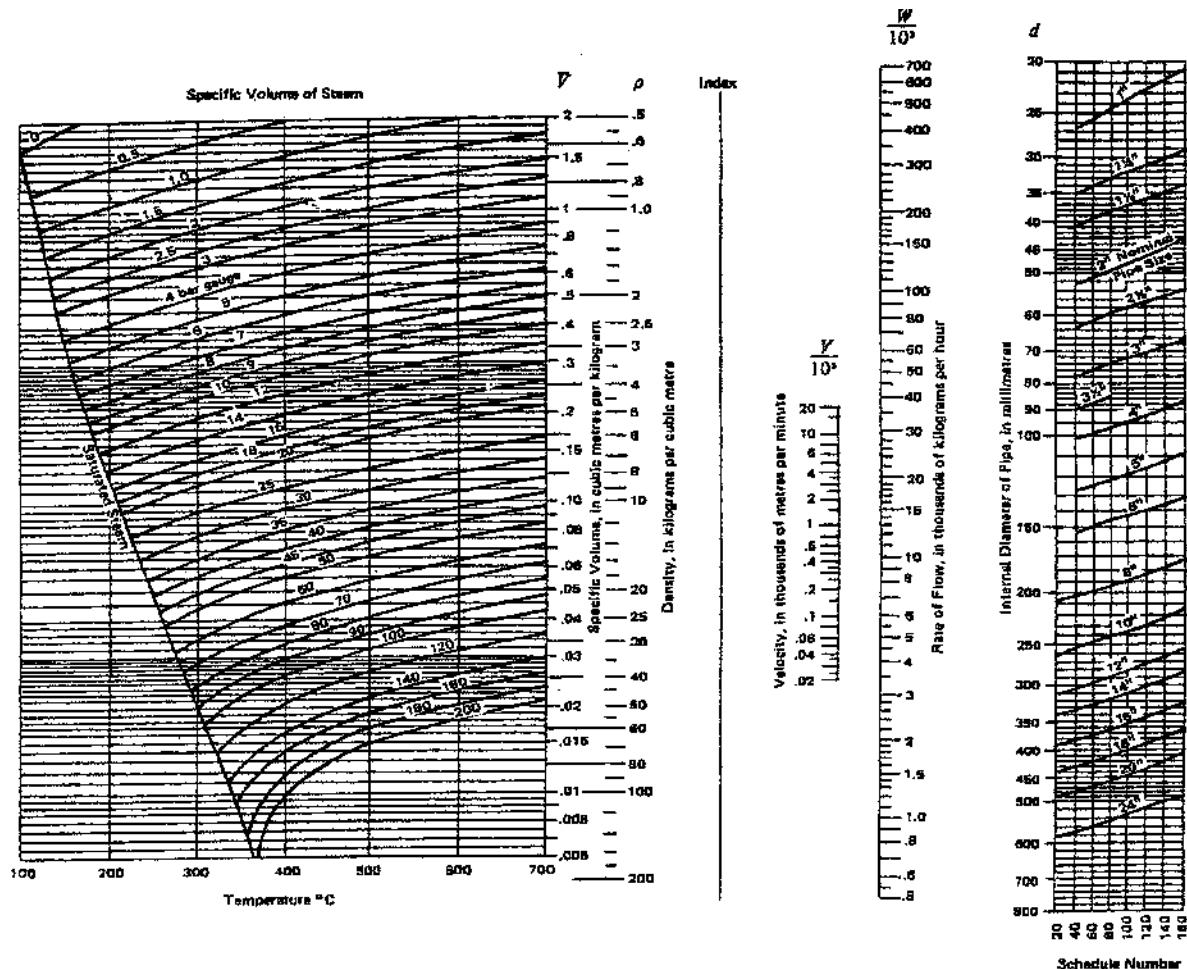
Internal diameter of  $3/4$  in. Sch. 160 = 0.614 in. = 0.051 ft

$$\begin{aligned} \text{Upstream density is } \rho_1 &= \frac{M_w P_1}{10.72 Z T} \\ &= \frac{(18.7)(1110 + 14.7)}{(10.72)(0.9)(100 + 460)} \\ &= 3.892 \text{ lb/ft}^3 \end{aligned}$$

Total resistance (loss) coefficient  $K_{\text{Total}}$  is

$$\begin{aligned} K_{\text{Total}} &= \left( f_T \frac{L}{D} + \sum K_f + K_{\text{entrance}} + K_{\text{exit}} \right) \\ &= \left( 0.025 \frac{20}{0.051} + 0.5 + 1.0 \right) \\ &= 11.30 \end{aligned}$$

(continued)



**Figure 4-43b** Velocity in compressible flow lines (Metric units). (Reprinted/Adapted with permission from "Flow of Fluids Through Valves, Fittings, and Pipe", Technical Paper No. 410, 1999, Crane Co. All rights reserved.)

#### EXAMPLE 4-14—(continued)

Pressure in the exchanger,  $P_1 = 1110 + 14.7 = 1124.7 \text{ psia}$

Relief valve set pressure,  $P_2 = 400 + 14.7 = 414.7 \text{ psia}$

Using Eq. (4-153), the gas flow rate through the ruptured tube is

$$G = 1335.6 (0.614)^2 \left[ \left( \frac{3.892}{11.3 + 2 \ln (1124.7 / 414.7)} \right) \times \left( \frac{1124.7^2 - 414.7^2}{1124.7} \right) \right]^{0.5}$$

$$= 8492.5 \text{ lb/h}$$

Gas Reynolds number is

$$Re = 6.31 \frac{W}{d\mu} = 6.31 \frac{8492.5}{(0.614)(0.1)}$$

$$= 8.7 \times 10^5 (\text{fully turbulent})$$

Gas velocity  $v$  is

$$v = 0.0509 \frac{W}{d^2 \rho_1} = 0.0509 \frac{8492.5}{(0.614)^2 (3.892)}$$

$$= 294.61 \text{ ft/s}$$

Sonic velocity  $v_s$  is

$$v_s = 68.1 \sqrt{k P_1 / \rho_1}$$

$$= 68.1 \sqrt{(1.3)(1124.7) / (3.892)}$$

$$= 1319.9 \text{ ft/s}$$

The Mach number at the inlet  $Ma_1$  is

$$Ma_1 = \frac{v}{v_s} = \frac{294.61}{1319.9}$$

$$= 0.223$$

(continued)

 <b>A.K.C. Technology</b> <small>Consultant A.K. Coker Ph.D., C.Eng., M.I.Chem.E 199 High Street, Dorking, Surrey, RH4 1RL, U.K. E-Mail Address: kcoke11@hotmail.com Telephone/Fax: (0121) 3553923</small>		SHEET NO. _____																																																
By AKC	<u>LINE SIZE SHEET</u>	Job No. _____																																																
Date _____		Charge No. _____																																																
Line No. LP-61	Flow Sheet Drawing No. _____																																																	
Line Description Vent through Exchanger for Tower T - 3																																																		
Fluid in line N <sub>2</sub> +Hydrocarbon	Temperature 140 °F																																																	
GPM (Calc.) _____	Pressure 5.3 psig																																																	
CFM (Calc.) 2060	sp gr 0.975																																																	
lb/h (Calc.) 10,841	sp vol 11.3 cu.ft./lb																																																	
Recommended Velocity fp	Viscosity 0.019 cp																																																	
<table border="1" style="width: 100%; border-collapse: collapse;"> <thead> <tr> <th colspan="4">Straight pipe, fittings, valves expansions, contraction, etc</th> </tr> <tr> <th>Item</th> <th>No.</th> <th>Unit Eq. ft.</th> <th>Total Eq. ft.</th> </tr> </thead> <tbody> <tr> <td>St. Line</td> <td></td> <td></td> <td>5</td> </tr> <tr> <td>Gate Valve.</td> <td>1</td> <td>11</td> <td>11</td> </tr> <tr> <td></td> <td>1</td> <td>6</td> <td>6</td> </tr> <tr> <td>Tee - S0</td> <td>1</td> <td>50</td> <td>50</td> </tr> <tr> <td></td> <td></td> <td></td> <td></td> </tr> <tr> <td></td> <td></td> <td>Total</td> <td>72</td> </tr> </tbody> </table>		Straight pipe, fittings, valves expansions, contraction, etc				Item	No.	Unit Eq. ft.	Total Eq. ft.	St. Line			5	Gate Valve.	1	11	11		1	6	6	Tee - S0	1	50	50							Total	72	<table border="1" style="width: 100%; border-collapse: collapse;"> <thead> <tr> <th>Item</th> <th>Pressure Drop in psi</th> </tr> </thead> <tbody> <tr> <td>Pipe &amp; Equivalent</td> <td>0.0617</td> </tr> <tr> <td>Orifice</td> <td></td> </tr> <tr> <td>Motor Valve</td> <td></td> </tr> <tr> <td>Miscellaneous</td> <td></td> </tr> <tr> <td>Exchanger drop</td> <td>1.50</td> </tr> <tr> <td></td> <td></td> </tr> <tr> <td>Total</td> <td>1.56</td> </tr> </tbody> </table>	Item	Pressure Drop in psi	Pipe & Equivalent	0.0617	Orifice		Motor Valve		Miscellaneous		Exchanger drop	1.50			Total	1.56
Straight pipe, fittings, valves expansions, contraction, etc																																																		
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Total	1.56																																																	
Cross-sect. area, 10" pipe = 0.547sq ft		Estimated line size 10" (existing)																																																
Velocity = 2270 / 0.547 = 4150 ft/min		Actual Velocity 4150 fps																																																
		Unit Loss per 100 ft. 0.0857 psi																																																
		Total head loss in feet of liquid 37																																																
		Total pressure drop in psi 1.56																																																
Selected pipe size 10"	Material & Weight Schedule 40, Steel																																																	
Calculations:	$Re=6.31 \frac{W}{d\mu}=[(6.31)(12,000)/(10.02)(0.019)]=3.98\times 10^5$																																																	
	$f=0.0158 \quad \rho=1/V$																																																	
$\Delta P/100ft = \frac{(0.000336)(f)(w)^2}{\rho g} = \frac{(0.000336)(0.0158)(12,000)}{(10.02)^5(1/11.3)}$	= -0.0857psi/100 equivalent feet of pipe (as pipe, fittings, valves, etc.)																																																	
$\Delta P_{Total} \text{ (friction)} = (0.0857/100)(72) = 0.0617 \text{ psi}$																																																		
Checked by: _____ Date: _____																																																		

Figure 4-44 Line size sheet. Example of pressure drop for a vapor system (Example 4-12).

**EXAMPLE 4-14—(continued)**

Since the Mach number is less than 1, the flow through the pipe is subsonic.

At adiabatic condition, the net expansion factor  $Y$  is

$$\Delta P/P_1 = (1124.7 - 414.7) / 1124.7 = 0.63$$

The loss coefficient  $K_{Total} = 11.3$

From Figure 4-38a,  $Y$  at  $\Delta P/P = 0.63$  and  $k = 1.3$  by interpolation is  $Y = 0.77$

Using Eq. (4-198), the gas flow rate is

$$w_s = (1891)(0.77)(0.614)^2 \sqrt{\frac{(710)(3.892)}{(11.3)}} \\ = 8584.1 \text{ lb/h}$$

The percentage deviation between the two conditions (isothermal and adiabatic) is 1.07%. As such, there is very little difference in flow rate between the two conditions. However, the adiabatic flow rate is always greater than the isothermal flow rate. Table 4-29 shows the computer result of Example 4-14 at isothermal condition. The Excel spreadsheet Example 4-14.xls gives calculations for isothermal and adiabatic conditions of the above example.

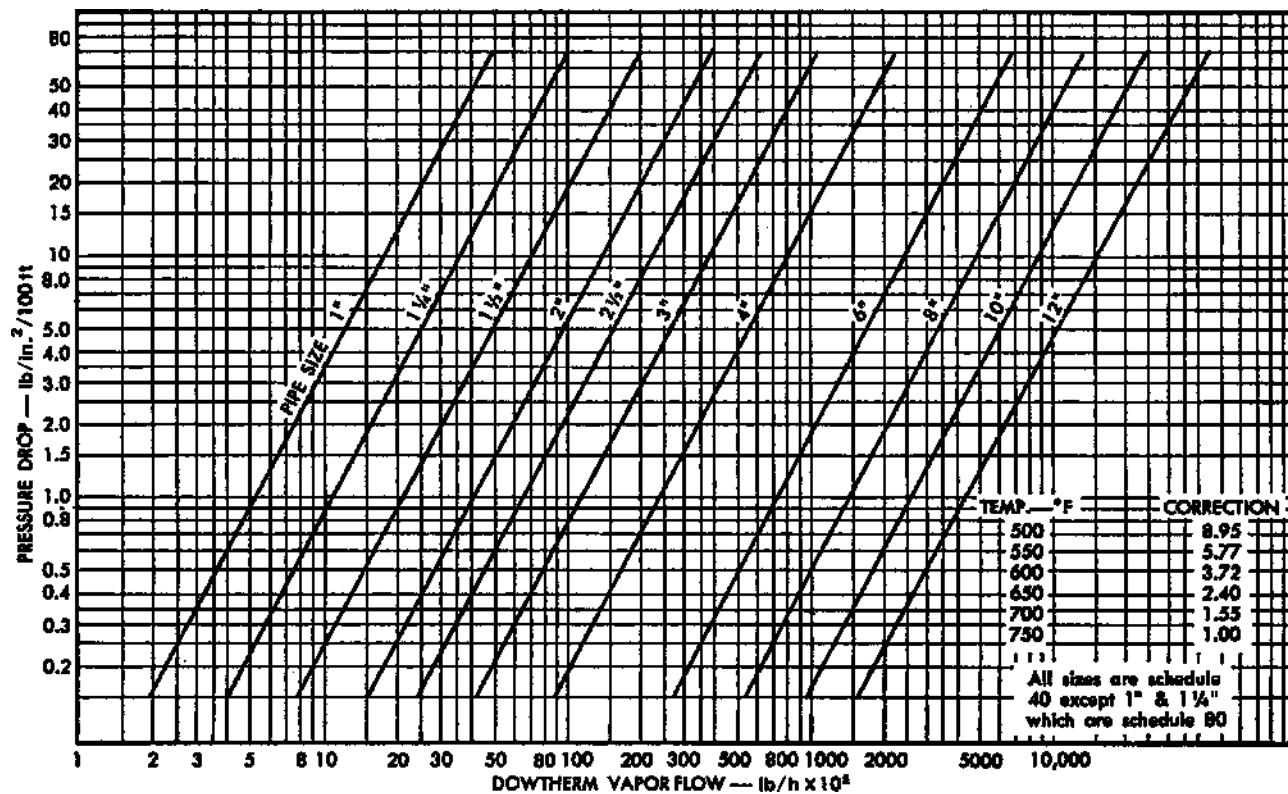


Figure 4-45 Pressure drop, Dowtherm "A"® vapor in steel pipe. (By permission from Struthers Wells Corp., Bull. D-45.)

#### EXAMPLE 4-15

##### Gas Flow Through Sharp-edged Orifice

A 1-in. Sch. 40 pipe is flowing methane at 40 psig and 50°F. The flange taps across the orifice (0.750 in. diameter) show a 3 psi pressure differential. Determine the flow rate through the orifice.

*Solution*

$\text{CH}_4$ ; sp gr =  $S_g = 0.553$

Gas constant =  $R = 96.5$

Ratio sp ht =  $k = 1.26$

Absolute system pressure =  $P = 40 + 14.7 = 54.7 \text{ psia}$

$$\Delta P/P_1 = 3.0/54.7 = 0.0549$$

Pipe ID = 1.049 in.

$$d_o/d_1 = 0.750/1.049 = 0.7149$$

From Figure 4-38a,  $Y = 0.97$ ; from Figure 4-20

$$C' (\text{assumed turbulent}) = \frac{C_d}{\left[1 - (d_o/d_1)^4\right]^{1/2}}$$

where  $C_d$  = orifice discharge coefficient, uncorrected for velocity of approach.

$$C' = 0.74 \text{ at estimated } Re \geq 2000$$

$$\text{Temperature} = 460 + 50 = 510^\circ \text{F}$$

$$\text{Density} = \rho = \frac{144P}{RT} = \frac{144(54.7)}{(96.5)(510)} = 0.160 \text{ lb/ft}^3$$

$$W = 1891 Y d_o^2 C (\Delta P \rho)^{1/2} \quad (4-227)$$

$$W = 1891 (0.97) (0.750)^2 (0.74) [(3)(0.160)]^{1/2}$$

$$W = 529 \text{ lb/h methane}$$

Checking:

Calculate  $Re$  to verify turbulence; if not in reasonable agreement, recalculate  $C'$  and balance of solution,

Viscosity of methane = 0.0123 cP

$$= 6.31 W / d \mu$$

$$= 6.31 (529)/(0.750)(0.0123)$$

$$Re = 361,841$$

This is turbulent and satisfactory for the assumption. For helpful quick reference for discharge of air through an orifice, see Table 4-27b.

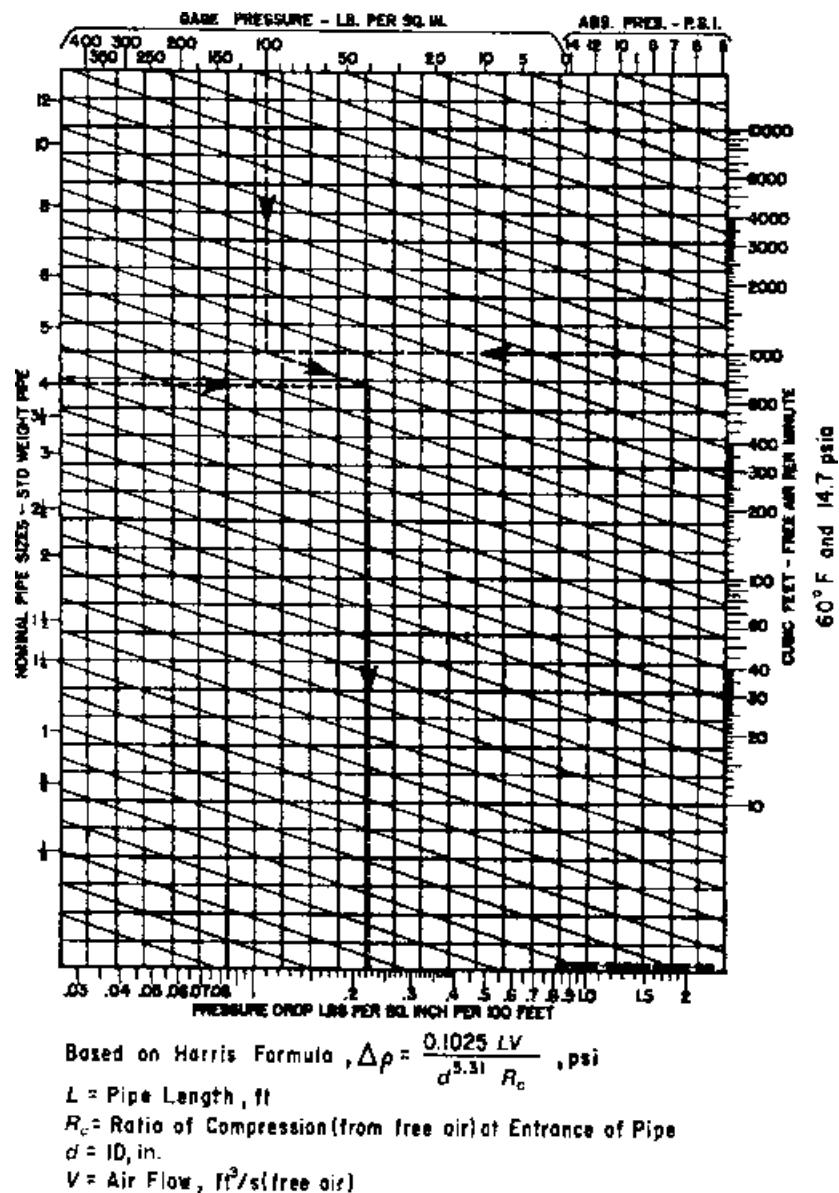


Figure 4-46 Compressed air flow chart. (By permission from Walworth Co.) Note: Use for estimating only (Ludwig [19].)

RE - ENTRANT TUBE	SHARP - EDGED	SQUARE EDGED	RE - ENTRANT TUBE	SQUARE EDGED	WELL ROUNDED
LENGTH: 1/2 to 1 DIA.	STREAM CLEARS SIDES		LENGTH: 2-1/2 DIA.	TUBE FLOWER TUBE	
$C = .52$	$C = .61$	$C = .61$	$C = .73$	$C = .82$	$C = .98$

Figure 4-47 Discharge coefficients for liquid flow (By permission, Cameron Hydraulic data, Ingersol-Rand Co., Washington, NJ, 1979.)

**TABLE 4-29 Computer Results of Compressible Fluid Flow of Example 4-14 (Isothermal condition)**

MAXIMUM FLOW OF A COMPRESSIBLE FLUID	
*****	
PIPE INTERNAL DIAMETER, inch:	.612
MAXIMUM MASS FLOW RATE, lb/h:	8315.029
FLUID DENSITY, lb/ft <sup>3</sup> :	3.893
PIPE FRICTION FACTOR:	.0260
RESIST. COEFF. DUE TO FRICTIONAL LOSS:	10.1961
RESIST. COEFF DUE TO FITTINGS+VALVES:	1.5002
TOTAL RESISTANCE COEFFICIENT:	11.6963
INLET FLUID PRESSURE, psia:	1124.700
OUTLET FLUID PRESSURE, psia:	414.700
PRESSURE DROP, psi:	710.000
FLUID VELOCITY, ft/sec.:	290.286
FLUID SONIC VELOCITY, ft/sec.:	1391.393
MACH NUMBER AT INLET:	.2086
MACH NUMBER AT CRITICAL CONDITION:	.8771
CRITICAL PRESSURE, psia:	242.168
FLUID VISCOSITY, cP:	.1000
FLUID FLOW IS:	SUBSONIC
-----	

results vary accordingly, even though they are not two-phase flow equations.

Table 4-30 [26] tabulates the transmission factors of the various equations. Most of these are established as correction factors to the correlation of various test data.

Dunning [43] recommends this formula (from [42]) for 4- to 24-in. (102–575 mm) diameter lines with specific gravity of gas near 0.60, and actual mean velocities from 15 to 30 fps (4.6–9.1 m/s) at temperature near 60° F (16° C).

The Bureau of Mines report states that minor corrections for bends, tees, and even compressibility are unnecessary due to the greater uncertainties in actual line conditions. Their checks with the Weymouth relation omitted these corrections. The relation with

**TABLE 4-30 Dry-Gas Flow Transmission Factors\***

Title	Transmission Factor ( $\sqrt{f}$ )
Weymouth	$11.2D^{0.167}$
Blasius	$3.56Re^{0.125}$
Panhandle A	$6.87Re^{0.073}$
Modified Panhandle	$16.5Re^{0.0196}$
Smooth pipe law (Nikuradse)	$4 \log(Re\sqrt{f}) - 0.4$
Rough pipe law (Nikuradse)	$4 \log \frac{(D)}{(2\varepsilon)} + 3.48$
Colebrook	$4 \log \frac{(D)}{(2\varepsilon)} + 3.48 - 4 \log \left(1 + 9.35 \frac{D}{2Re\sqrt{f}}\right)$

(Source: By permission from Hope, P.M. and R.G. Nelson, "Fluid Flow, Natural Gas", McKetta, J.J. ed., *Encyclopedia of Chemical Processing and Design*, Vol. 22, M. Dekker, 1985, p. 304 [26].)

Note: D = inches

\*See listing of source references in [15].

pressure base of 14.4 psia is to be used with the Bureau of Mines multipliers [42].

$$q_h \text{ (at 14.4 psia and } 60^\circ \text{ F)} = 36.926 d^{2.667} \left[ \frac{P_1^2 - P_2^2}{L_m} \right]^{1/2}, \text{ scfh} \quad (4-228)$$

$$q'_h \text{ (at 14.4 psia and } 60^\circ \text{ F)}$$

$$= 28.0 d^{2.667} \left[ \frac{P_1^2 - P_2^2}{S_g L_m} \left( \frac{520}{T} \right) \right]^{1/2}, \text{ scfh} \quad (4-229)$$

Weymouth's formula [8] has friction established as a function of diameter and may be solved by using alignment charts.

The Weymouth formula is also expressed (at standard condition) as:

$$q_d = 433.9 E d^{2.667} \left( \frac{T_s}{P_s} \right), \left[ \frac{(P'_1)^2 - (P'_2)^2}{S_g T_1 L_m Z} \right]^{1/2} \quad (4-230)$$

where

E = transmission factor, usually taken as  $1.10 \times 11.2 d^{0.167}$   
(omit for pipe size smaller than 24 in.)

d = Pipe, ID, in.

$T_s = 520^\circ \text{ R}$

$P_s = 14.7 \text{ psia}$

$T_1 = \text{flowing temperature of gas, } ^\circ \text{R}$

$q_d = \text{flow rate, ft}^3/\text{day} \text{ ( at standard conditions of 14.7 psia and } 520^\circ \text{ R) }$

$P'_1 = \text{inlet pressure, psia}$

$P'_2 = \text{outlet pressure, psia}$

Z = compressibility factor

L = pipe length, miles

SCC = Standard condition (14.7 psia and 60° F).

or from [4]:

$$q_h = 28.0 d^{2.667} \left\{ \left[ \frac{(P'_1)^2 - (P'_2)^2}{S_g L_m} \right] \left[ \frac{520}{T} \right] \right\}^{1/2} \quad (4-231)$$

where

d = pipe internal diameter, in.

T = flowing temperature of gas, °R

$S_g = \text{specific gravity of gas}$

$q_h = \text{gas flow rate ft}^3/\text{h, at } 60^\circ \text{ F and 14.4 psia}$

$P'_1 = \text{inlet pressure, psia}$

$P'_2 = \text{outlet pressure, psia}$

Z = compressibility factor

$L_m = \text{pipe length, miles.}$

In SI units,

$$q'_h = 2.61 \times 10^{-8} d^{2.667} \left[ \left( \frac{(P'_1)^2 - (P'_2)^2}{S_g L_m} \right) \left( \frac{288}{T} \right) \right]^{1/2}, \text{ m}^3/\text{h} \quad (4-232)$$

where

d = internal pipe diameter, mm

T = flowing temperature of gas, K = (273 + °C)

$q_h = \text{m}^3/\text{h gas at metric standard conditions (MSC) of } P_s \text{ and } T_s$

$P'_1 = \text{inlet pressure, N/m}^2 \text{ abs}$

$P'_2 = \text{outlet pressure, N/m}^2 \text{ abs}$

Z = compressibility factor

$L_m = \text{pipe length, km}$

MSC = Metric standard conditions (1.01325 bara and 15° C).

**EXAMPLE 4-16****Use of Base Correction Multipliers**

Tables 4-31–4-34 are set up with base reference conditions. In order to correct or change any base condition, the appropriate multiplier(s) must be used.

A flow of  $5.6 \times 10^6 \text{ ft}^3/\text{day}$  has been calculated using Weymouth's formula [8], with these conditions: measuring base of  $60^\circ\text{F}$  and 14.4 psia; flowing temperature of  $60^\circ\text{F}$ ; and specific gravity of 0.60. Suppose for comparison purposes the base conditions must be changed to measuring base of  $70^\circ\text{F}$  and 14.7 psia; flowing temperature of  $80^\circ\text{F}$ ; and specific gravity of 0.74.

Multipliers from the tables are

Pressure base	0.9796
Temperature base	1.0192
Specific gravity base	0.9005
Flowing temperature base	0.9813

New base flow

$$\begin{aligned} &= (5,600,000)(0.9796)(1.0192)(0.9005)(0.9813) \\ &= 4,940,000 \text{ ft}^3/\text{day} \end{aligned}$$

**TABLE 4-31 Pressure-Base Multipliers for Quantity**

New Pressure Base (lb/in. <sup>2</sup> abs)	Multiplier
12.00	1.2000
13.00	1.1077
14.00	1.0286
14.40	1.0000
14.65	0.9829
14.7	0.9796
14.9	0.9664
15.4	0.9351
16.4	0.8780

(Source: By permission from Johnson, T.W. and W.B. Berwald, *Flow of Natural Gas Through High Pressure Transmission Lines*, Monograph No. 6, US Dept. of Interior, Bureau of Mines, Washington, DC.)

Note: Multiplier =  $\frac{14.4}{\text{New pressure base, lb/in.}^2 \text{ abs}}$

**TABLE 4-32 Temperature-Base Multipliers for Quantity**

New Pressure Base (lb/in. <sup>2</sup> abs)	Multiplier
45	0.9712
50	0.9808
55	0.9904
60	1.0000
65	1.0096
70	1.0192
75	1.0288
80	1.0385
85	1.0481
90	1.0577
95	1.0673
100	1.0769

(Source: By permission from Johnson, T.W. and W.B. Berwald, *Flow of Natural Gas Through High Pressure Transmission Lines*, Monograph No. 6, US Dept. of Interior, Bureau of Mines, Washington, DC.)

Note: Multiplier =  $\frac{460 + \text{new temperature base, } ^\circ\text{F}}{460 + 60}$

$$q_g = 0.028 E \left[ \frac{P_1^2 - P_2^2}{S_g^{0.961} TL_m Z} \right]^{0.51} d^{2.53} \quad (4-233)$$

or

$$P_1^2 - P_2^2 = \left[ \frac{q_g}{0.028 E d^{2.53}} \right]^{1.9607} S_g^{0.961} TL_m Z \quad (4-234)$$

## 4.56 PANHANDLE-A GAS FLOW FORMULA [4]

The Panhandle equation assumes that the friction factor can be represented by a straight line of constant negative slope in the moderate Reynolds number region of the Moody diagram (Figure 4-5). The equation is considered to be slightly better than the  $\pm 10\%$  accuracy of the Weymouth formula and is given by

**TABLE 4-33 Specific Gravity Multipliers for Quantity**

Specific Gravity	0	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08	0.09
0.5	1.0954	1.0847	1.0742	1.0640	1.0541	1.0445	1.0351	1.0260	1.0171	1.0084
0.6	1.0000	0.9918	0.9837	0.9759	0.9682	0.9608	0.9535	0.9463	0.9393	0.9325
0.7	0.9258	0.9193	0.9129	0.9066	0.9005	0.8944	0.8885	0.8827	0.8771	0.8715
0.8	0.8660	0.8607	0.8554	0.8502	0.8452	0.8402	0.8353	0.8305	0.8257	0.8211
0.9	0.8165	0.8120	0.8076	0.8032	0.7989	0.7947	0.7906	0.7865	0.7825	0.7785
1.0	0.7746	0.7708	0.7670	0.7632	0.7596	0.7559	0.7524	0.7488	0.7454	0.7419
1.1	0.7385	0.7352	0.7319	0.7287	0.7255	0.7223	0.7192	0.7161	0.7131	0.7101

(Source: By permission from Johnson, T.W. and W.B. Berwald, *Flow of Natural Gas Through High Pressure Transmission Lines*, Monograph No. 6, US Dept. of Interior, Bureau of Mines, Washington, DC.)

Note: Multiplier =  $\left( \frac{0.600}{\text{Actual specific gravity}} \right)^{1/2}$

TABLE 4-34 Flowing -Temperature Multipliers For Quantity

Temperature (° F)	0	1	2	3	4	5	6	7	8	9
10	1.0632	1.0621	1.0609	1.0598	1.0586	1.0575	1.0564	1.0552	1.0541	1.0530
20	1.0518	1.0507	1.0496	1.0485	1.0474	1.0463	1.0452	1.0441	1.0430	1.0419
30	1.0408	1.0398	1.0387	1.0376	1.0365	1.0355	1.0344	1.0333	1.0323	1.0312
40	1.0302	1.0291	1.0281	1.0270	1.0260	1.0249	1.0239	1.0229	1.0219	1.0208
50	1.0198	1.0188	1.0178	1.0167	1.0157	1.0147	1.0137	1.0127	1.0117	1.0107
60	1.0098	1.0088	1.0078	1.0068	1.0058	1.0048	1.0039	1.0029	1.0019	1.0010
70	1.0000	0.9990	0.9981	0.9971	0.9962	0.9952	0.9943	0.9933	0.9924	0.9915
80	0.9905	0.9896	0.9887	0.9877	0.9868	0.9859	0.9850	0.9841	0.9831	0.9822
90	0.9813	0.9804	0.9795	0.9786	0.9777	0.9768	0.9759	0.9750	0.9741	0.9732
	0.9723	0.9715	0.9706	0.9697	0.9688	0.9680	0.9671	0.9662	0.9653	0.9645

(Source: By permission from Johnson, T.W. and W.B. Berwald, *Flow of Natural Gas Through High Pressure Transmission Lines*, Monograph No. 6, US Dept. of Interior, Bureau of Mines, Washington, DC.)

$$\text{Note: Multiplier} = \left( \frac{460 + 60}{460 + \text{actual flowing temperature}} \right)^{1/2}$$

where

$d$  = pipe internal diameter, in.

$L$  = pipe length, miles

$P_1$  = upstream pressure, psia

$P_2$  = downstream pressure, psia

$S_g$  = gas specific gravity

$Z$  = gas compressibility factor

$q_g$  = gas flow rate, MMscfd (at 14.7 psia and 60° F)

$T$  = gas flowing temperature, °R = 460° F +  $t$

$E$  = efficiency factor for flow, use

In SI units,

Panhandle formula for natural gas pipe lines 150–600 mm diameter and  $Re = (5 \times 10^6)$  to  $(14 \times 10^6)$  is

$$q'_h = 5.06 \times 10^{-3} E \left[ \frac{(p'_1)^2 - (p'_2)^2}{L_m} \right]^{0.5394} d^{2.6182}, \text{ m}^3/\text{h} \quad (4-235)$$

or

$$(p'_1)^2 - (p'_2)^2 = L_m \left[ \frac{10^3 q'_h}{5.06 d^{2.6182} E} \right]^{1.8539} \quad (4-235a)$$

where

$d$  = internal pipe diameter, mm

$L_m$  = length of pipe, km

$p'_1$  = inlet pressure, bara

$p'_2$  = outlet pressure, bara

$q'_h$  = rate of flow in  $\text{m}^3/\text{h}$  at metric standard conditions (MSC)

1.103 bara and 15° C.

In practice, the Panhandle equation is commonly used for longer pipe with a large pipe size (greater than 10 in.) where the Reynolds number is on the straight line portion of the Moody diagram (Figure 4-5). Neither the Weymouth nor the Panhandle represents a "conservative" assumption. If the Weymouth formula is assumed, and the flow is at moderate Reynolds number, the friction factor will be higher than the horizontal portion of the Moody curve, and the actual pressure drop will be higher than calculated. If the Panhandle formula is used and the flow is in a high Reynolds number, the friction factor will be higher than assumed and the actual pressure drop will be higher than calculated.

For bends in pipe add to length [27]:

Bend Radius	Add*, as Pipe Diameter, $d_e$
1 pipe dia.	17.5
1.5 pipe dia.	10.4
2 pipe dia.	9
3 pipe dia.	8.2

\* These must be converted to the unit of length used in the formula.

If a line is made up of several different sizes, these may be resolved to one, and then the equation solved once for this total equivalent length. If these are handled on a per size basis, and totaled on the basis of the longest length of one size of line, then the equivalent length,  $L_e$ , for any size  $d$  referenced to a basic diameter,  $d_e$ , is

$$L_e = L_m (d_e/d)^{4.854} \quad (4-236)$$

where

$L_m$  is the length of pipe size  $d$  to be used.

$L_e$  is the equivalent length of pipe size  $d$ , length  $L_m$  after conversion to basis of reference diameter  $d_e$ .

The calculations can be based on diameter  $d_e$  and a length of all the various  $L_e$  values in the line plus the length of line of size  $d_e$ , giving a total equivalent length for the line system.

#### 4.57 MODIFIED PANHANDLE FLOW FORMULA [26]

$$q_{DS} = 737.2E \left( \frac{T_o}{P_o} \right)^{1.02} \left\{ \left[ \frac{P_1^2 (1 + 0.67ZP_1) - P_2^2 (1 + 0.67ZP_2)}{TL_m G^{0.961}} \right] \right\}^{0.51} d^{2.53} \quad (4-237)$$

where

$L_m$  = length, mi

$d$  = inside diameter, in.

$T$  = flowing temperature, °R

$Z$  = gas deviation, compressibility factor

$T_o$  = base temperature, 520° R

$G$  = gas specific gravity

$P$  = pressure, psia

$P_o$  = base pressure, 14.73 psia

$E$  = "Efficiency factor", which is really an adjustment to fit the data

$q_{DS}$  = flow rate, SCF/day.

#### EXAMPLE 4-17

##### Series System

Determine the equivalent length of a series of lines: 5 mi of 14 in. (13.25-in. ID) connected to 3 mi of 10 in. (10.136-in. ID) connected to 12 mi of 8 in. (7.981-in. ID).

Select 10 in. as the base reference size.

The five-mile section of 14-in. pipe is equivalent to

$$L_1 = 5 (10.136 / 13.25)^{5.33} = 1.199 \text{ mi of 10 in.}$$

The 12-mile section of 8 in. is equivalent to

$$L_1 = 12 (10.136 / 7.981)^{5.33} = 42.9 \text{ mi of 10-in.}$$

Total equivalent length of line to use in calculations is

$$1.199 + 3.0 + 42.9 = 47.099 \text{ mi of 10 in. (10.136-in. ID).}$$

An alternative procedure is to calculate (1) the pressure drop series-wise one section of the line at a time, or (2) capacity for a fixed inlet pressure, series-wise.

#### 4.58 AMERICAN GAS ASSOCIATION (AGA) DRY GAS METHOD

See Uhl et al. [44] AGA, Dry Gas Manual. Some tests indicate that this method is one of the most reliable above a fixed Reynolds number.

#### 4.59 COMPLEX PIPE SYSTEMS HANDLING NATURAL (OR SIMILAR) GAS

The method suggested in the Bureau of Mines Monograph No. 6 [42] has found wide usage and is outlined here using the Weymouth Formula as a base.

##### 1. Equivalent lengths of pipe for different diameters

$$L_1 = L_2 (d_1/d_2)^{16/3} \quad (4-238)$$

where  $L_1$  = the equivalent length of any pipe of length  $L_2$  and diameter  $d_2$ , in terms of diameter  $d_1$ .

$$d_1 = d_2 (L_1/L_2)^{3/16} \quad (4-239)$$

where  $d_1$  = the equivalent diameter of any pipe of a given diameter  $d_2$  and length  $L_2$ , in terms of any other length  $L_1$ .

##### 2. Equivalent diameters of pipe for parallel lines

$$d_o = \left( d_1^{8/3} + d_2^{8/3} + \dots + d_n^{8/3} \right)^{3/8} \quad (4-240)$$

where  $d_o$  is the diameter of a single line with the same delivery capacity as that of the individual parallel lines of diameters  $d_1$ ,  $d_2$ , ..., and  $d_n$ . Lines of same length.

This value of  $d_o$  may be used directly in the Weymouth formula.

#### EXAMPLE 4-18

##### Looped System

Determine the equivalent length of 25 mi of 10 in. (10.136-in. ID) pipe which has parallel loop of 6 mi of 8 in. (7.981-in. ID) pipe tied in near the mid-section of the 10-in. line.

Figure the looped section as parallel lines with 6 mi of 8 in. and 6 mi of 10 in. the equivalent diameter for one line with the same carrying capacity is

$$d_o = \left[ (7.981)^{8/3} + (10.136)^{8/3} \right]^{3/8} = 11.9 \text{ in.}$$

This simplifies the system to one section 6 mi long of 11.9 in. ID (equivalent) pipe, plus one section of 25 minus 6, or 19 mi of 10 in. (10.136-in. ID) pipe.

Now convert the 11.9-in. pipe to a length equivalent to the 10 in. diameter.

$$L_1 = 6 (10.136 / 11.9)^{5.33} = 2.53 \text{ mi}$$

Total length of 10-in. pipe to be used in calculating capacity is  $19 + 2.53 = 21.53$  mi. By the principles outlined in the examples, gas pipe line systems may be analyzed, paralleled, cross-tied, and so on.

#### 4.60 TWO-PHASE LIQUID AND GAS FLOW IN PROCESS PIPING

An understanding of two-phase flow is necessary for sound piping design. This is because almost all chemical process plants encounter two-phase flow conditions. The concurrent flow of liquid and gas in pipelines has received considerable study [45–48]. However, pressure drop prediction is not extremely reliable except for several gas pipeline conditions. The general determinations of pressure drop for plant process lines can only be approximated.

**EXAMPLE 4-19****Parallel System: Fraction Paralleled**

Determine the portion of a 30-mile, 18-in. (17.124-in. ID) line which must be paralleled with 20-in. (19.00-in. ID) pipe to raise the total system capacity 1.5 times the existing rate, keeping the system inlet and outlet conditions the same.

$$x = \frac{(q_{da}/q_{db})^2 - 1}{\left[ \frac{1}{[1 + (d_b/d_a)^{2.667}]^2} - 1 \right]} \quad (4-241)$$

For this example,  $q_{db} = 1.5 q_{da}$

$$x = \frac{(1/1.5)^2 - 1}{\left[ \frac{1}{[1 + (19.00/17.124)^{2.667}]^2} - 1 \right]} = 0.683$$

The latest two-phase flow research and design studies have broadened the interpretation of some of the earlier flow patterns and refined some design accuracy for selected situations. The method presented here serves as a fundamental reference source for further studies. It is suggested that the designer compares several design concept results and interprets which best encompasses the design problem under consideration. Some of the latest references are included in the Reference Section. However, no one reference has a solution to all two-phase flow problems.

If two-phase flow situations are not recognized, pressure drop problems may develop which can prevent systems from operating. It requires very little percentage of vapor, generally above 7–8% (by volume), to establish volumes and flow velocities that must be solved by two-phase flow analysis. The discharge flow through a pressure relief valve or a process reactor is often an important example where two-phase flow exists, and must be recognized for its backpressure impact.

Two-phase flow often presents design and operational problems not associated with liquid or gas flow. For example, several different flow patterns may exist along the pipeline. Frictional pressure losses are more difficult to estimate, and in the case of a cross-country pipeline, a terrain profile is necessary to predict pressure drops due to elevation changes. The downstream end of a pipeline often requires a separator to separate the liquid and gas phases, and a slug catcher may be required to remove liquid slugs.

Static pressure losses in gas–liquid flow differ from those in single-phase flow because an interface can be either smooth or rough, depending on the flow pattern. Two-phase pressure losses may be up to a factor of 10 higher than those in single-phase flow. In the former, the two phases tend to separate and the liquid lags behind. Most published correlations for two-phase pressure drop are empirical and, therefore, limited by the range of data for which they were derived [49–51].

## 4.61 FLOW PATTERNS

In determining the (type of flow) phase distribution in a process pipeline, designers refer to a diagram similar to Figures 4-48a and b which is known as the Baker map. Figure 4-49 shows the types of flow regimes that can exist in a horizontal pipe, and Table 4-35 lists the characteristic linear velocities of the gas and liquid phases in each flow regime. Seven types of flow patterns are considered in

This means 68.3% of the 30 mi must be parallel with the new 19-in. ID pipe.

**Parallel System: New Capacity after Paralleling**

Solve this relation, rearranged conveniently to [42]

$$q_{db} = \frac{q_{da}}{\left\{ x \left[ \frac{1}{[1 + (d_b/d_a)^{2.667}]^2} - 1 \right] + 1 \right\}^{1/2}} \quad (4-242)$$

evaluating two-phase flow, and only one type can exist in a pipeline at a time. But as conditions change (e.g., velocity, roughness, and elevation), the type of flow pattern may also change. The pressure drop can also vary significantly between the flow regimes. The seven types of flow regimes in order of increasing gas rate at a constant liquid flow rate are given below.

*Bubble or Froth flow:* Bubbles of gas are dispersed throughout the liquid, and are characterized by bubbles of gas moving along the upper part of the pipe at approximately the same velocity as the liquid.

This type of flow can be expected when the gas content is less than about 30% of the total (weight) volume flow rate. (Note: 30% gas by weight is over 99.9% by volume, normally.)

*Plug flow:* Alternate plugs of liquid and gas move along the upper part of the pipe and liquid moves along the bottom of the pipe.

*Stratified flow:* The liquid phase flows along the bottom of the pipe while the gas flows over a smooth liquid–gas interface.

*Wave flow:* Wave flow is similar to stratified flow except that the gas is moving at a higher velocity and the gas–liquid interface is distributed by waves moving in the direction of flow.

*Slug flow:* This pattern occurs when waves are picked up periodically by the more rapidly moving gas. These form frothy slugs that move along the pipeline at a much higher velocity than the average liquid velocity. This type of flow causes severe and, in most cases, dangerous vibrations in equipment because of the high velocity slugs against fittings.

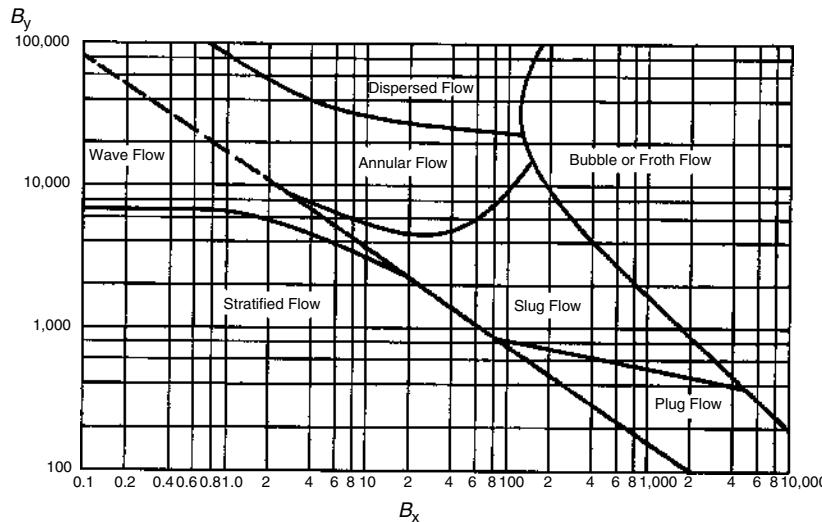
*Annular flow:* In annular flow, liquid forms around the inside wall of the pipe and gas flows at a high velocity through the central core.

*Dispersed, Spray, or Mist flow:* Here, all of the liquid is entrained as fine droplets by the gas phase. This type of flow can be expected when the gas content is more than roughly 30% of the total weight flow rate. Some overhead condenser and reboiler-return lines have dispersed flow.

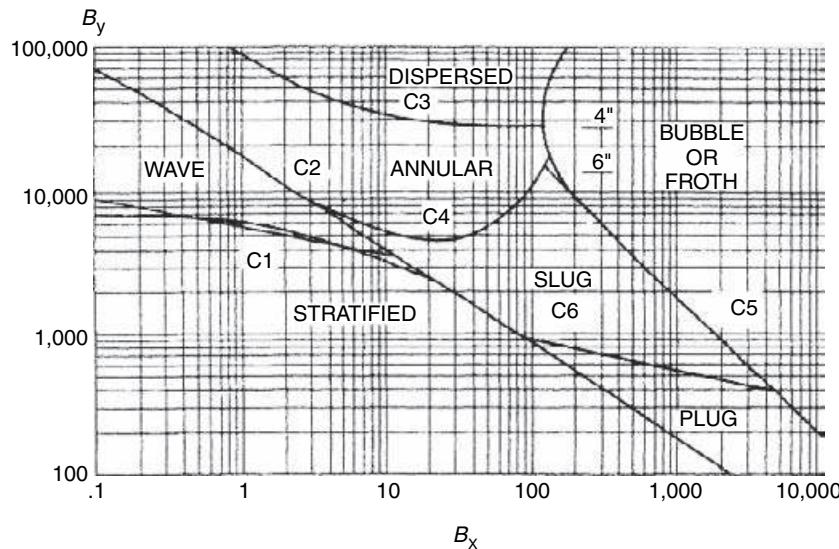
## 4.62 FLOW REGIMES

Establishing the flow regime involves determining the Baker parameters  $B_x$  and  $B_y$ , from the two-phase system's characteristics and physical properties. The Baker parameters can be expressed as:

$$B_x = 531 \left( \frac{W_L}{W_G} \right) \left\{ \frac{(\rho_L \rho_G)^{0.5}}{\rho_L^{2/3}} \right\} \left( \frac{\mu_L^{1/3}}{\sigma_L} \right) \quad (4-243)$$



**Figure 4-48a** Flow patterns for horizontal two-phase flow. (Based on data from 1-, 2-, and 4-in. pipe). (By permission from Baker O., *Oil Gas J.*, Nov 10, 1958, p. 156.)



**Figure 4-48b** Baker parameters for horizontal two-phase flow regimes with modified boundaries. (Based on data from 1-, 2-, and 4-in. pipe.)

$$B_y = 2.16 \left( \frac{W_G}{A} \right) \frac{1}{(\rho_L \rho_G)^{0.5}} \quad (4-244)$$

where

$A$  = Internal pipe cross-sectional area,  $\text{ft}^2$

$d$  = Internal pipe diameter, in.

$D$  = Internal pipe diameter, ft

$W_G$  = Gas flow rate,  $\text{lb}/\text{h}$

$W_L$  = Liquid flow rate,  $\text{lb}/\text{h}$

$\rho_L$  = Liquid density,  $\text{lb}/\text{ft}^3$

$\rho_G$  = Gas density,  $\text{lb}/\text{ft}^3$

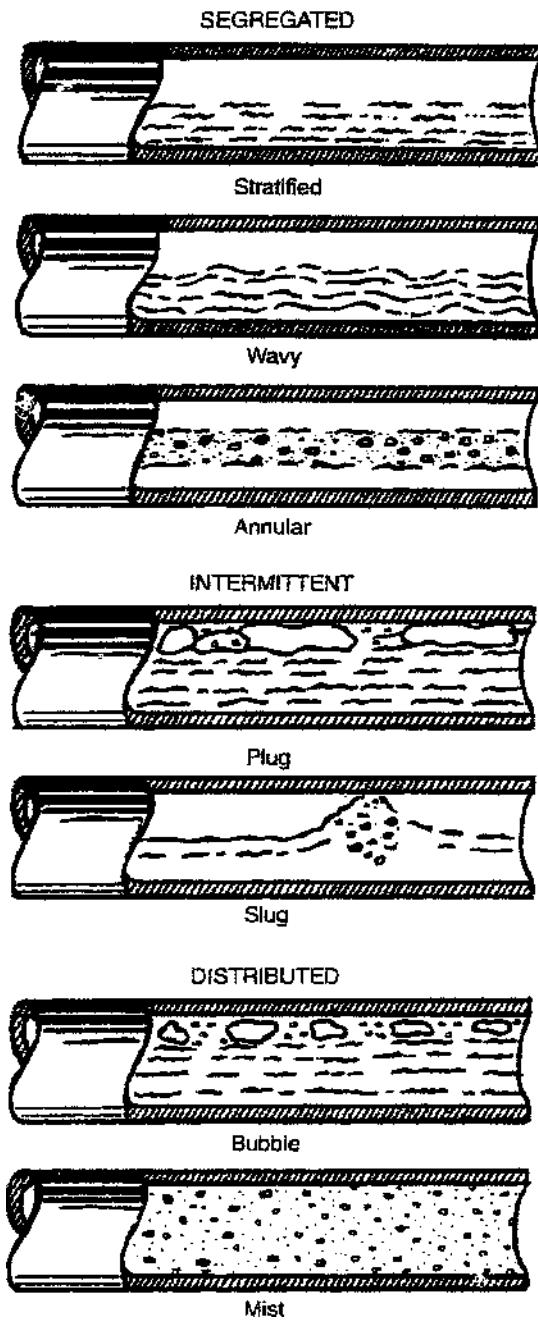
$\mu_L$  = Liquid viscosity,  $\text{cP}$

$\mu_G$  = Gas viscosity,  $\text{cP}$

$\sigma_L$  = Liquid surface tension,  $\text{dyn}/\text{cm}$ .

$B_x$  depends on the mass flow ratio and the physical properties of the liquid and gas phases. It is independent of pipe size; therefore, it remains constant once calculated from the characteristics of the liquid and gas, and its position on the Baker map changes only if the liquid-gas proportion changes.  $B_y$  depends on the gas-phase flow rate, the gas and liquid densities, and pipe size. The practical significance of pipe size is that by changing pipe diameters, the type of flow might also be changed, which in turn changes friction losses in the pipe. The intersection of  $B_x$  and  $B_y$  in Figure 4-48a determines the flow regime for the calculated liquid-gas ratio and the liquid's and gas's physical properties. With increasing gas content  $B_x$ ,  $B_x$  moves up and to the left on the map.

The boundaries of the various flow pattern regions depend on the mass velocity of the gas phase. These boundaries are represented by analytical equations developed by Coker [52]. These equations



**Figure 4-49** Representatives forms of horizontal tow-phase flow patterns, same as indicated in Figure 4.48a. (By permission from Hein, H., *Oil Gas J.*, Aug 2, 1982, p. 132.)

are used as the basis for determining the prevailing regime for any given flow rates and physical properties of the liquid and gas. The mathematical models representing the boundaries of the flow regimes are (Figure 4-48b):

$$C_1 : \ln B_y = 9.774459 - 0.6548 (\ln B_x) \quad (4-245)$$

$$C_2 : \ln B_y = 8.67694 - 0.1901 (\ln B_x) \quad (4-246)$$

**TABLE 4-35 Characteristics Linear Velocities of Two-Phase Flow Regimes**

Regime	Liquid Phase (ft/s)	Vapor Phase (ft/s)
Bubble or froth	5–15	0.5–2
Plug	2	< 4
Stratified	< 0.5	0.5–10
Wave	< 1.0	> 15
Slug	15 (but less than vapor velocity)	15–50
Annular	< 0.5	> 20
Dispersed, Spray, or Mist	Close to vapor velocity	> 200

$$C_3 : \ln B_y = 11.3976 - 0.6084 (\ln B_x) + 0.0779 (\ln B_x)^2 \quad (4-247)$$

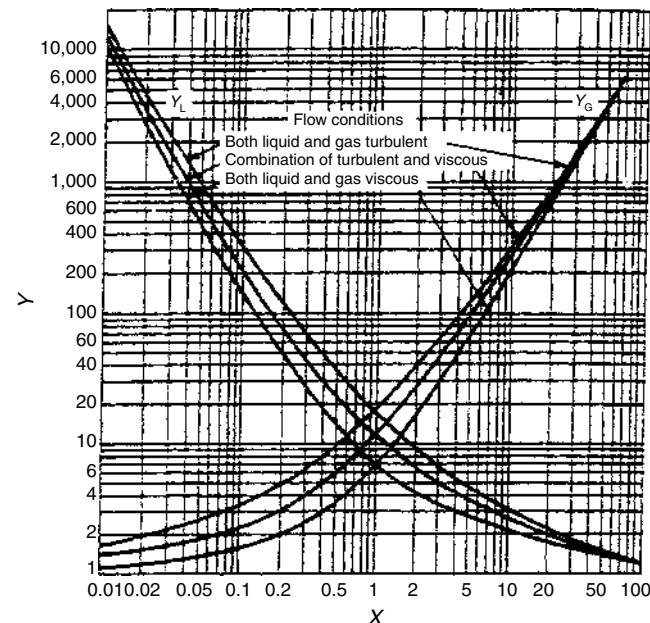
$$C_4 : \ln B_y = 10.7448 - 1.6265 (\ln B_x) + 0.2839 (\ln B_x)^2 \quad (4-248)$$

$$C_5 : \ln B_y = 14.569802 - 1.0173 (\ln B_x) \quad (4-249)$$

$$C_6 : \ln B_y = 7.8206 - 0.2189 (\ln B_x) \quad (4-250)$$

#### 4.63 PRESSURE DROP

Various studies have been conducted in predicting the two-phase frictional pressure losses in pipes. The Lockhart–Martinelli correlations [53] shown in Figure 4-50 are employed. The basis of the correlations is that the two-phase pressure drop is equal to the single-phase pressure drop of either phase multiplied by a factor that is derived from the single-phase pressure drops of the two phases. The total pressure drop is based on the gas phase pressure



**Figure 4-50** Lockhart–Martinelli pressure drop correlation. (Source: Lockhart and Martinelli [53].)

drop. The pressure drop computation is based on the following assumptions:

1. The two-phase flow is isothermal and turbulent in both liquid and gas phases.
2. The pressure loss is less than 10% of the absolute upstream pressure.

The two-phase pressure drop can be expressed as:

$$\Delta P_{TP} = \Delta P_G \Phi^2 \quad (4-251)$$

where  $\Delta P_{TP}$  = two-phase flow pressure drop and  $\Phi$  ( $Y_G$ ) is the two-phase flow modulus.  $\Phi$  is a function of the Lockhart–Martinelli two-phase flow modulus  $X$ , which is defined as:

$$X = \left[ \frac{(\Delta P / \Delta L)_L}{(\Delta P / \Delta L)_G} \right]^{0.5} \quad (4-252)$$

where

$$\Phi = f(X) \quad (4-253)$$

and

$(\Delta P / \Delta L)_L$  = pressure drop of liquid being assumed to flow alone in the pipe (actual liquid mass flow through total pipe cross section area)

$(\Delta P / \Delta L)_G$  = pressure drop of gas being assumed to flow alone in the pipe (actual gas mass flow through total pipe cross section area).

An alternative form of calculating the Martinelli modulus by using the Darcy relationship is

$$X = \sqrt{\left( \frac{W_L}{W_G} \right)} \left[ \left( \frac{\rho_G}{\rho_L} \right) \left( \frac{f_{D,L}}{f_{D,G}} \right) \right]^{0.5} \quad (4-254)$$

The Martinelli modulus is then used to calculate the two-phase flow modulus  $\Phi$  from a relationship taking the general form

$$\Phi = aX^b \quad (4-255)$$

where  $a$  and  $b$  are empirical constants for the specific flow regime. Equation (4-255) for a specific flow regime is correlated as follows.

#### BUBBLE OR FROTH FLOW

$$\Phi = \left[ \frac{14.2 X^{0.75}}{(W_L/A)^{0.1}} \right] \quad (4-256)$$

#### PLUG FLOW

Plug flow develops when the interface is high in the pipe's cross section. The waves congregate and touch the high point of the pipe wall. Here, plugs of liquid alternating with pockets of gas in the liquid close to the upper wall move along the horizontal line.

Gravity flow lines have such types of flow. The initial, minimum pipe sizes can be selected with velocities typical of those of pump suction lines.

$$\Phi = \left[ \frac{27.315 X^{0.855}}{(W_L/A)^{0.17}} \right] \quad (4-257)$$

#### STRATIFIED FLOW

$$\Phi = \left[ \frac{15,400 X}{(W_L/A)^{0.8}} \right] \quad (4-258)$$

#### WAVE FLOW

Wave flow is similar to stratified flow except the interface has waves moving in the direction of flow. The Schneider–White–Huntington [54] correlation  $H_x$  is used to determine the two-phase pressure loss:

$$H_x = \left( \frac{W_L}{W_G} \right) \left( \frac{\mu_L}{\mu_G} \right) \quad (4-259)$$

The Huntington friction factor  $f_H$  is

$$\ln f_H = 0.211 (\ln H_x) - 3.993 \quad (4-260)$$

$$\frac{\Delta P_{TP}}{L (\text{ft})} = \frac{0.00000336 (f_H) (W_G)^2}{d^5 \rho_G} \quad (4-261)$$

#### SLUG FLOW

$$\Phi = \left[ \frac{1190 X^{0.815}}{(W_L/A)^{0.5}} \right] \quad (4-262)$$

#### ANNULAR FLOW

$$\Phi = (aX^b) \quad (4-263)$$

where

$$a = 4.8 - 0.3125d \quad (4-264)$$

$$b = 0.343 - 0.021d \quad (4-265)$$

$d$  = Pipe inside diameter, in.

$d$  = 10 for 12 in. and larger sizes.

Once the two-phase flow modulus ( $\Phi$ ) for the particular flow regime has been calculated, it can be used to determine the two-phase pressure drop by

$$\Delta P_{100,TP} = \Delta P_{100,G} \Phi^2$$

so that

$$\Delta P_{TP} = \Delta P_G \Phi^2 \quad (4-266)$$

where  $\Delta P_{100,TP}$  = two-phase flow pressure drop per 100 ft of pipe, psi/100 ft.

#### DISPERSED OR SPRAY FLOW

This type of flow can be expected when the gas content is more than roughly 30% of the total volume flow rate. Some overhead condenser and reboiler return lines have dispersed flow.

$$\Phi = \exp[C_0 + C_1 (\ln X) + C_2 (\ln X)^2 + C_3 (\ln X)^3] \quad (4-267)$$

**TABLE 4-36 Two-Phase Flow Moduli as a Function of Martinelli Moduli X**

X	$\Phi_{L,tt}$	$\Phi_{G,tt}$	$\Phi_{L,vt}$	$\Phi_{G,vt}$	$\Phi_{L,tv}$	$\Phi_{G,tv}$	$\Phi_{L,vv}$	$\Phi_{G,vv}$
0.01	(128)	(1.28)	(120)	(1.20)	(112)	(1.12)	(105)	(1.05)
0.02	(64.8)	(1.37)	(64)	(1.28)	(58)	(1.16)	(53.5)	(1.02)
0.04	38.5	1.54	(34)	(1.36)	(31)	(1.24)	(28.0)	(1.12)
0.07	24.4	1.71	20.7	1.45	(19.3)	(1.35)	(17.0)	(1.19)
0.10	18.5	1.85	15.2	1.52	(14.5)	(1.45)	(12.4)	(1.24)
0.20	11.2	2.23	8.9	1.78	(8.7)	(1.74)	(7.0)	(1.40)
0.4	7.05	2.83	5.62	2.25	(5.5)	(2.20)	4.25	1.7
0.7	5.04	3.53	4.07	2.85	(4.07)	(2.85)	3.08	2.16
1.0	4.20	4.20	3.48	3.48	(3.48)	(3.48)	2.61	2.61
2.0	3.1	6.20	2.62	5.25	(2.62)	(5.24)	2.06	4.12
4.0	2.38	9.5	2.05	8.20	(2.15)	(8.60)	1.76	7.0
7.0	1.96	13.7	1.73	12.0	(1.83)	(12.8)	1.60	11.2
10.0	1.75	17.5	1.59	15.9	(1.66)	(16.6)	1.50	15.0
20.0	1.48	29.5	(1.40)	28.0	(1.44)	(28.8)	1.36	27.3
40.0	1.29	51.5	(1.25)	50.0	(1.25)	(50.0)	1.25	50.0
70.0	1.17	82.0	(1.17)	82.0	(1.17)	(82.0)	(1.17)	(82.0)
100.0	1.11	111.0	(1.11)	(111.0)	(1.11)	(111.0)	(1.11)	(111.0)

(Source: H.J. Sandler and E.T. Luckiewicz, *Practical Process Engineering – A working Approach to Plant Design*, McGraw-Hill, Inc., 1987.)

Note: Parentheses indicate area of limited data.

where

$$C_0 = 1.4659$$

$$C_1 = 0.49138$$

$$C_2 = 0.04887$$

$$C_3 = -0.000349.$$

The Lockhart–Martinelli correlation for dispersed flow requires the calculation of a two-phase flow modulus  $\Phi$  to be used in conjunction with either the liquid-phase pressure drop or the gas-phase pressure drop to calculate the two-phase pressure drop in a pipe; that is,

$$\Delta P_{100,TP} = \Delta P_{100,L} \Phi_L \quad (4-268)$$

$$\Delta P_{100,TP} = \Delta P_{100,G} \Phi_G^2 \quad (4-269)$$

The two-phase flow modulus is also dependent on whether each phase is laminar (viscous) or turbulent\*. Table 4-36 shows the modulus symbol used in relation to the phase type and flow characteristics.

The flow regime of each phase is determined along with their individual pressure drops. The  $X$  value and flow regime are used for determining the Martinelli two-phase flow modulus ( $\Phi$ ). Table 4-37 shows the correlation between  $X$  and the various possible combinations of laminar and turbulent flow in the gas and liquid phases.

The frictional pressure drop of liquid or gas flowing alone in a straight pipe can be expressed as:

$$\frac{\Delta P}{d(\text{ft})} = \frac{0.00000336 f_D W_x^2}{d^5 \rho_x}, \text{psi/L (ft)} \quad (4-270)$$

Equation (4-270) can be modified for calculating the overall frictional pressure drop of the two-phase flow for the total length of pipe plus fittings,  $L_{\text{Total}}$  (i.e.,  $L_{\text{Total}} = L_{\text{st}} + L_{\text{eq}}$ ) based on the gas-phase pressure drop.

$$\Delta P_{\text{Total}} = \frac{0.00000336(f_D)(W_G^2)(\Phi^2)(L_{\text{Total}})}{d^5 \rho_G}, \text{psi} \quad (4-271)$$

\* The Martinelli correlation considers transitional flow as turbulent flow.

**TABLE 4-37 Martinelli Symbols Used for Two-Phase Modulus**

Symbol	Flow Characteristics	
	Liquid Phase	Gas Phase
<i>Liquid-phase moduli</i>		
$\Phi$	Turbulent	Turbulent
$\Phi_{L,tt}$	Turbulent	Viscous
$\Phi_{L,vt}$	Viscous	Turbulent
$\Phi_{L,vv}$	Viscous	Viscous
<i>Gas-phase moduli</i>		
$\Phi_{G,tt}$	Turbulent	Turbulent
$\Phi_{G,tv}$	Turbulent	Viscous
$\Phi_{G,vt}$	Viscous	Turbulent
$\Phi_{G,vv}$	Viscous	Viscous

(Source: H.J. Sandler and E.T. Luckiewicz, *Practical Process Engineering – A working Approach to Plant Design*, McGraw-Hill, Inc., 1987.)

where

$$d = \text{internal pipe diameter, in.}$$

$$f_D = \text{Darcy friction factor}$$

$$L_{\text{Total}} = \text{total length of pipe, ft}$$

$$L_{\text{st}} = \text{length of straight pipe, ft}$$

$$L_{\text{eq}} = \text{equivalent length of pipe, ft}$$

$$W_G = \text{gas mass flow rate, lb/h}$$

$$\rho_G = \text{gas density, lb/ft}^3$$

$$\Phi = \text{the Martinelli two-phase flow modulus.}$$

The velocity of the two-phase fluid is

$$v = \frac{0.0509}{d^2} \left[ \frac{W_G}{\rho_G} + \frac{W_L}{\rho_L} \right] \quad (4-272)$$

#### 4.64 EROSION-CORROSION

Depending on the flow regime, the liquid in a two-phase flow system can be accelerated to velocities approaching or exceeding

the gas velocity. In some cases, these velocities are higher than what would be desirable for process piping. Such high velocities lead to a phenomenon known as erosion-corrosion, where the corrosion rate of a material is accelerated by an erosive material or force (in this case, the high-velocity liquid). It also depends on the chemical nature of gas and liquid, presence of oxygen, and so on.

An index [55] based on velocity head can indicate whether erosion-corrosion may become significant at a particular velocity. It can be used to determine the range of mixture densities and velocities below which erosion-corrosion should not occur.

This index is

$$\rho_M U_M^2 \leq 10,000 \quad (4-273)$$

In SI units,

$$\rho_M U_M^2 \leq 15,000 \quad (4-274)$$

where the mixture density is

$$\rho_M = \frac{W_L + W_G}{\left( \frac{W_L}{\rho_L} + \frac{W_G}{\rho_G} \right)} \quad (4-275)$$

and the mixture velocity is

$$U_M = U_G + U_L \quad (4-276)$$

$$= \left( \frac{W_G}{3600\rho_G A} + \frac{W_L}{3600\rho_L A} \right) \quad (4-277)$$

where

- A = internal pipe cross-sectional area, ft<sup>2</sup> (m<sup>2</sup>)
- $U_G$  = superficial linear gas velocity, ft/s (m/s)
- $U_L$  = superficial linear liquid velocity, ft/s (m/s)
- $W_G$  = gas mass flow rate, lb/h (kg/h)
- $W_L$  = liquid mass flow rate, lb/h (kg/h)
- $\rho_G$  = gas density, lb/ft<sup>3</sup> (kg/m<sup>3</sup>)
- $\rho_L$  = liquid density, lb/ft<sup>3</sup> (kg/m<sup>3</sup>).

## AVOIDING SLUG FLOW

Typically, this phenomenon may occur in a pocketed line between an overhead condenser at grade and an elevated reflux drum. At low velocities, liquid collects at the low point and periodically can become a slug in the line. With a sufficiently high velocity, the liquid phase can be carried through without developing a slug flow.

Slug flow can be avoided in process piping by [56]:

- Using a low point effluent drain or bypass.
- Reducing line sizes to the minimum permitted by available pressure drops.
- Arranging the pipe configuration to protect against slug flow. For example, in a pocketed line where liquid can collect, slug flow might develop.
- Designing for parallel pipe runs that will increase flow capacity without increasing overall friction loss.
- Using valved auxiliary pipe runs to regulate alternative flow rates and thus avoid slug flow.

Slug flow will not occur in a gravity flow line, or a hard tee connection (e.g., flow through the branch) at a low point can provide sufficient turbulence for more effective liquid carryover.

A diameter adjustment, coupled with gas injections, can also alter a slug flow pattern to bubble or dispersed flow.

The following are ways to prevent two-phase problems [57]:

- Piping should slope away from the liquid source of supply.
- Install low-point drains or traps before any riser, at the end of mains; before any pressure control valves; and at every 300–500 ft.
- Mount Y-strainers and basket strainers horizontally, to prevent the buildup of liquid in them that could then be picked up in the vapor stream.
- Avoid vertical lifts in the hot-condensate return lines.
- Design and install equipment to allow complete draining, and eliminate formation of pockets in the piping that can trap liquid.
- Include traps, pressure control valves, safety valves, and rupture disks in your preventive maintenance program.
- Conduct a review of abandoned in-place equipment and piping. These can be a major source of liquid buildup.
- Be sure that materials of construction for equipment and piping should have a reasonable tensile strength to resist dynamic transients. Do not use cast iron.
- Check insulation to minimize condensate formation.
- Check design of supports for transient performance.
- When there is condensable gas, use old-fashioned precautions. Open low-point drain valves and initiate the flow slowly at reduced pressure.

*Note:* Most of these apply to predominately gas flow, with a low volume fraction of liquid.

**Maintaining the Proper Flow Regime.** When designing piping systems for two-phase flow, it is best to avoid slug flow. Water hammer is caused by the slug of liquid impinging on pipe and equipment walls at every change of flow direction. If slug flow is not eliminated, there may be equipment damage due to erosion-corrosion and/or overstressing of joints due to cyclic loading.

If slug flow enters a distillation column, the alternating composition and density of the gas and liquid slugs cause cycling of composition and pressure gradients along the length of the column. This cycling causes problems with product quality and process control. The problem of pressure gradients is particularly disturbing in vacuum columns.

In addition to slug flow, which is undesirable, dispersed flow may pose problems. Dispersed flow is nearly a homogeneous mixture of the liquid phase in the gas phase, which makes its behavior very similar to that of a compressible fluid. While dispersed flow is an ideal flow regime in piping systems, it causes phase-disengagement problems in flash tanks and distillation columns.

## 4.65 TOTAL SYSTEM PRESSURE DROP

The pressure drop for a system of horizontal and vertical (or inclined) pipe is the sum of the horizontal pressure drop plus the additional drop attributed to each vertical rise, regardless of initial and final elevations of the line [45].

$$\Delta P_{TPh} = \Delta P_{PT}(\text{horizontal pipe}) + \frac{nhF_e\rho_L}{144} \quad (4-278)$$

where

$n$  = number of vertical rises (or hills) in two-phase pipe line flow

$h$  = average height of all vertical rises (or hills) in two-phase pipe line, ft

$F_e$  = elevation factor using gas velocity,  $v$

$\rho_L$  = density of liquid, lb/ft<sup>3</sup>.

$$F_e = 0.00967 \frac{W_m^{0.5}}{v^{0.7}}, \text{ for } v > 10 \quad (4-279)$$

where

$W_m$  = Mass flow rate of liquid phase, lb/h ft<sup>2</sup> (of total pipe cross section area).

$v$  = Flow velocity (mean) or superficial velocity in pipe lines at flowing conditions for entire pipe cross section, ft/s. or as an alternative:

$$F_e = 1.7156v_G^{-0.702} \quad (4-280)$$

where  $v_G$  = gas velocity, ft/s.

A. To determine most probable type of two-phase flow using Figure 4-48a.

1. Calculate  $B_x$

2. Calculate  $B_y$

3. Read intersection of ordinate and abscissa to identify probable type of flow. Since this is not an exact, clear-cut position, it is recommended that the adjacent flow types be recorded.

B. Calculate the separate liquid and gas flow pressure drops.

1. For general process application both  $\Delta P_L$  and  $\Delta P_G$  may be calculated by the general flow equation:

$$\Delta P_L \text{ or } \Delta P_G = \frac{3.36 f_D L W^2 (10^{-6})}{d^5 \rho} \quad (4-281)$$

where  $f_D$  is the Darcy friction factor obtained from (Reynolds) Moody-Friction Factor chart (Figure 4-5) for an assumed line size,  $d$ , or from Chen's explicit equation for friction factor ( $f_D = 4f_C$ ).

2. For gas transmission, in general form [45]

$$\Delta P_G = \frac{(q_{d14.65}) L S_g T Z f}{20,000 d^5 P_{avg}} \quad (4-282)$$

where  $q_{d14.65}$  is the thousands of standard cubic feet of gas per day, measured at 60° F and 14.65 psia, and  $P_{avg}$  is the average absolute pressure (psia) in the pipe system between inlet and outlet. This is an estimated value and may require correction and recalculation of the final pressure drop if it is very far off.

where

$d$  = internal pipe diameter, in.

$f$  = Friction factor, Moody

$L$  = Pipe length, ft

$S_g$  = specific gravity of gas relative to air (= ratio of molecular weight gas/ 29)

$T$  = Absolute temperature, °R = 460 + °F

$Z$  = compressibility factor.

For oil flow in natural gas transmission lines [45]

$$\Delta P_L = \frac{f L Q_b^2 \rho}{181,916 d^5} \quad (4-283)$$

where

$Q_b$  = Flow rate in bbl/day

$L$  = Pipe length, ft

$\rho$  = Fluid density, lb/ft<sup>3</sup>.

3. Calculate

$$X = (\Delta P_L / \Delta P_G)^{1/2} \quad (4-284)$$

4. Calculate  $\Phi$  for types of flow selected from Figure 4-48a and as summarized below [45].

Type Flow	Equation for $\Phi_{GTT}$
Froth or bubble	$\Phi = 14.2X^{0.75} / (W_L/A)^{0.1}$
Plug	$\Phi = 27.315X^{0.855} / (W_L/A)^{0.17}$
Stratified	$\Phi = 15,400X / (W_L/A)^{0.8}$
Slug	$\Phi = 1,190X^{0.185} / (W_L/A)^{0.5}$
Annular*	$\Phi = (4.8 - 0.3125d)X^{0.343 - 0.021d}$

\* set  $d = 10$  for any pipe larger than 10 in.

$$X = [\Delta P_L / \Delta P_G]^{1/2}$$

5. Calculate two-phase pressure drop, horizontal portions of lines. For all types of flow, except wave and fog or spray:

$$\Delta P_{TP} = \Delta P_G \Phi_{GTT}^2, \text{ psi/ft} \quad (4-285)$$

For wave [54],

$$\Delta P_{TP} = f_{TP} (G'_G)^2 / 193.2 d \rho_G, \text{ psi/ft} \quad (4-286)$$

where

$$f_{TP} = 0.0043 \left( \frac{W_m \mu_L}{G \mu_G} \right)^{0.214} \quad (4-287)$$

where

$f_{TP}$  = Two-phase friction for wave flow

$G'_G$  = Mass rate, lb/s (ft<sup>2</sup> cross section)

$G$  = Mass flow rate of gas phase, lb/h ft<sup>2</sup>

$W_m$  = Mass flow rate of liquid phase, lb/h ft<sup>2</sup>

$\mu_L$  = Liquid viscosity, cP

$\mu_G$  = Gas viscosity, cP.

6. Total two-phase pressure drop, including horizontal and vertical sections of line. Use calculated value times 1.1–2.0, depending upon critical nature of application.

$$\Delta P_{TPh} = \Delta P_{TP} L + \frac{n h F_e \rho_L}{144} \quad (4-288)$$

where  $\rho_L$  is the density, lb/ft<sup>3</sup>, of the liquid flowing in the line, and  $F_e$  the elevation factor using gas velocity,  $v_G$ .

Use Figure 4-51 for  $v$  less than 10. Most gas transmission lines flow at 1–15 ft/s

For fog- or spray-type flow, Baker [45] suggests using Martinelli's correlation and multiplying results by two [53].

1. For gas pipeline flow, the values of ( $\Phi_{GTT}$ ) may be converted to "efficiency  $E$ " values and used to calculate the flow for the horizontal portion using a fixed allowable pressure drop in the general flow equation [45]. The effect of the vertical component must be added to establish the total pressure drop for the pumping system.

$$q_{d14.65} = \left[ \frac{38.7744 T_s (P_1^2 - P_2^2) d^5}{1000 P_s L_m S_g TZ} \right]^{0.5} \left( \frac{E}{f_g} \right)^{0.5} \quad (4-289)$$

where (14.65 refers to reference pressure  $P_s$ )

$d$  = internal pipe diameter, in.

$E$  = Gas transmission "efficiency" factor, varies with line size and surface internal condition of pipe

$f_g$  = Moody or "regular" Fanning friction for gas flow

$L_m$  = Pipe length, mi

$P_1$  = initial pressure, psi

$P_2$  = final pressure, psi

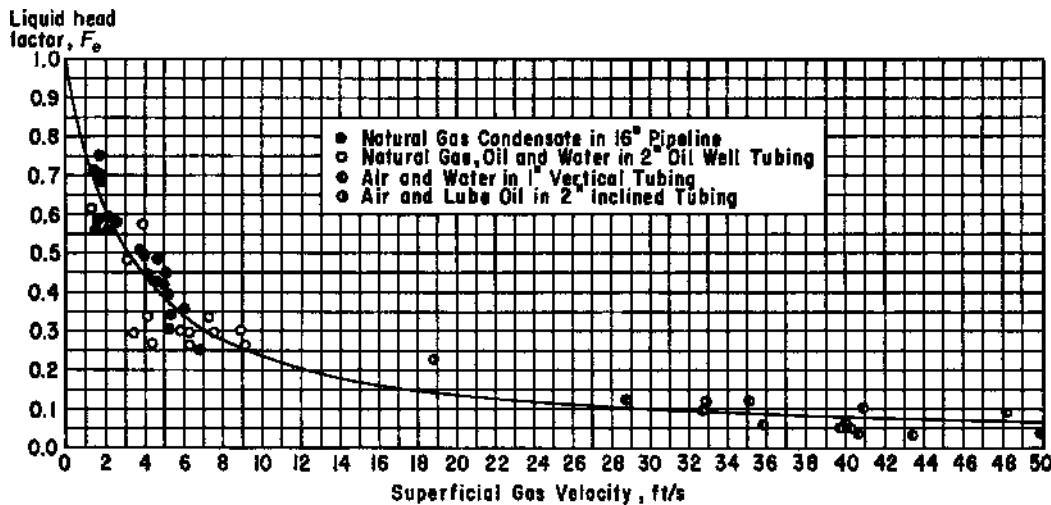
$P_s$  = standard pressure for gas transmission, psia

$S_g$  = specific gravity of gas relative to air (= ratio of molecular weight gas/ 29)

$T$  = Absolute temperature, °R = 460 + °F

$T_s$  = Standard temperature for gas measurement, °R = 460 + °F

$Z$  = Gas compressibility factor.



**Figure 4-51** Estimating pressure drop in uphill sections of pipeline for two-phase flow. (By permission from O. Flanigan, *Oil Gas J.*, Mar 10, 1958, p. 132.)

#### EXAMPLE 4-20 Two-phase Flow

A liquid-gas mixture is to flow in a line having 358 ft of level pipe and three vertical rises of 10 ft each plus one vertical rise of 50 ft. Evaluate the type of flow and expected pressure drop.

	Liquid	Gas
Flow $W$ , lb/h	1000	3000
Density $\rho$ , lb/ft <sup>3</sup>	63.0	0.077
Viscosity $\mu$ , cP	1.0	0.00127
Surface tension $\sigma$ , dyn/cm	15	

Pipe Schedule is 40 stainless steel.

Use maximum allowable gas velocity = 15,000 ft/min

*Solution*

#### 1. Determine probable types of flow:

$$B_x = 531 \left( \frac{W_L}{W_G} \right) \left\{ \frac{(\rho_L \rho_G)^{0.5}}{\rho_L^{2/3}} \right\} \left\{ \frac{\mu_L^{1/3}}{\sigma_L} \right\}$$

$$B_x = 531 \left( \frac{1000}{3000} \right) \left\{ \frac{[(63)(0.077)]^{0.5}}{(63)^{2/3}} \right\} \left\{ \frac{1}{15} \right\}$$

$$= 1.64$$

$$B_y = 2.16 \left( \frac{W_G}{A} \right) \frac{1}{(\rho_L \rho_G)^{0.5}}$$

Try 3-in. Sch. 40 pipe (ID = 3.068 in)

$$\text{Pipe cross-sectional area, } A = \pi D^2 / 4 = \pi (0.2556)^2 / 4 \\ = 0.0513 \text{ ft}^2$$

$$(W_L/A) = 1000 / 0.0513 = 19,493 \text{ lb/ft}^2 \text{ h}$$

$$(W_G/A) = 3000 / 0.0513 = 58,480 \text{ lb/ft}^2 \text{ h}$$

$$B_y = 2.16 \left( \frac{W_G}{A} \right) \frac{1}{(\rho_L \rho_G)^{0.5}}$$

$$B_y = 2.16 (58,480) \frac{1}{\sqrt{(63)(0.077)}} \\ = 57,352$$

Reading Figure 4-48a, the flow pattern type is probably annular, but could be wave or dispersed, depending on many undefined and unknown conditions.

#### 2. Liquid pressure drop:

$$\Delta P_L = 3.36 f_D L W^2 (10^{-6}) / d^5 \rho \quad (4-281)$$

Determine  $Re$  for 3-in. pipe:

From Figure 4-13;  $\varepsilon/d = 0.00058$  for steel pipe

The liquid velocity,  $v_L$ , is:

$$v_L = \left( \frac{W_L}{\rho_L A} \right) = \frac{1000}{(63)(3600)(0.0513)} = 0.086 \text{ ft/s}$$

$$\mu_e = 1 \text{ cP}/1488 = 0.000672 \text{ lb/ft s}$$

$$D = 3.068/12 = 0.2557 \text{ ft}$$

$$\rho_L = 63.0 \text{ lb/ft}^3$$

$$Re = D v_L \rho_L / \mu_e = 0.2557 (0.086) (63.0)/0.000672$$

$$Re = 2061 \text{ (this is borderline, and in critical region).}$$

Relative pipe roughness is

$$\frac{\varepsilon}{D} = \frac{0.00015}{0.25567} = 0.000587$$

(continued)

**EXAMPLE 4-20—(continued)**  
then

$$\begin{aligned} A &= \frac{\varepsilon/D}{3.7} + \left( \frac{6.7}{Re} \right)^{0.9} \\ &= \frac{0.000587}{3.7} + \left( \frac{6.7}{2060} \right)^{0.9} \\ &= 5.926 \times 10^{-3} \end{aligned}$$

Friction factor can be calculated by substituting the values of  $\varepsilon/D$  and  $A$  in Eq. (4-35)

$$\begin{aligned} \frac{1}{\sqrt{f_c}} &= -4 \log \left[ \frac{0.000587}{3.7} - \frac{5.02}{2060} \log (5.926 \times 10^{-3}) \right] \\ &= 9.01154 \\ f_c &= 0.01231 \end{aligned}$$

The Darcy friction factor,  $f_D = 4f_c$

$$\begin{aligned} f_D &= 4 (0.01231) \\ &= 0.0492 \end{aligned}$$

The liquid pressure drop,  $\Delta P_L$ , is

$$\begin{aligned} \Delta P_L &= 3.36 (10^{-6}) (0.0492) (1000)^2 (1 \text{ ft}) / (3.068)^5 (63) \\ &= 0.96 (10^{-5}) \text{ psi/ft} \end{aligned}$$

### 3. Gas pressure drop:

The gas velocity,  $v_G$ , is

$$\begin{aligned} v_G &= \left( \frac{W_G}{\rho_G A} \right) \\ v_G &= \frac{3000}{0.077 (3600) (0.0513)} = 211 \text{ ft/s} \\ \mu_e &= 0.00127/1488 = 0.000000854 \text{ lb/ft s} \\ Re &= D v_G \rho_G / \mu_e \\ &= (0.2557) (211) (0.077) / (0.000000854) \\ &= 4,900,000 \end{aligned}$$

$$\begin{aligned} A &= \frac{0.000587}{3.7} + \left( \frac{6.7}{4,900,000} \right)^{0.9} \\ &= 1.6392 \times 10^{-4} \end{aligned}$$

By substituting in Eq. (4-35) the friction factor,  $f_c$ , is

$$\begin{aligned} \frac{1}{\sqrt{f_c}} &= -4 \log \left[ \frac{0.000587}{3.7} - \frac{5.02}{4,900,000} \right] \\ &\quad \times \log (1.6392 \times 10^{-4}) \\ &= 15.1563 \\ f_c &= 0.004353 \end{aligned}$$

The Darcy friction factor,  $f_D = 4f_c$ , is

$$\begin{aligned} f_D &= 4 (0.004353) \\ &= 0.0174 \end{aligned}$$

The gas pressure drop,  $\Delta P_G$ , is

$$\begin{aligned} \Delta P_G &= 3.36 (10^{-6}) (0.0174) \\ &\quad \times (3000)^2 (1 \text{ ft}) / (3.068)^5 (0.077) \\ &= 0.0251 \text{ psi/ft} \end{aligned}$$

### 4. Lockhart–Martinelli two-phase flow modulus $X$ :

$$\begin{aligned} X &= (\Delta P_L / \Delta P_G)^{1/2} = (0.96 \times 10^{-5} / 2.51 \times 10^{-2})^{1/2} \\ &= 1.95 \times 10^{-2} \end{aligned}$$

### 5. For annular flow:

$$\begin{aligned} \Phi_{GTT} &= (4.8 - 0.3125d) X^{0.343-0.021d} \\ &= [4.8 - (0.3125) (3.068)] \\ &\quad \times (1.95 \times 10^{-2})^{0.343-0.021(3.068)} \\ &= 1.28 \end{aligned}$$

### 6. Two-phase flow for horizontal flows:

$$\Delta P_{TP} = \Delta P_G \Phi_{GTT}^2 = (0.0251) (1.28)^2 = 0.0411 \text{ psi/ft}$$

$$\begin{aligned} 7. F_e &= 0.00967 (W_L/A)^{0.5} / v_G^{0.7} \\ &= 0.00967 (19,494)^{0.5} / (211)^{0.7} \\ &= 0.032 \end{aligned}$$

Vertical elevation pressure drop component:

$$\begin{aligned} &= nh F_e \rho_L / 144 = [(3) (10) + (1) (50)] (0.032) (63) / 144 \\ &= 1.12 \text{ psi total} \end{aligned}$$

Total:

$$\begin{aligned} \Delta P_{TPh} &= (0.0411) (358) + 1.12 \\ &= 15.8 \text{ psi, total for pipe line} \end{aligned}$$

Because these calculations are somewhat uncertain due to lack of exact correlations, it is best to calculate pressure drop for other flow patterns, and apply a generous safety factor to the results.

Table 4-38 gives calculated results for other flow patterns in several different sizes of lines.

**TABLE 4-38 Two-Phase Flow Example**

Horizontal Flow Pattern					
Pipe ID (in.)	Annular (psi/ft)	Stratified (psi/ft)	Wave (psi/ft)	Elevation Factor, $F_e$	Gas velocity (ft/s)
3.068	0.0438	0.000367	0.131	0.032	210.9
4.026	0.0110	0.000243	0.0336	0.0465	122.5
6.065	0.00128	0.000131	0.00434	0.0826	53.9
7.981	0.00027	0.000087	0.00110	0.121	31.1
10.02	0.000062	0.000062	0.00035	0.166	19.7

or

$$\Delta P_{TP} = \frac{(q_{d14.65})^2 LS_g TZ}{20,000 d^5 P_{avg}} \left( \frac{f_g}{E^2} \right) \quad (4-290)$$

where  $E = 1/\Phi_{GTT}$ .

2. For the Panhandle equation, Baker [45] summarizes

$$q_{d14.65} = 0.43587 \left( \frac{T_s}{P_s} \right)^{1.07881} \left( \frac{P_1^2 - P_2^2}{ZTL_m} \right)^{0.5394} \left( \frac{d^{2.618}}{S_g^{0.4606}} \right) (E) \quad (4-291)$$

$$\text{where } E(\text{Panhandle}) = 0.9/\Phi_{GTT}^{1.077} \quad (4-292)$$

## 4.66 PIPE SIZING RULES

All two-phase flow correlations shown have been developed for long, horizontal pipes where uniform flow types are more likely to develop. In general, the application of all two-phase flow correlations to process piping design is arbitrary. These correlations

do not take into account the three-dimensional reality of process piping. Process piping has varying alternating flow regions because of pipe configurations, elevation changes, offsets, branch connections, manifolds, pipe components, reducers, and other restrictions. Large deviations can occur from friction loss predictions, compared to actual friction losses [56]. In using the correlations and graphs presented for two-phase flow, the following general pipe sizing rules are suggested:

*Dispersed Flow:* Apply  $\Delta P_{TP}$  (two-phase) throughout three-dimensional pipe (horizontal, up, and downflow sections). Use dispersed flow correlation for pipe smaller than  $2\frac{1}{2}$  in. for all flow regions.

*Annular and Bubble Flow:* Apply  $\Delta P_{TP}$  (two-phase) for 3 in. and larger pipe throughout the process line. Check the vertical upflow correlations and unit flow loss for long, vertical upflow runs. Use the upflow losses if they are greater than the annular or bubble flow unit losses.

*Stratified and Wave Flow:* Use stratified and wave flow correlations only for long, horizontal runs. Use annular flow correlations for three-dimensional, process pipe sizing where stratified and wave flow regions are determined.

### EXAMPLE 4-21

Figure 4-52 shows the piping configuration with a 4-in. control valve and a vessel. The available pressure difference  $\Delta P$  is 10 psi, including that of the control valve. The two-phase flow data in the line after the control valve are given below [56]. Determine a reasonable pipe size downstream of the control valve.

	Liquid	Gas
Flow, $W$ , lb/h	59,033	9336
Molecular weight, $M_w$	79.47	77.2
Density, lb/ft <sup>3</sup>	31.2	1.85
Viscosity, $\mu$ , cP	0.11	0.0105
Surface tension, $\sigma$ , dyn/cm	5.07	

(Source: R. Kern, "Piping Design For Two-Phase Flow", *Chem. Eng.*, Jun. 23, 1975).

*Solution*

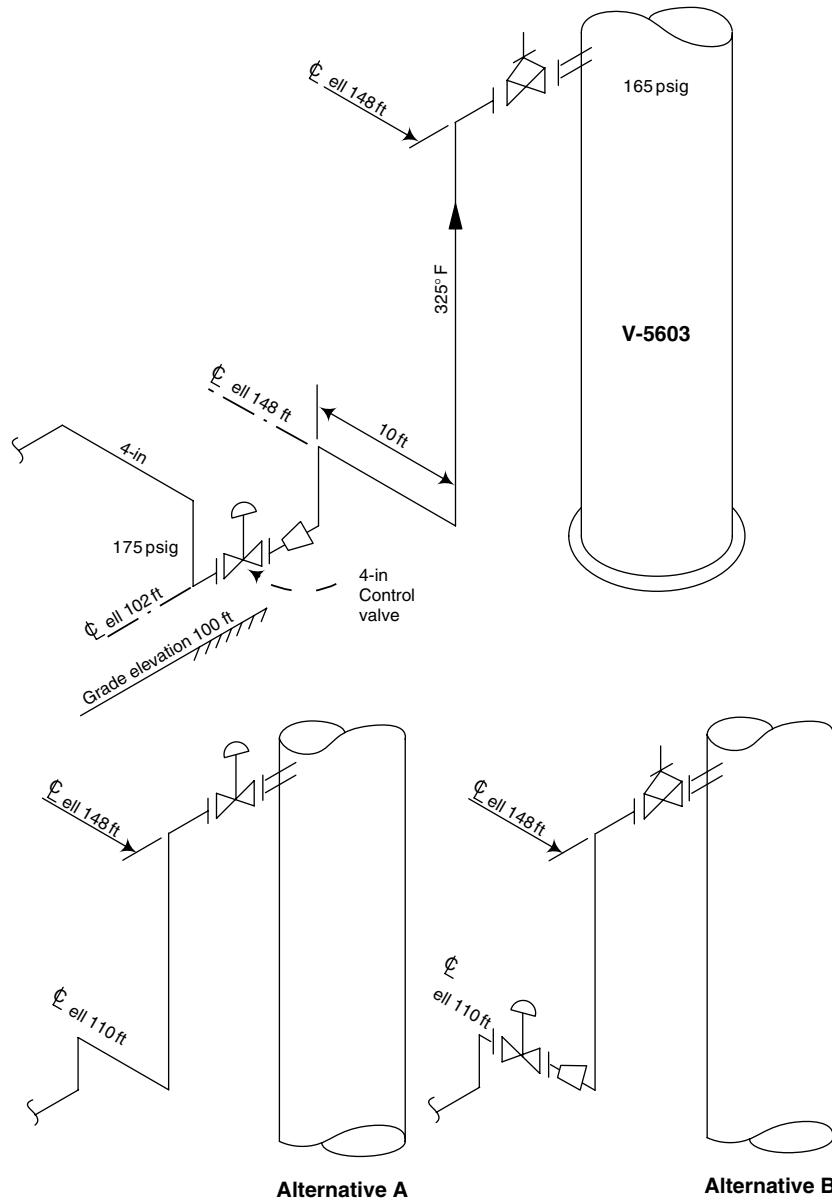
Table 4-39 shows typical computer results of Example 4-21 using a pipe size of 3 in. and Table 4-40 shows the results of 3, 4, 6, and 8 in. Sch. 40 respectively. The results of  $\Delta P_{Total}$  exclude the vertical sections of the piping configuration. These results show that the overall pressure drop of the two-phase ( $\Delta P_{TP}$ ) for both the

3-and 4-in. pipe sizes is greater than the available pressure drop of 10 psi (including that for the control valve; see Figure 4-52). The 6-and 8-in. pipe sizes show an overall pressure drop of the two-phase less than 10 psi, however the 8-in. pipe size indicates an undesirable slug flow pattern on the Baker map. The 6-in. pipe size gives a bubble flow on the Baker's map, and thus this pipe line is the optimum size.

Figure 4-52 provides alternate locations for the control valve for alternative A. Here, slug flow cannot develop. However, for alternative B, a shortened and self-draining pipe line improves the pipe configuration but at the expense of convenient access to the control valve. In both alternatives, there is considerable turbulence after the control valve, which helps to provide slug-free liquid-phase carry over.

The following are ways to adjust the pressure loss distribution in a pipe system [56]:

1. Change the pipe size.
2. Design a section of the pipe line with either an increase or a decrease in pipe diameter.
3. Adjust the static head of elevated vessels.
4. Change valve or orifice restrictions to consume more or less pressure drop (differentials).



**Figure 4-52** Configurations of piping for sample problem of Example 4-21 [56].

For stratified flow, use the Eq. (4-258) to determine the two-phase modulus and apply the Huntington correlation to determine the wave flow unit loss calculations.

**Plug Flow:** The process piping designer rarely meets with plug flow conditions. (But slug flow is not uncommon.)

#### 4.67 A SOLUTION FOR ALL TWO-PHASE PROBLEMS

Dukler et al. [58] have pointed out that only three flow regimes are apparent in any piping configuration: [*segregated, intermittent, and distributed.*]

Segregated flow occurs when the gas and liquid are continuous in the axial direction. Stratified flow is easily recognized as belonging to this category, as do the wavy and annular regimes (see Figure 4-49).

Intermittent flow results when the phases form alternating pockets. Plug and slug flows therefore fall in this grouping. Flow

is considered distributive when one fluid phase is continuous and flows to some degree in the directions which are both perpendicular and parallel to the pipe axis. The other phase may not necessarily be distributed uniformly over the same section of the pipe, but should be locally continuous. Mist flow and bubble flow are included in this type of regime.

These regimes, which completely characterize any flow type, simplify the analysis of a physical situation by resolving into three the numerous regimes described earlier.

Erwin [59] expressed that in considering Baker's froth zone flow regime, to have a froth or homogeneous flowing gas-liquid mixture, a high Reynolds number is required ( $Re \geq 200,000$ ). Every case of refinery, oil and gas, and chemical plant piping involves higher Reynolds number for economic pipe sizing; even pipelines are sized for higher Reynolds numbers. A pipe flowing at 3 ft/s would qualify for the minimum Reynolds number of 200,000. Dukler's [60, 61] work resolved two-phase flow pipe sizing and

**TABLE 4-39 Computer Results of Two – Phase Pressure Drop Calculation of Example 4-21**

TWO-PHASE PRESSURE DROP CALCULATION IN A PIPE LINE	
PIPE INTERNAL DIAMETER, inch:	3.068
EQUIVALENT LENGTH OF PIPE, ft.:	.000
ACTUAL LENGTH OF PIPE, ft.:	100.000
TOTAL LENGTH OF PIPE, ft.:	100.000
LIQUID DENSITY, lb/ft <sup>3</sup> :	63.000
LIQUID VISCOSITY, cP:	1.000
SURFACE TENSION, dyne/cm.:	15.000
LIQUID FLOWRATE, lb/hr.:	1000.00
LIQUID REYNOLDS NUMBER:	2057.
LIQUID FRICTION FACTOR:	.0311
PRESSURE DROP OF LIQUID PER 100ft., psi/100ft.:	.0007
GAS FLOW RATE, lb/hr.:	3000.00
GAS DENSITY, lb/ft <sup>3</sup> :	.077
GAS VISCOSITY, cP:	.001
GAS REYNOLDS NUMBER:	4746264.
GAS FRICTION FACTOR:	.0174
PRESSURE DROP OF GAS PER 100ft., psi/100ft.:	2.7406
FLOW REGIME IS:	ANNULAR
LOCKHART-MARTINELLI TWO PHASE FLOW MODULUS:	.0156
VELOCITY OF FLUID IN PIPE, ft./sec.:	210.7728
BAKER PARAMETER IN THE LIQUID PHASE:	1.641
BAKER PARAMETER IN THE GAS PHASE:	57308.790
TWO-PHASE FLOW MODULUS:	1.4518
PRESSURE DROP OF TWO-PHASE MIXTURE, psi/100 ft.:	3.9789
OVERALL PRESSURE DROP OF THE TWO-PHASE, psi:	3.9789
INDEX 4564. IS LESS THAN 10000.0 PIPE EROSION IS UNLIKELY	

configuration problems. The key to the success is maintaining  $Re \geq 200,000$ , which is accomplished by making the pipe size small enough. Dukler's work is summarized as follows:

- For pressure loss due to friction, first determine the homogeneous flow liquid ratio  $\lambda$ , volume of liquid per volume of mixed fluid flow.

$$\lambda = \frac{Q_{LPL}}{Q_{GPL} + Q_{LPL}} \quad (4-293)$$

where

$$Q_{LPL} = \text{volume of liquid flow } W_L / \rho_L, \text{ ft}^3/\text{h}$$

$$Q_{GPL} = \text{volume of gas flow, } W_G / \rho_G, \text{ ft}^3/\text{h}$$

$$W_L = \text{liquid flow, lb/h}$$

$$W_G = \text{gas flow, lb/h}$$

$$\rho_L = \text{liquid density at flow pressure and temperature, lb/ft}^3$$

$$\rho_G = \text{gas density at flow pressure and temperature, lb/ft}^3$$

The calculated  $\lambda$  value is valid only over a range in which the pressure loss in the pipe does not exceed 15% of inlet value. For a large pressure loss, the pipe run is divided into several segments with each segment having different pressure inlet and different temperature due to the gas flashing cooling effect.

- The ratio of two-phase friction factor to gas-phase friction factor in the pipeline is determined as:

$$S = 1.281 + 0.478 (\ln \lambda) + 0.444 (\ln \lambda)^2 + 0.09399999 (\ln \lambda)^3 + 0.008430001 (\ln \lambda)^4 \quad (4-294)$$

**TABLE 4-40 Computer Results of Two-Phase Pressure Drop Calculation of Example 4-21**

Pipe Internal Diameter (in. Sch. 40)	3.068	4.026	6.065	7.981
Equivalent length of pipe, ft	76.445	98.822	145.729	188.042
Actual length of pipe, ft	56.0	56.0	56.0	56.0
Total length of pipe, ft	132.445	154.822	201.729	244.042
Liquid density, lb/ft <sup>3</sup>	31.2	31.2	31.2	31.2
Liquid viscosity, cP	0.11	0.11	0.11	0.11
Surface tension, dyn/cm	5.07	5.07	5.07	5.07
Liquid flow rate, lb/h	59,033.0	59,033.0	59,033.0	59,033.0
Liquid Reynolds number	1,103,764.0	841,120.0	58,343.0	424,301.0
Liquid friction factor	0.0177	0.170	0.0162	0.0159
Pressure drop of liquid, psi/100 ft	2.6685	0.6555	0.0805	0.0201
Gas flow rate, lb/h	9,336.0	9336.0	9,336.0	9,336.0
Gas density, lb/ft <sup>3</sup>	1.85	1.85	1.85	1.85
Gas viscosity, cP	0.01	0.01	0.01	0.01
Gas Reynolds number	1,828,713.0	1,393,565.0	925,060.0	702,981.0
Gas friction factor	0.0176	0.0167	0.0157	0.0153
Pressure drop of liquid, psi/100 ft	1.115	0.2723	0.033	0.0082
Flow regime	Bubble	Bubble	Bubble	Slug
Lockhart–Martinelli two-phase modulus	1.547	1.5514	1.5617	1.5709
Velocity of fluid in pipe, ft/s	37.5212	21.7891	9.6012	5.5446
Baker parameter in the liquid phase	243.246	243.246	243.246	243.246
Baker parameter in the gas phase	51702.71	30,024.55	13,230.08	7,640.28
Two-phase flow modulus	23.806	26.652	31.711	17.400
Pressure drop of two-phase mixture, psi/100 ft	26.544	7.258	1.047	0.1418
Overall pressure drop of the two-phase, psi	35.156	11.238	2.113	0.3461
Index for pipe erosion	13888.0	4,684.0	909.0	303.0

$$\frac{f_{TP}}{f_o} = 1 - \frac{\ln \lambda}{S} \quad (4-295)$$

where

$f_{TP}$  = two-phase flow friction factor in the pipe run  
 $f_o$  = gas-phase friction factor in the pipe run  
 $\ln$  = natural logarithm of base e, 2.7183.

3. The Reynolds number is calculated. Dukler developed experimental data in calculating liquid holdup in two-phase flow systems.  $Re > 200,000$  are free of liquid slugs and holdup. If  $Re$  is greater than 200,000 then the flow is in the froth regime, or it is homogeneous flow as a mixture. For homogeneous flow, the average density of the two-phase fluid mixture is

$$\rho_m = \rho_L \lambda + \rho_G (1 - \lambda) \quad (4-296)$$

The average viscosity  $\mu_m$ , lb/ft s:

$$\mu_m = \frac{\mu_L}{1488} \lambda + \frac{\mu_G}{1488} (1 - \lambda) \quad (4-297)$$

where

$\mu_L$  = liquid viscosity, cP  
 $\mu_G$  = gas viscosity, cP.

Calculate the mixture flowing velocity  $v_m$ , ft/s

$$v_m = \frac{Q_{GPL} + Q_{LPL}}{3600 (\pi D^2 / 4)} \quad (4-298)$$

where  $D$  = pipe internal diameter, ft.

The two-phase Reynolds number is

$$Re = \frac{D v_m \rho_m}{\mu_m} \quad (4-299)$$

4. Calculate the two-phase flow friction factor  $f_{TP}$  as follows. First define  $f_{TP}$  from  $f_o$ . In charting  $f_{TP}/f_o$  against  $\lambda$ , Dukler expressed

$$f_o = 0.0014 + \frac{0.125}{Re^{0.32}} \quad (4-300)$$

$$f_{TP} = \frac{f_o}{f_o} \quad (4-301)$$

$Re$  must be 200,000 or greater before these equations can be applied. Knowing  $Re$  and the ratio of  $f_{TP}/f_o$ ,  $f_{TP}$  is calculated from Eqs (4-300) and (4-301).

5. The pipe friction pressure loss of straight pipe,  $\Delta P_f$ , is

$$\Delta P_f = 4 \frac{f_{TP}}{144g} \left( \frac{L}{D} \right) \rho_m \frac{v_m^2}{2}, \text{ psi} \quad (4-302)$$

where

$L$  = straight pipe length, ft  
 $g$  = acceleration of gravity, 32.2 ft/s<sup>2</sup>.

6. Calculate the pressure drop due to elevation changes  $\Delta P_E$ , psi. First determine the superficial gas velocity  $v_{sg}$  as

$$v_{sg} = \frac{Q_{GPL}}{3600 (\pi D^2 / 4)}, \text{ ft/s} \quad (4-303)$$

$v_{sg}$  is the velocity of the gas alone in the full cross-sectional area of the pipe, ft/s. A factor  $\phi$  is related to the two-phase gas velocity  $v_{sg}$ , and its value increases as the gas velocity decreases. A curve fit equation of  $\phi$  vs.  $v_{sg}$  is

$$\begin{aligned} \phi = & 0.76844 - 0.085389 v_{sg} + 0.0041264 v_{sg}^2 \\ & - 0.0000087165 v_{sg}^3 + 0.00000066422 v_{sg}^4 \end{aligned} \quad (4-304)$$

Equation (4-304) determines the  $\phi$  value, which is the correction to the static leg rise or fall of the gas phase. As the gas velocity approaches 0,  $\phi$  approaches unity. Equation (4-304) has a range limit that is

If  $v_{sg} > 50$ , then  $\phi = 0.04$

If  $v_{sg} < 0.5$ , then  $\phi = 0.85$

$$\Delta P_E = \frac{\phi \rho_L H_T}{144} \quad (4-305)$$

where  $H_T$  = height of static leg (−) for rise and (+) for fall, ft.

Here the liquid density,  $\rho_L$ , is used in Eq. (4-305), since  $\phi$  corrects for the gas-phase static leg  $\Delta P$ .

7. Calculate the pressure drop due to acceleration or pipe fittings and valves  $\Delta P_A$ , psi.

The 90° standard elbow  $\Delta P_{ell}$  is calculated as follows:

$$\Delta P_A = \frac{\rho_G Q_{GPL}^2}{1 - \lambda} + \frac{\rho_L Q_{LPL}^2}{\lambda} \quad (4-306)$$

$$\Delta P_{ell} = \frac{\Delta P_A}{[3.707 \times 10^{10} + (d/12)^4]} \quad (4-307)$$

where  $d$  = internal pipe diameter, in.

If a 15% pressure loss in a pipe segment results, then a new pipe segment is required.

Tee angle	$\Delta P_A = 3.0 \Delta P_{ell}$
Tee straight	$\Delta P_A = 1.0 \Delta P_{ell}$
Check valve	$\Delta P_A = 2.5 \Delta P_{ell}$

For two-phase flow pipe entrance and exit,

$$\Delta P_A = 4 f_{TP} [6.469 (\ln d) + 24] \frac{\rho_m}{144g} \left( \frac{v_m^2}{2} \right) \quad (4-308)$$

Pipe sharp – edge exit:

$$\Delta P_A = 4 f_{TP} [14.403 (\ln d) + 42] \frac{\rho_m}{144g} \left( \frac{v_m^2}{2} \right) \quad (4-309)$$

8. The total two-phase pressure loss  $\Delta P_T$  is

$$\Delta P_T = \Delta P_f + \Delta P_E + \Delta P_A \quad (4-310)$$

Note: All preceding seven steps are made on the assumption that  $Re$  is 200,000 or greater.

Table 4-41 shows a glossary of two-phase flow.

**TABLE 4-41 A Glossary of Two-Phase Flow**

**Critical Flow:** When a point is reached in the system where the increase in specific volume for a small decrease in pressure is so great that the pressure and the enthalpy can no longer be simultaneously lowered across a cross section of pipe, it is called critical flow. It is analogous to sonic flow in a single phase flow. This does not imply, however, that the sonic velocity of a superficially flowing gas phase in a two-phase system is equal to the sonic or critical velocity of the two-phase system. Critical flow occurs in the so-called mist flow regime.

**Flow Regimes:** A flowing two-phase fluid can exhibit several “patterns” of flow, such as the liquid occupying the bottom of the conduit with the gas phase flowing above, or a liquid phase with bubbles of gas distributed throughout. In essence, flow regimes are the physical geometry exhibited by the two-phase mixture in the conduit. They are influenced by pipe geometry as well as the physical properties of the fluid mixture and flow rate.

**Flowing Volume Holdup:** This term is given as the ratio of the superficial liquid velocity to the sum of the superficial gas and liquid velocities. The term arises in computing properties of homogeneous system and results naturally from the assumption of no slip flow.

**Homogeneous Flow:** A mathematical model that considers a two-phase system as a single homogeneous fluid with properties representing the volumetric flow averages of the liquid and gas phases. Homogeneous flow does not exist in real physical situations.

**Mist Flow:** At high gas flow velocities, the majority of the liquid becomes distributed as droplets in the gas phase. The liquid is said to be entrained and flow is described as mist flow.

**Plug Flow:** This is a flow regime where most, but not all, aggregates of the liquid phase occupy most of the cross section of the pipe for a given length of conduit. A similar length is occupied by all gas. The regimes alternate down the conduit.

**Slip:** For the majority of the fluid's history, one of the phases is flowing faster than the other. Thus, one phase seems to slip by the slower phase. Slip velocity is the difference in the phase velocities.

**Slug Flow:** A flow regime characterized by each phase alternately occupying the entire cross section for a large length of the conduit: a “slug” of liquid or gas.

**Superficial Liquid Velocity:** The velocity that the liquid phase would have in the pipe if there were no gas-phase flowing. Thus, it is the volumetric liquid flow rate divided by the cross-sectional area of the pipe.

**Superficial Gas Velocity:** Defined in a manner similar to superficial liquid velocity.

(Source: A.E. DeGance and R.W. Atherton, *Chem. Eng.*, Mar 23, 1970, p. 135.)

### EXAMPLE 4-22

Using the data in Example 4-21, determine the total two-phase pressure drop downstream of the control valve.

	Liquid	Gas
Flow, $W$ , lb/h	59,033	9,336
Molecular weight, $M_w$	79.47	77.2
Density, lb/ft <sup>3</sup>	31.2	1.85
Viscosity, $\mu$ , cP	0.11	0.0105
Surface tension, $\sigma$ , dyn/cm	5.07	

(Source: R. Kern, “Piping Design For Two-Phase Flow”, *Chem. Eng.*, Jun 23, 1975).

Data:

Assume a 6-in. pipe size, ID = 6.065 in.

Straight pipe length = 56 ft.

Elevation (Static leg rise),  $H_T$  = 38 ft.

Fittings:

4 × 90° ell

1 pipe exit to vessel V-5603.

*Solution*

Two-phase flow after the control valve pressure drop.

1. Calculate the homogeneous flow liquid ratio  $\lambda$ :

$$\lambda = \frac{Q_{LPL}}{Q_{GPL} + Q_{LPL}} \quad (4-293)$$

where

$$Q_{LPL} = \frac{59,033}{31.2} = 1892.1 \text{ ft}^3/\text{h}$$

$$Q_{GPL} = \frac{9336}{1.85} = 5046.5 \text{ ft}^3/\text{h}$$

$$\lambda = \frac{1892.1}{(5046.5 + 1892.1)} = 0.2727$$

2. Calculate  $f_{TP}/f_o$ :

The ratio of two-phase friction factor to gas-phase friction factor in the pipeline is determined as

$$\begin{aligned} S &= 1.281 + 0.478 (\ln 0.2727) + 0.444 (\ln 0.2727)^2 \\ &\quad + 0.09399999 (\ln 0.2727)^3 \\ &\quad + 0.008430001 (\ln 0.2727)^4 \end{aligned} \quad (4-294)$$

$$S = 1.2274$$

(continued)

**EXAMPLE 4-22—(continued)**

Substituting the value of  $\lambda$  and  $s$  in Eq. (4-295)

$$\frac{f_{\text{TP}}}{f_o} = 1 - \frac{\ln (0.2727)}{1.2274} = 2.0587$$

- 3.** Calculate the Reynolds number  $Re$  of the mixture density  $\rho_m$ , lb/ft<sup>3</sup>:

$$\rho_m = \rho_L \lambda + \rho_G (1 - \lambda) \quad (4-296)$$

$$\begin{aligned} \rho_m &= (31.2) (0.2727) + (1.85) (1 - 0.2727) \\ &= 9.854 \text{ lb/ft}^3 \end{aligned}$$

The average viscosity  $\mu_m$ , using Eq. (4-297) is given by

$$\begin{aligned} \mu_m &= \frac{0.11}{1488} (0.2727) + \frac{0.0105}{1488} (1 - 0.2727) \\ &= 2.529 \times 10^{-5} \text{ lb/ft s.} \end{aligned}$$

Calculate the mixture flowing velocity  $v_m$ , ft/s using Eq. (4-298)

$$\begin{aligned} v_m &= \frac{(5046.5 + 1892.1)}{3600 [\pi (0.5054)^2 / 4]} \\ &= 9.607 \text{ ft/s} \end{aligned}$$

The two-phase Reynolds number  $Re$  is

$$Re = \frac{D v_m \rho_m}{\mu_m} \quad (4-299)$$

$$\begin{aligned} Re &= \frac{(0.5054) (9.607) (9.854)}{(2.529 \times 10^{-5})} \\ &= 1.89 \times 10^6 (Re > 200,000) \end{aligned}$$

- 4.** Calculate the two-phase flow friction factor  $f_{\text{TP}}$  as follows:

First define  $f_{\text{TP}}$  from  $f_o$ . In charting  $f_{\text{TP}}/f_o$  against  $\lambda$ , Dukler expressed

$$f_o = 0.0014 + \frac{0.125}{Re^{0.32}} \quad (4-300)$$

$$f_o = 0.0014 + \frac{0.125}{(1.89 \times 10^6)^{0.32}} = 0.00263$$

$$f_{\text{TP}} = \frac{f_{\text{TP}}}{f_o} f_o \quad (4-301)$$

$$f_{\text{TP}} = (2.0587) (0.00263) = 0.0054$$

- 5.** Calculate the friction pressure loss  $\Delta P_f$  of straight pipe:

$$\Delta P_f = 4 \frac{f_{\text{TP}}}{144 g} \left( \frac{L}{D} \right) \rho_m \frac{v_m^2}{2}, \text{ psi} \quad (4-302)$$

$$\begin{aligned} \Delta P_f &= 4 \frac{(0.0054)}{(144) (32.2)} \left( \frac{56}{0.5054} \right) (9.854) \frac{(9.607^2)}{2} \\ &= 0.235 \text{ psi} \end{aligned}$$

- 6.** Calculate the pressure drop due to elevation changes  $\Delta P_E$ , psi: First determine the superficial gas velocity  $v_{sg}$  as

$$v_{sg} = \frac{Q_{\text{GPL}}}{3600 (\pi D^2 / 4)} \quad (4-303)$$

$$\begin{aligned} v_{sg} &= \frac{5046.5}{3600 [\pi (0.5054)^2 / 4]} \\ &= 6.99 \text{ ft/s} \end{aligned}$$

$$\begin{aligned} \phi &= 0.76844 - 0.085389 v_{sg} + 0.0041264 v_{sg}^2 \\ &\quad - 0.000087165 v_{sg}^3 + 0.00000066422 v_{sg}^4 \end{aligned} \quad (4-304)$$

$$\begin{aligned} \phi &= 0.76844 - 0.085389 (6.99)^2 + 0.0041264 (6.99)^2 \\ &\quad - 0.000087165 (6.99)^3 + 0.00000066422 (6.99)^4 \end{aligned}$$

$$\phi = 0.345$$

$$\Delta P_E = \frac{\phi \rho_L H_T}{144} \quad (4-305)$$

$$\Delta P_E = \frac{(0.345) (31.2) (-38)}{144}, \text{ psi}$$

$$= -2.84 \text{ psi}$$

- 7.** Calculate the pressure drop due to acceleration or pipe fittings and valves  $\Delta P_A$ , psi:

The 90° standard elbow  $\Delta P_{\text{ell}}$  is calculated as follows:

$$\Delta P_A = \frac{\rho_G Q_{\text{GPL}}^2}{1 - \lambda} + \frac{\rho_L Q_{\text{LPL}}^2}{\lambda} \quad (4-306)$$

$$\begin{aligned} \Delta P_A &= \frac{(1.85) (5046.5)^2}{(1 - 0.2727)} + \frac{(31.2) (1892.1)^2}{0.2727} \\ &= 474,377,476.2 \text{ psi} \end{aligned}$$

(continued)

**EXAMPLE 4-22—(continued)**

$$\Delta P_{\text{ell}} = \frac{\Delta P_A}{[3.707 \times 10^{10} + (d/12)^4]} \quad (4-307)$$

$$\Delta P_{\text{ell}} = \frac{(474,377,476.2)}{[3.707 \times 10^{10} + (6.065/12)^4]}$$

$$= 0.0128 \text{ psi}$$

4 × 90° standard elbow

$$\Delta P_A = 4 \times 0.0128$$

$$= 0.051 \text{ psi.}$$

Pipe sharp – edge exit:

$$\Delta P_A = 4f_{\text{TP}} [14.403 (\ln d) + 42] \frac{\rho_m}{144 g} \left( \frac{v_m^2}{2} \right) \quad (4-309)$$

$$\Delta P_A = 4 (0.0054) [14.403 (\ln 6.065) + 42]$$

$$\frac{(9.854)}{(144)(32.2)} \left( \frac{9.607^2}{2} \right)$$

$$= 0.144 \text{ psi}$$

8. The total two-phase pressure loss  $\Delta P_T$  is

$$\Delta P_T = \Delta P_f + \Delta P_E + \Delta P_A \quad (4-310)$$

$$\Delta P_T = 0.235 - 2.84 + (0.051 + 0.144)$$

$$\Delta P_T = -2.41 \text{ psi}$$

The Excel spreadsheet Example 4-22.xls has been developed to determine the two-phase pressure drop of Example 4-22 and Figure 4-53 shows snapshots of the calculations.

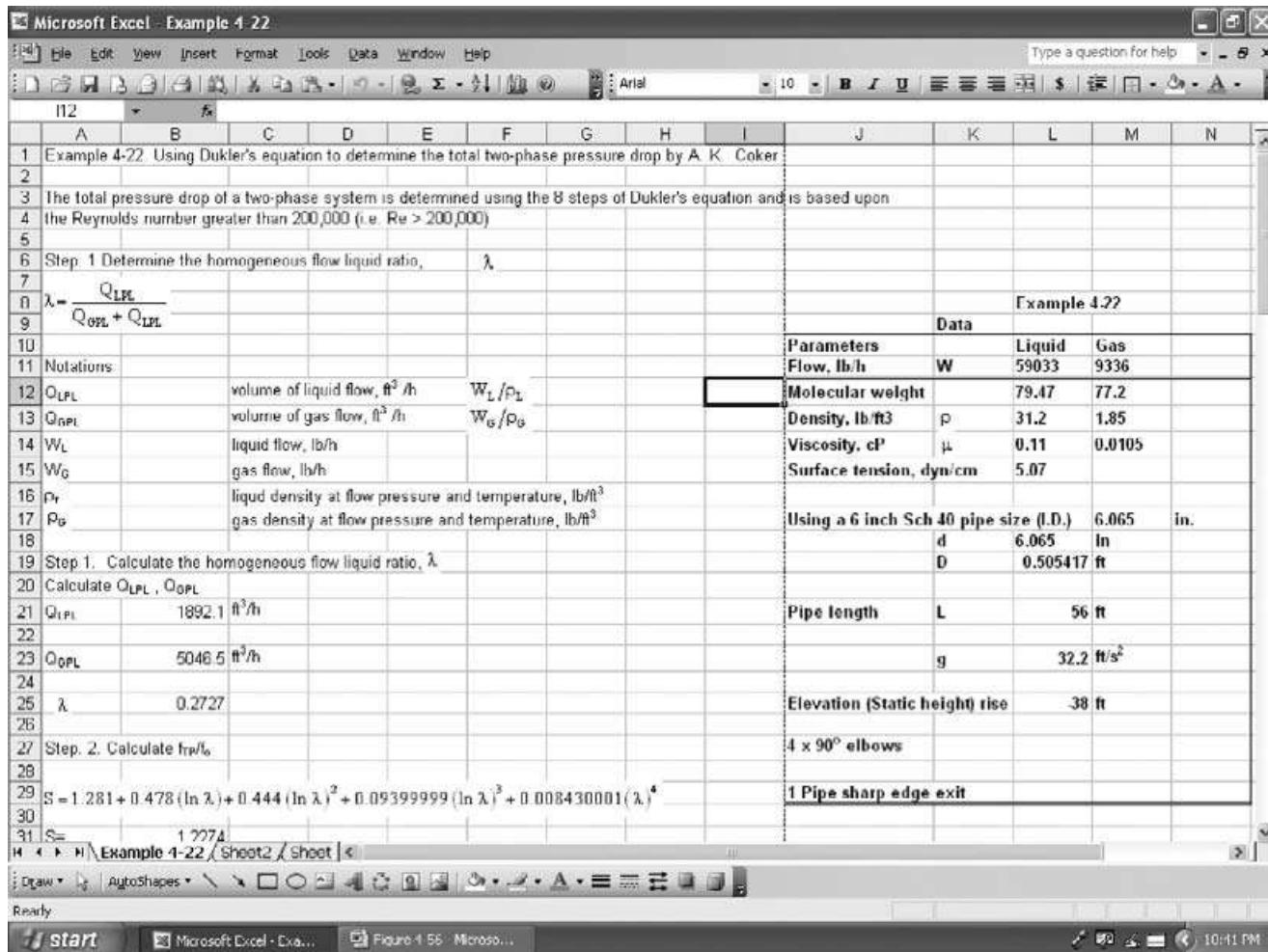


Figure 4-53 Excel spreadsheet calculation of Example 4-22.

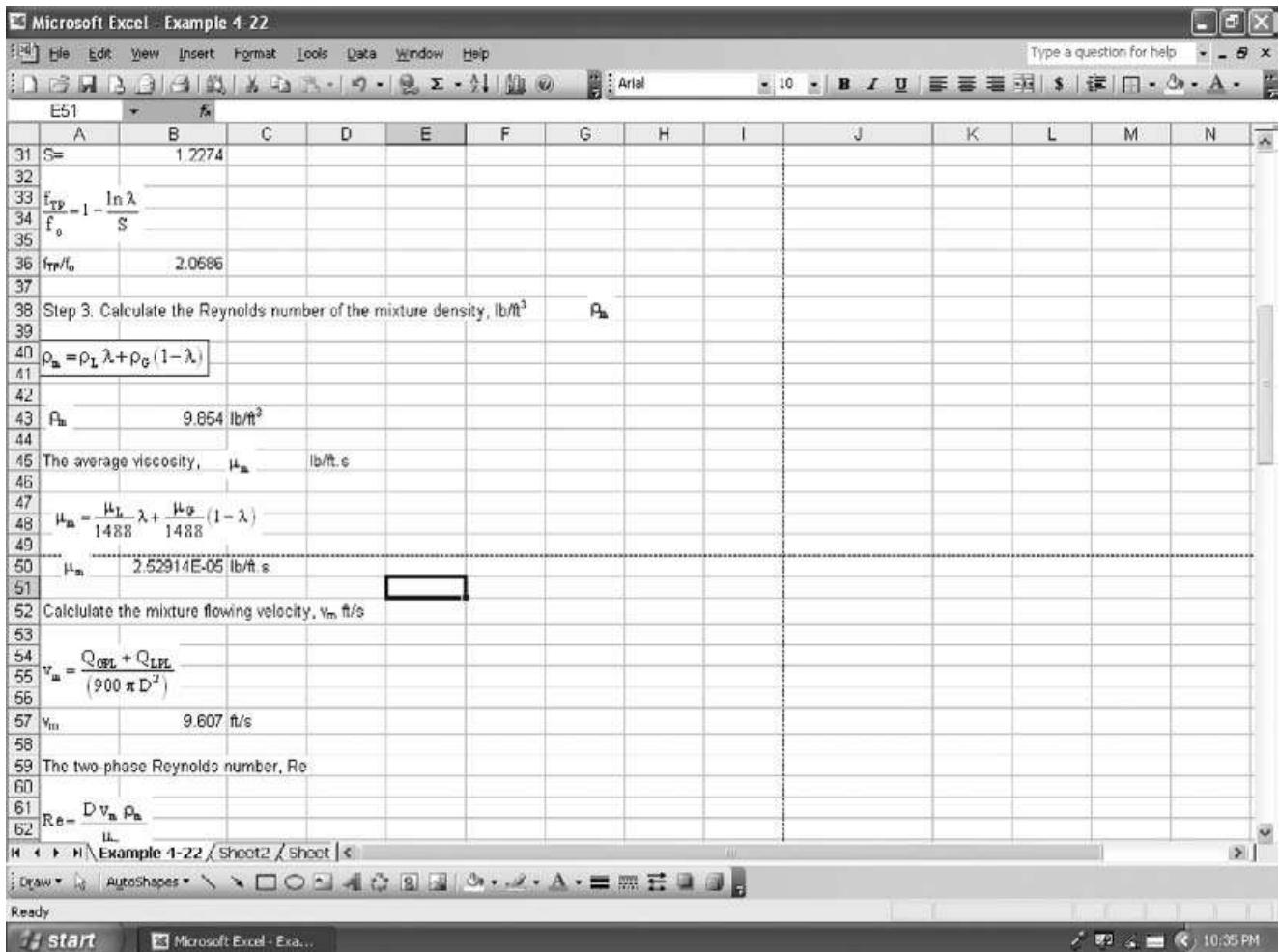


Figure 4-53—(continued)

#### 4.68 GAS-LIQUID TWO-PHASE VERTICAL DOWNFLOW

Two-phase vertical downflow presents its own problems as often occurs in horizontal pipe line. In a vertical flow, large gas bubbles are formed in the liquid stream resulting in a flow regime known as slug flow. This flow regime (is associated with) can result in pipe vibration and pressure pulsation. With bubbles greater than 1 in. in diameter and the liquid viscosity less than 100 cP, slug flow region can be represented by dimensionless numbers for liquid and gas phases respectively (Froude numbers  $(N_{Fr})_L$  and  $(N_{Fr})_G$ ). These are related by the ratio of inertial to gravitational force and are expressed as

$$(N_{Fr})_L = \frac{v_L}{(gD)^{0.5}} \left( \frac{\rho_L}{\rho_L - \rho_G} \right)^{0.5} \quad (4-311)$$

$$(N_{Fr})_G = \frac{v_G}{(gD)^{0.5}} \left( \frac{\rho_G}{\rho_L - \rho_G} \right)^{0.5} \quad (4-312)$$

The velocities  $v_G$  and  $v_L$  are superficial velocities based on the total pipe cross section. These Froude numbers exhibit several features in the range  $0 < (N_{Fr})_L$  and  $(N_{Fr})_G < 2$ . Simpson [4] illustrates the

values of  $(N_{Fr})_L$  and  $(N_{Fr})_G$  with water flowing at an increased rate from the top of an empty vertical pipe. As the flow rate further increases to the value  $(N_{Fr})_L = 2$ , the pipe floods and the total cross section is filled with water. If the pipe outlet is further submerged in water and the procedure is repeated, long bubbles will be trapped in the pipe below  $(N_{Fr})_L = 0.31$ . However, above  $(N_{Fr})_L = 0.31$ , the bubbles will be swept downward and out of the pipe.

If large long bubbles are trapped in a pipe ( $d \geq 1$  in.) in vertically down-flowing liquid having a viscosity less than 100 cP and the Froude number for liquid phase  $(N_{Fr})_L \leq 0.3$ , the bubbles will rise. At higher Froude numbers, the bubbles will be swept downward and out of the pipe. A continuous supply of gas causes the Froude number in the range  $0.31 \leq (N_{Fr})_L < 1$  to produce pressure pulsations and vibration. These anomalies are detrimental to the pipe and must be avoided. If the Froude number is greater than 1.0, the frictional force offsets the effect of gravity, and thus requires no pressure gradient in the vertical downflow liquid. This latter condition depends on the Reynolds number and pipe roughness. Figure 4-54 shows the flow patterns in a vertical liquid-gas flow and Figure 4-55 shows a correlation in a cocurrent vertical upflow of air-water mixture in terms of Froude numbers.

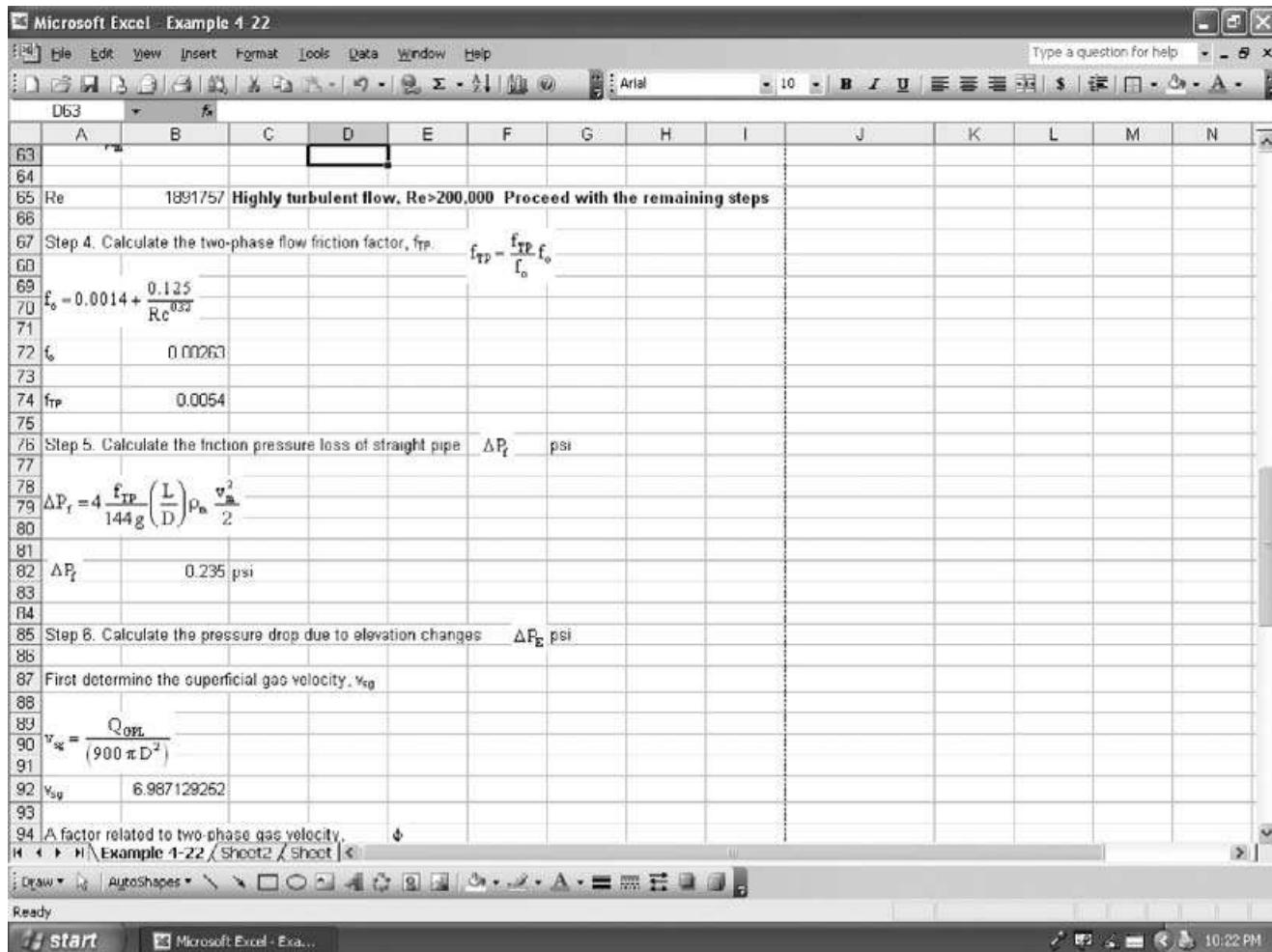


Figure 4-53—(continued)

## THE EQUATIONS

The following equations will calculate Froude numbers for both the liquid and the gas phases. A developed spreadsheet program will print out a message indicating if the vertical pipe is self-venting, if pulsating flow occurs, or if no pressure gradient is required.

$$D = \frac{d}{12}, \text{ ft}$$

$$\text{Area} = \frac{\pi D^2}{4}, \text{ ft}^2 \quad (4-313)$$

$$v_L = \frac{W_L}{(3600)(\rho_L)(\text{Area})}, \text{ ft/s} \quad (4-314)$$

$$v_G = \frac{W_G}{(3600)(\rho_G)(\text{Area})}, \text{ ft/s} \quad (4-315)$$

$$\text{FRNL} = \frac{v_L}{(gD)^{0.5}} \left[ \frac{\rho_L}{\rho_L - \rho_G} \right]^{0.5} \quad (4-316)$$

$$\text{FRNG} = \frac{v_G}{(gD)^{0.5}} \left[ \frac{\rho_G}{\rho_L - \rho_G} \right]^{0.5} \quad (4-317)$$

where

Area = inside cross-sectional area of pipe,  $\text{ft}^2$

$d$  = inside diameter of pipe, in.

$D$  = inside diameter of pipe, ft

FRNL ( $N_{Fr}$ )<sub>L</sub> = Froude number of liquid phase, dimensionless

FRNG ( $N_{Fr}$ )<sub>G</sub> = Froude number of gas phase, dimensionless

$g$  = gravitational constant,  $32.2 \text{ ft/s}^2$

$v_L$  = liquid velocity,  $\text{ft/s}$

$v_G$  = gas velocity,  $\text{ft/s}$

$W_L$  = liquid flow rate,  $\text{lb/h}$

$W_G$  = gas flow rate,  $\text{lb/h}$

$\rho_L$  = liquid density,  $\text{lb/ft}^3$

$\rho_G$  = gas density,  $\text{lb/ft}^3$

## THE ALGORITHMS

If  $\text{FRNL} < 0.31$ , vertical pipe is SELF VENTING

ELSE

$0.3 \leq \text{FRNL} < 1.0$ , PULSE FLOW, and may result in pipe vibration.

$\text{FRNL} > 1.0$ , NO PRESSURE GRADIENT.

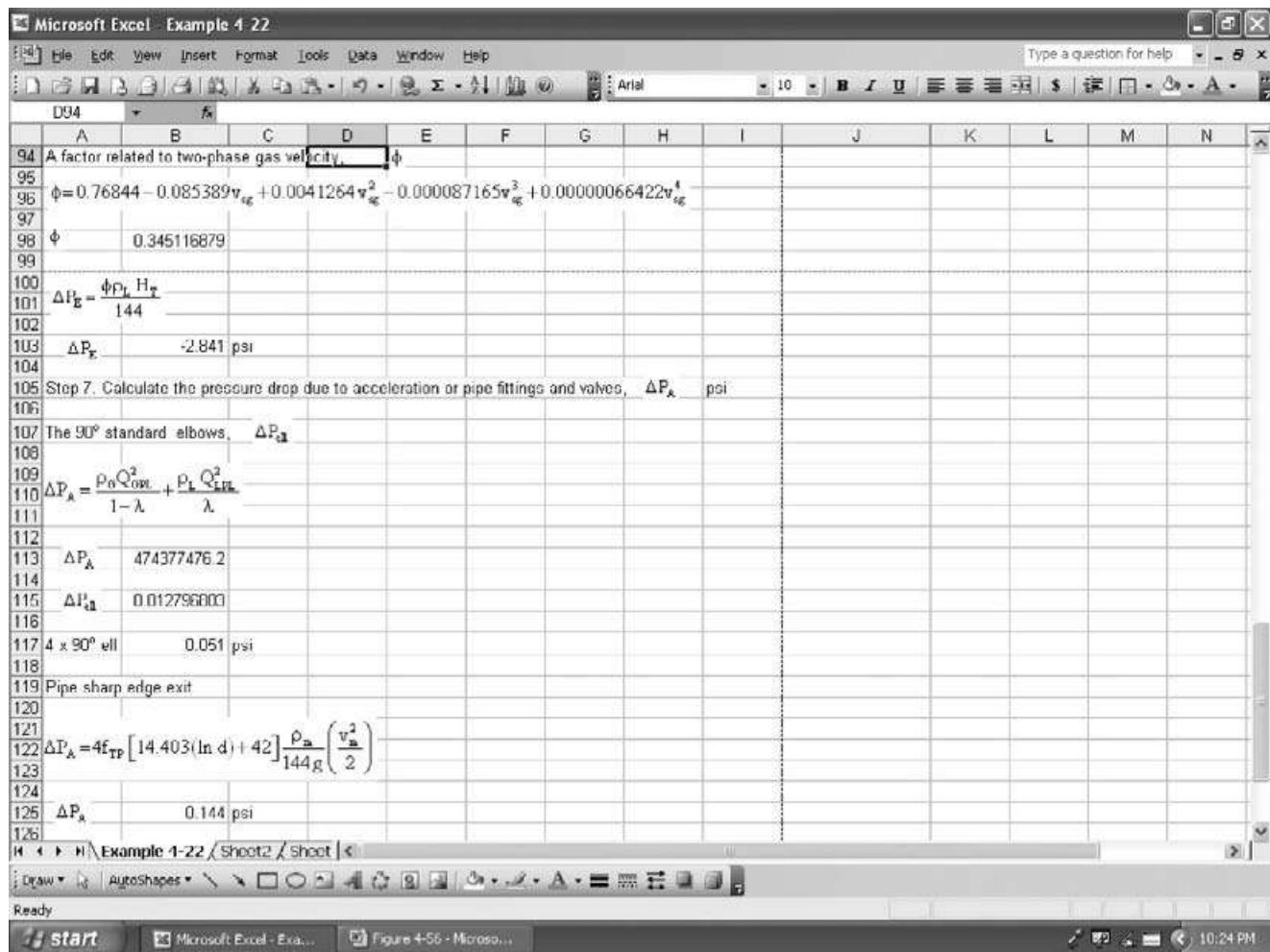


Figure 4-53—(continued)

**EXAMPLE 4-23**

Calculate the Froude numbers and flow conditions for the 2-, 4-, and 6-in. (Sch. 40) vertical pipes having the following liquid and vapor flow rates and densities.

	Liquid	Vapor
Mass flow rate, lb/h	6930	1444
Density, lb/ft <sup>3</sup>	61.8	0.135

*Solution*

2-in. pipe diameter (Sch. 40) ID = 2.067 in.

$$D = 2.067/12 = 0.17225 \text{ ft}$$

$$\begin{aligned} \text{Area} &= \frac{\pi D^2}{4} = \frac{\pi (0.17225)^2}{4} \\ &= 0.0233 \text{ ft}^2 \end{aligned} \quad (4-313)$$

Liquid velocity,  $v_L$ , is

$$\begin{aligned} v_L &= \frac{W_L}{(\rho_L A)} \\ &= \frac{6930}{(61.8)(0.0233)(3600)} \\ &= 1.34 \text{ ft/s} \end{aligned} \quad (4-314)$$

Vapor velocity,  $v_G$ , is

$$\begin{aligned} v_G &= \frac{W_G}{(\rho_G A)} \\ &= \frac{1444}{(0.135)(0.0233)(3600)} \\ &= 127.52 \text{ ft/s} \end{aligned} \quad (4-315)$$

(continued)

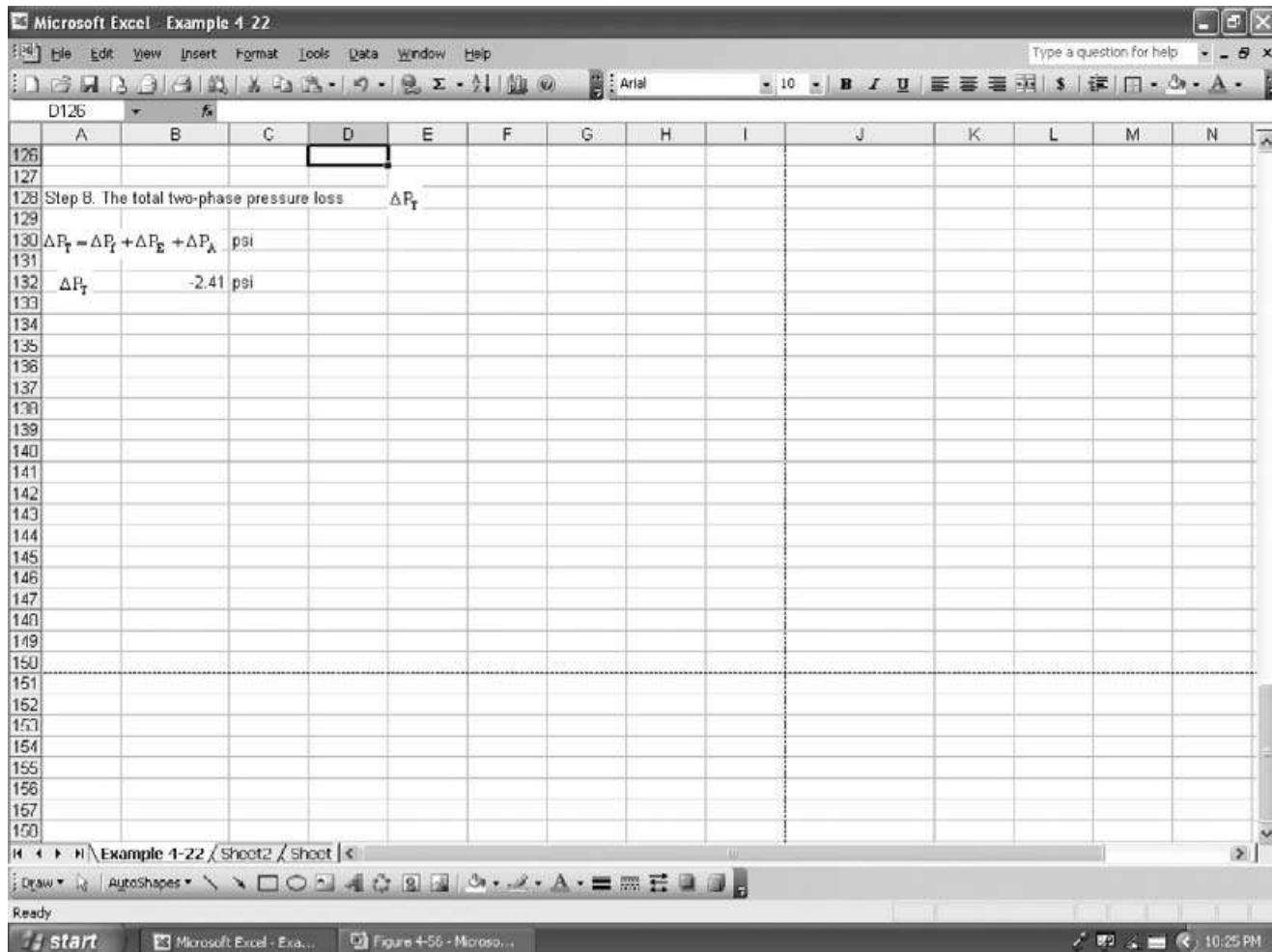


Figure 4-53—(continued)

**EXAMPLE 4-23—(continued)**  
Froude number for liquid phase is

$$\text{FRNL} = \frac{v_L}{(gD)^{0.5}} \left[ \frac{\rho_L}{\rho_L - \rho_G} \right]^{0.5} \quad (4-316)$$

$$= \frac{(1.34)}{\sqrt{(32.2)(0.17225)}} \left( \frac{61.8}{61.8 - 0.135} \right)^{0.5}$$

$$= 0.57$$

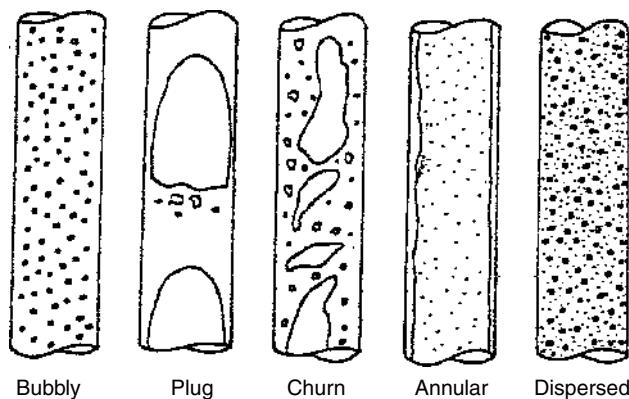
The Froude number for vapor phase is

$$\text{FRNG} = \frac{v_G}{(gD)^{0.5}} \left[ \frac{\rho_G}{\rho_L - \rho_G} \right]^{0.5} \quad (4-317)$$

$$= \frac{127.5}{\sqrt{(32.2)(0.17225)}} \left( \frac{0.135}{61.8 - 0.135} \right)^{0.5}$$

$$= 2.53$$

Since the Froude number for liquid phase is greater than 0.31 and less than 1.0, the 2-in. pipe can produce a pulse flow which may result in pipe vibration. The Excel spreadsheet Example 4-23.xls calculates the Froude numbers for 4 and 6-in. pipe sizes. Table 4-42 shows typical computed results for 2-, 4, and 6-in. pipe sizes.



**Figure 4-54** Flow patterns in vertical liquid–gas flow. (Source: S.M. Walas, “Chemical Process Equipment – Selection and Design”, Butterworth Publishers, 1988.)

#### 4.69 PRESSURE DROP IN VACUUM SYSTEMS

Vacuum in process systems refers to an absolute pressure that is less than or below the local barometric pressure at the location. It is a measure of the degree of removal of atmospheric pressure to some level between atmospheric barometer and absolute vacuum (which cannot be attained in the real world), but is used for a reference of measurement. In most situations, a vacuum is created by pumping air out of the container (pipe, vessels) and thereby lowering the pressure. See Figure 4-3 to distinguish between vacuum gauge and vacuum absolute.

This method [62] is for applications involving air or steam in cylindrical piping under conditions of (a) turbulent flow, (b) sub-atmosphere pressure, (c) pressure drop is limited to 10% of the final pressure (see comment to follow), and (d) the lower limit for application of the method is

$$W/d = 20 \quad (4-318)$$

where  $W$  is the flow rate in lb/h and  $d$  is the inside pipe diameter in inches. If the above ratio is less than 20, the flow is “streamlined” and the data do not apply.

If the pressure drop is greater than 10% of the final pressure, the pipe length can be divided into sections and the calculations made for each section, maintaining the same criteria of (c) and (d) above.

#### Method [62]

The method solves the equation (see Figure 4-56)

$$\Delta P_{\text{vac}} = \frac{(F_1 C_{D1} C_{T1}) + (F_2 C_{D2} C_{T2})}{P_1} \quad (4-319)$$

where

$\Delta P_{\text{vac}}$  = pressure drop, in. water/100 ft of pipe

$P_1$  = initial pressure, in. Hg abs

$F_1$  = base friction factor (Figure 4-56)

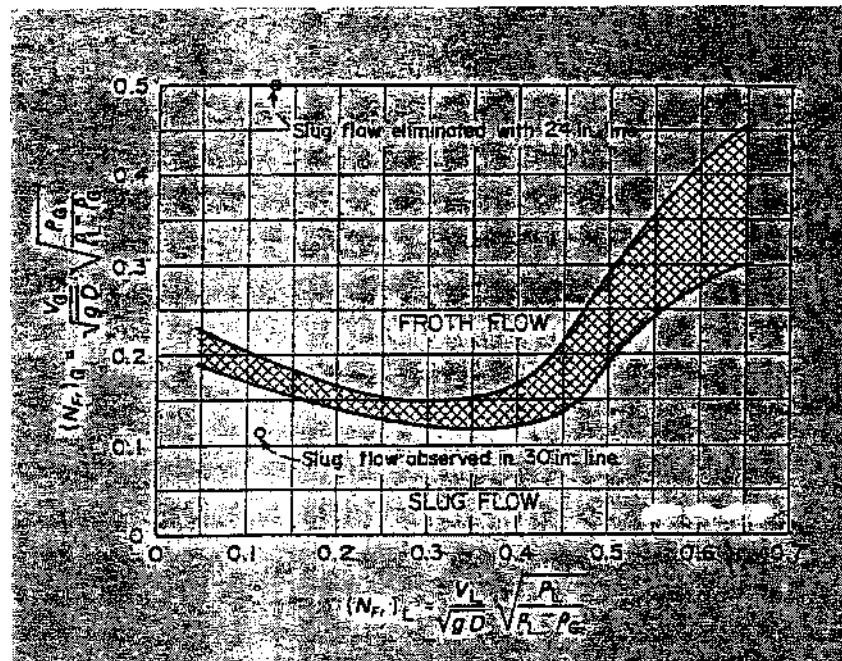
$F_2$  = base friction factor (Figure 4-56)

$C_{T1}$  = temperature correction factor (Figure 4-56)

$C_{T2}$  = temperature correction factor (Figure 4-56)

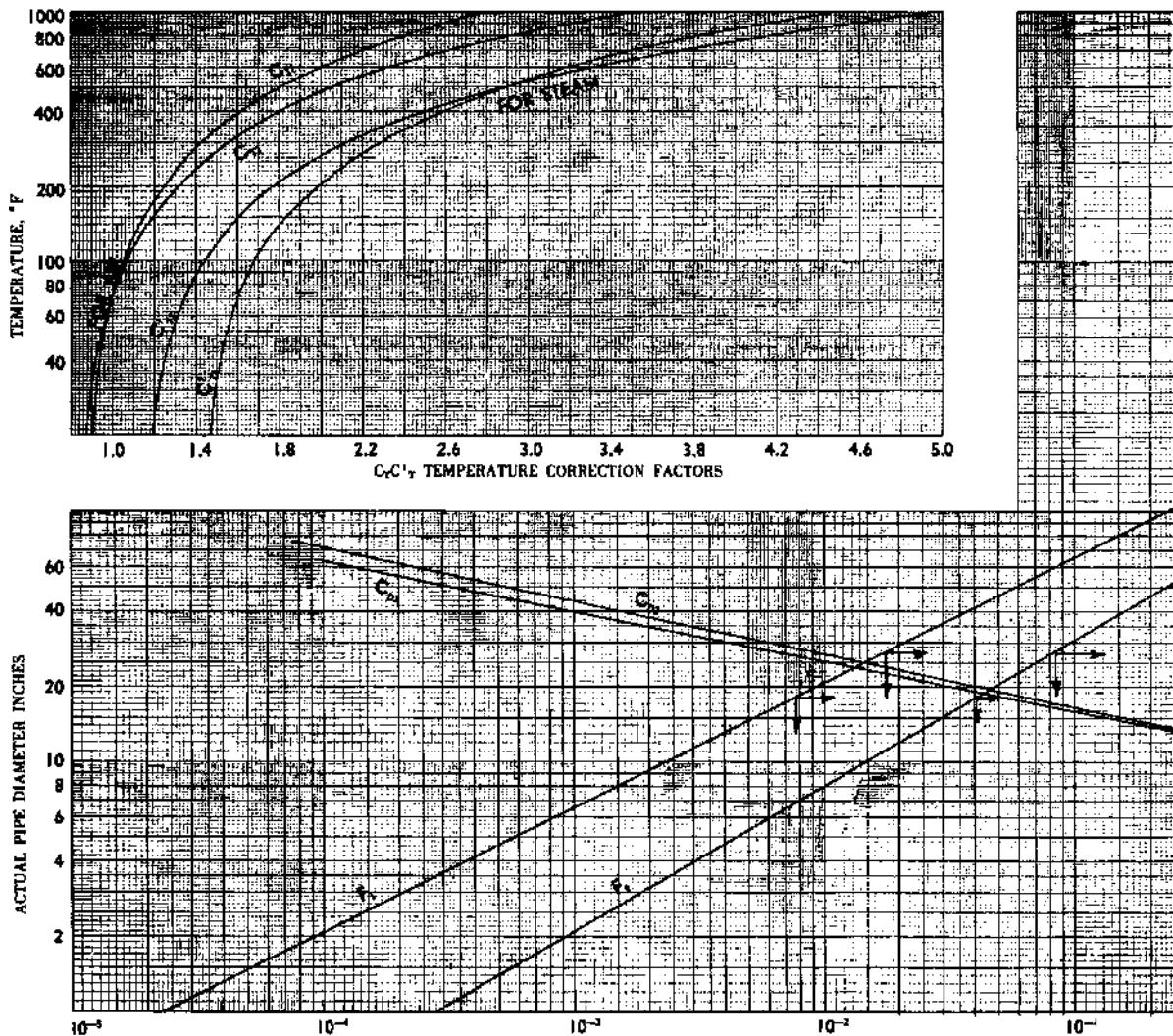
$C_{D1}$  = diameter correction factor (Figure 4-56)

$C_{D2}$  = diameter correction factor (Figure 4-56).



**Figure 4-55** Slug/Froth transition in cocurrent vertical upflow of air–water mixtures in terms of Froude numbers. (Source: L.L. Simpson, *Chem. Eng.* Jun 17, 1968, pp. 192–214.)

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Note: Friction Factors  $F_1$  and  $F_2$  are based on rate of flow, while Factors  $C_{D1}$  and  $C_{D2}$  are based on actual pipe diameter.

**Figure 4-56** Evaluation curves for friction losses of air steam flowing turbulently in commercial pipe at low pressures. (By permission from *Standards for Steam Jet Ejectors*, 4th ed., Heat Exchange Institute, 1988.)

**TABLE 4-42 Gas-Liquid Two-Phase Downflow**

Pipe Internal Diameter (in.)	2.067	4.026	6.065
Liquid flow rate, lb/h	6930.0	6930.0	6930.0
Liquid density, lb/ft <sup>3</sup>	61.80	61.80	61.80
Gas flow rate, lb/h	1444.0	1444.0	1444.0
Gas density, lb/ft <sup>3</sup>	0.135	0.135	0.135
Pipe area, ft <sup>2</sup>	0.023	0.088	0.201
Liquid velocity, ft/s	1.337	0.352	0.155
Gas velocity, ft/s	127.504	33.609	14.810
Froude number for liquid phase	0.5682	0.1073	0.0385
Froude number for gas phase	2.5332	0.4784	0.1718
	Flow is pulse and this may result in pipe vibration.	Line is self-venting. Therefore no vibration problems would be expected.	Line is self-venting. Therefore no vibration problems would be expected.

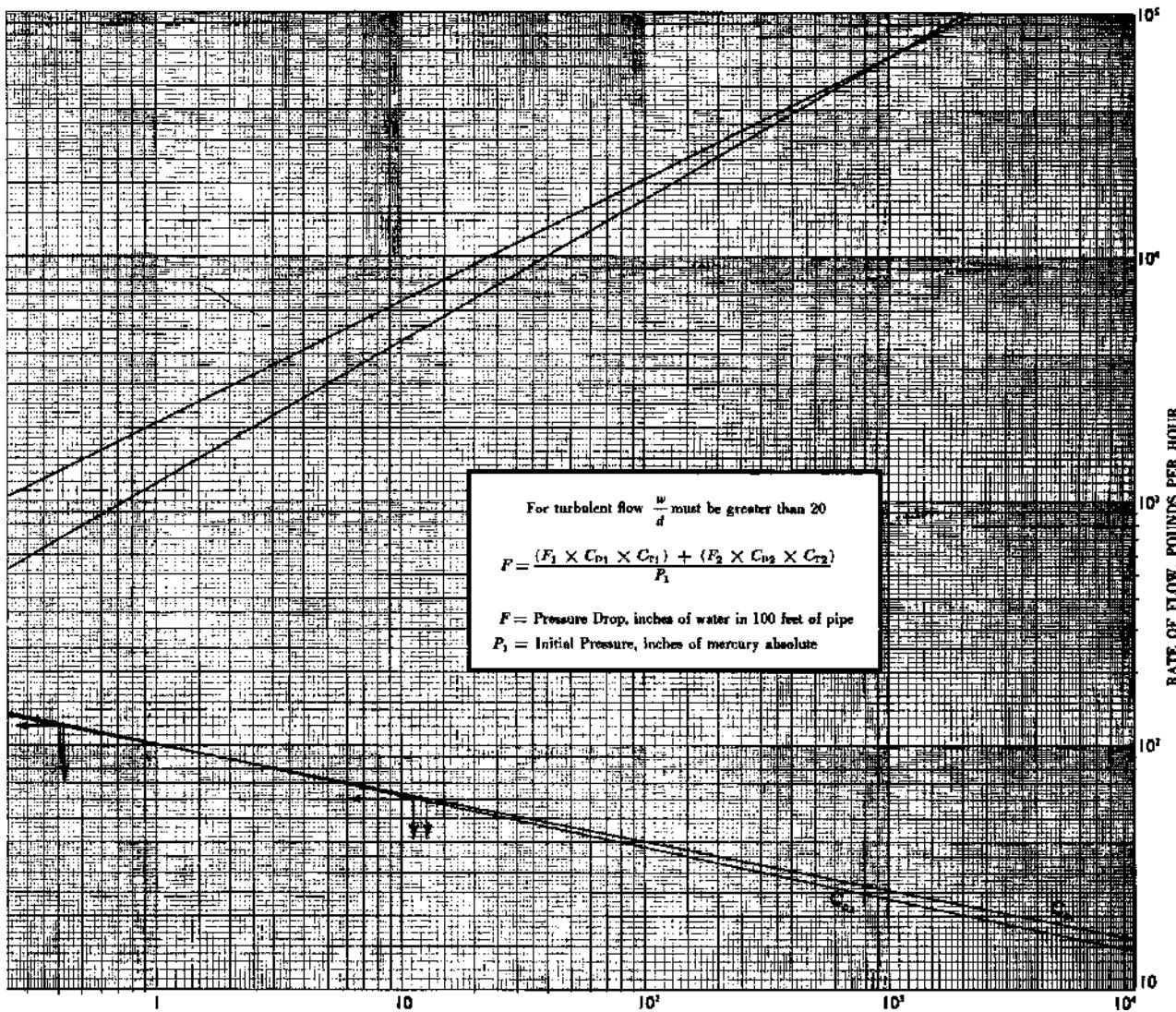


Figure 4-56—(continued)

**EXAMPLE 4-24****Line Sizing for Vacuum Conditions**

Determine the proper line size for a 350 equivalent feet vacuum jet suction line drawing air at 350°F, at a rate of 255 lb/h with an initial pressure at the source of 0.6 in.Hg abs. Assume 10-in. pipe reading (Figure 4-56). Note: Watch scales carefully.

$$\begin{aligned} F_1 &= 0.0155 \\ F_2 &= 0.071 \\ C_{D1} &= 0.96 \\ C_{D2} &= 0.96 \\ C_{T1} &= 1.5 \\ C_{T2} &= 1.67 \end{aligned}$$

$$\begin{aligned} \Delta P_{vac} &= [(0.0155)(0.96)(1.5) \\ &\quad + (0.071)(0.96)(1.67)]/0.6 \end{aligned} \tag{4-319}$$

$$= (0.02232 + 0.1138)/0.6$$

$$= 0.2269 \text{ in. water}/100 \text{ ft}$$

Total line pressure drop:

$$\begin{aligned} \Delta P_{vac} &= \left( \frac{0.2269}{100} \right) (350) = 0.794 \text{ in. water (for 350')} \\ &= (0.794/13.6) = 0.0584 \text{ in.Hg} \end{aligned}$$

Final calculated pressure =  $0.6 + 0.0584 = 0.6584 \text{ in.Hg}$   
 10% of 0.658 = 0.0658 in.Hg

Therefore the system is applicable to the basis of the method, since the calculated pressure drop is less than 10% of the final pressure, and  $w/d = 25.5$ , which is  $> 20$ .

## 4.70 LOW ABSOLUTE PRESSURE SYSTEMS FOR AIR [62]

For piping with air in streamline flow at absolute pressures in the range between  $50\mu$  and 1 mm Hg, the following is a recommended method. Calculation procedures in pressure regions below atmospheric are very limited and often not generally applicable to broad interpretations.

For this method to be applicable, the pressure drop is limited to 10% of the final pressure.

### Method [62]

Refer to Figure 4-57 for low pressure friction factor and air viscosity of Figure 4-58 to correspond to Figure 4-57.

$$P'_1 - P'_2 = \frac{4fL\rho v^2}{2gD(144)}, \text{ psi} \quad (4-127)$$

where

$P'_1$  = upstream static pressure, psia

$P'_2$  = downstream static pressure, psia

$f$  = friction factor, from Figure 4-57

$L$  = length of pipe (total equivalent), ft, including valves and fittings

$\rho$  = average density, lb/ft<sup>3</sup>

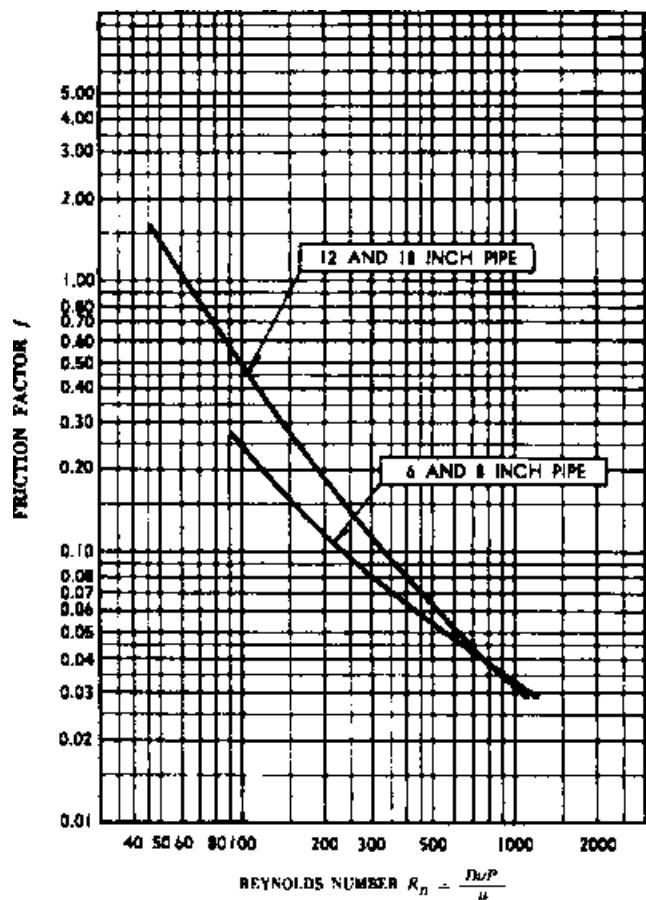


Figure 4-57 Friction factor for streamlined flow of air at absolute pressures from  $50\mu$ Hg. to 1 mmHg. (By permission from *Standards for Steam Jet Ejectors*, 3rd ed. Heat Exchange Institute, 1956 [62] and *Standards for Steam Jet Vacuum Systems*, 4th ed. 1988.) Note:  $f$  on same as Figure 4-5 [63].

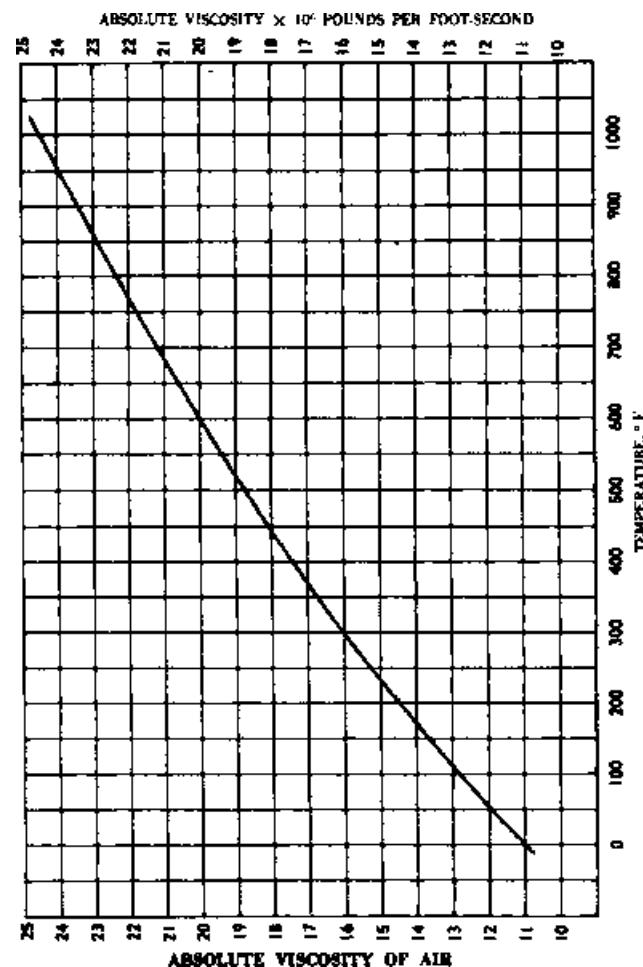


Figure 4-58 Absolute viscosity of air. (By permissions from *Standards for Steam Jet Ejectors*, 3rd ed., Heat Exchange Institute, 1956 [62]; also, *Standards for Steam Jet Vacuum Systems*, 4th ed., 1988 [38].)

$v$  = average velocity, ft/s

$g$  = acceleration due to gravity,  $32.2 \text{ ft/s}^2$

$D$  = inside diameter of pipe, ft

$\mu$  = viscosity of air, abs lb/ft s.

## 4.71 VACUUM FOR OTHER GASES AND VAPORS

Ryans and Roper categorize [64] vacuum in process systems as

Category	Absolute Vacuum (Absolute Pressure)
Rough vacuum	760–1 torr
Medium vacuum	$1-10^{-3}$ torr
High vacuum	$10^{-3}-10^{-7}$ torr
Ultra high vacuum	$10^{-7}$ torr and below

The majority of industrial chemical and petrochemical plants' vacuum operations are in the range of  $100\mu$ –760 torr. This is practically speaking the rough vacuum range noted above. For reference:

1 torr = 1 mmHg

1 in.Hg = 25.4 torr

$1(\mu\text{mHg}) = 0.0010$  torr.

For other conversions, see Appendix D.

In general, partially due to the size and cost of maintaining vacuum in a piping system, the lines are not long (certainly not transmission lines), and there is a minimum of valves, fittings, and bends to keep the resistance to flow low. The procedure recommended by [64] is based on the conventional gas flow equations, with some slight modifications. The importance in final line size determination is to determine what is a reasonable pressure loss at the absolute pressure required and the corresponding pipe size to balance these. In some cases a trial-and-error approach is necessary.

#### Method [64] (by permission)

- Convert mass flow rate to volumetric flow rate,  $q_m$

$$q_m = W(359/M)(760/P_t)(T/(32+460))(1/60), \text{ ft}^3/\text{min} \quad (4-320)$$

where

$P_t$  = pressure, torr

$T$  = temperature, °R

$W$  = mass flow, lb/h

$M$  = molecular weight.

- Calculate section by section from the process vessel to the vacuum pump (point of lowest absolute pressure)
- Assume a velocity  $v$ , ft/s, consistent with Figure 4-59. Use Table 4-43 for short, direct connected connections to the vacuum pump. Base the final specifications for the line on pump specifications. Also the diameter of the line should match the inlet connection for the pump. General good practice indicates that velocities of 100–200 ft/s are used, with 300–400 ft/s being the upper limit for the rough vacuum classification.

Sonic velocity,  $v_s = [\text{kg } (1544/M) T]^{1/2}$ , ft/s

Use  $v$  from Figure 4-59, and  $q_m$  from Eq. (4-320).

- Determine pipe diameter,  $D$ , ft

$$D = 0.146\sqrt{q_m/v} \quad (4-321)$$

Round this to the nearest standard pipe size. Recalculate  $v$  based on actual internal diameter of the line.

- Determine Reynolds Number,  $Re$

$$Re = \rho D v / \mu_e \quad (4-15)$$

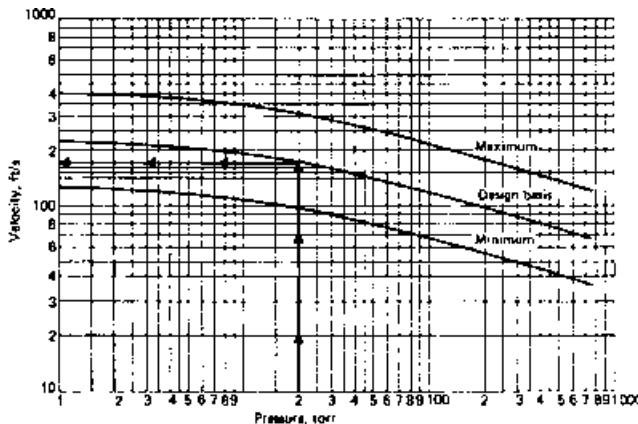


Figure 4-59 Typical flow velocities for vacuum lines. Note: 1 torr = 1.33 mb = 133.3 Pa 1 ft/s = 0.3048 m/s. (By permission from Ryans, J.L. and D.L. Roper, *Process Vacuum System Design and Operation*, McGraw-Hill Book Co., Inc., 1986 [64].)

TABLE 4-43 Criteria for Sizing connecting Lines in VAccum Service

Vacuum Pump	Assumed Flow Velocity (ft/s)
<i>Steam jet</i>	
System pressure, torr	
0.5–5	300
5–25	250
25–150	200
150–760	150
<i>Liquid ring pump</i>	
Single-stage*	100
Two-Stage	150
<i>Rotary piston</i>	
Single-stage	50
Two-stage	25
<i>Rotary vane</i> †	
Single-stage	200
Two-stage	400
<i>Rotary blowers</i>	
Atmospheric discharge	50
Discharging to backing pump	100

(Source: By permission from Ryans, J.L. and Roper, D.L., *Process Vacuum Systems Design and Operation*, McGraw-Hill Book Co. Inc., 1986 [18].)

\* Assumes the pump features dual inlet connections and that an inlet manifold will be used.

† Based on rough vacuum process pumps. Use 25 ft/s for high vacuum pumps.

$\rho$  = density, lb/ft<sup>3</sup> at flowing conditions

$D$  = pipe inside diameter, ft

$v$  = vapor velocity (actual), ft/s

$\mu_e$  = viscosity of vapor, lb/ft s.

- Determine friction factor,  $f$ , from Moody Friction Factor Charts (Figure 4-5) or Chen's explicit equation  
Or, calculate for turbulent flow using Blasius' equation [64]:

$$f = 0.316/(Re)^{1/4}, \text{ for } Re < 2.0 \times 10^5$$

(smooth pipe only)

- Tabulate the summation of equivalent lengths of straight pipe, valves, fittings, and entrance/exit losses as presented in earlier sections of this chapter

- Calculate the pressure drop for the specific line section (or total line) from

$$\Delta P_T = 0.625 \rho_i f_D L q_m^2 / d^5, \text{ torr} \quad (4-322)$$

or

$$\Delta P_T = 4.31 \rho_i f_D L v^2 / 2gd, \text{ torr} \quad (4-323)$$

where

$\rho$  = density, lb/ft<sup>3</sup>

$d$  = pipe inside diameter, in.

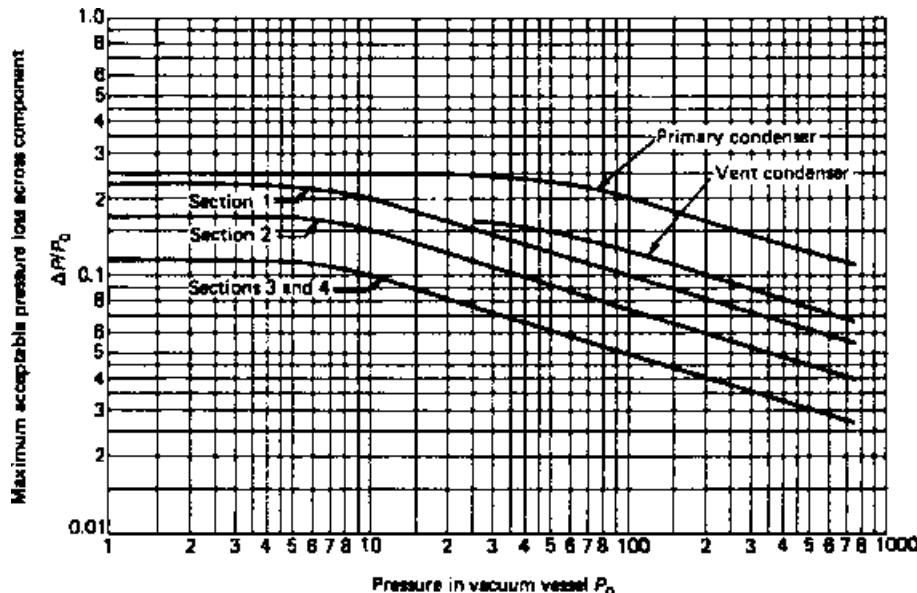
$q_m$  = volumetric flow rate, ft<sup>3</sup>/min

$f_D$  = friction factor (Moody-Darcy), Figure 4-5

$\Delta P_T$  = pressure drop, torr.

Calculate:

$$\rho_i = P_i M / 555 T_i, \text{ lb/ft}^3 \quad (4-324)$$



**Figure 4-60** Acceptable pressure losses between the vacuum vessel and the vacuum pump. Note: Reference sections on figure to system diagram to illustrate the sectional type hook-ups for connecting lines. Use 60% of the pressure loss read as acceptable loss for the system from process to vacuum pump, for initial estimate,  $P$  = pressure drop (torr) of line in question;  $P_0$  = operating pressure of vacuum process equipment, absolute, torr. (By permission from Ryans, J.L. and D.L. Roper *Process Vacuum System Design and Operation*, McGraw-Hill Book Co., Inc., 1986 [64].)

$P_i$  = pressure, torr

$M$  = average molecular weight of mixture flowing

$T_i$  = temperature, °R.

9. If the calculated pressure drop does not exceed the maximum given in Figure 4-60, use this calculated value to specify the line. If the  $\Delta P$  exceeds the limit of Figure 4-60, increase the pipe size and repeat the calculations until an acceptable balance is obtained. For initial estimates, the authors [64] recommend using 0.6 times the value obtained from Figure 4-60 for an acceptable pressure loss between vessel and the pump.

The suction pressure required at the vacuum pump (in absolute pressure) is the actual process equipment operating pressure minus the pressure loss between the process equipment and the source of the vacuum. Note that absolute pressures must be used for these determinations and not gauge pressures. Also keep in mind that the absolute pressure at the vacuum pump must always be a lower absolute pressure than the absolute pressure at the process.

#### 4.72 PIPE SIZING FOR NON-NEWTONIAN FLOW

Non-Newtonian fluids vary significantly in their properties that control flow and pressure loss during flow from the properties of Newtonian fluids. The key factors influencing non-Newtonian fluids are their shear thinning or shear thickening characteristics and time dependency of viscosity on the stress in the fluid.

Most conventional chemical and petrochemical plants do not process many, if any, non-Newtonian fluids. However, polymers, grease, heavy oils, cellulose compounds, paints, fine chalk suspensions in water, some asphalts, foods, cosmetics, inks, waste sludge, and other materials do exhibit one type or another of the characteristics of non-Newtonians, classified as:

- Bingham plastics
- Dilatant

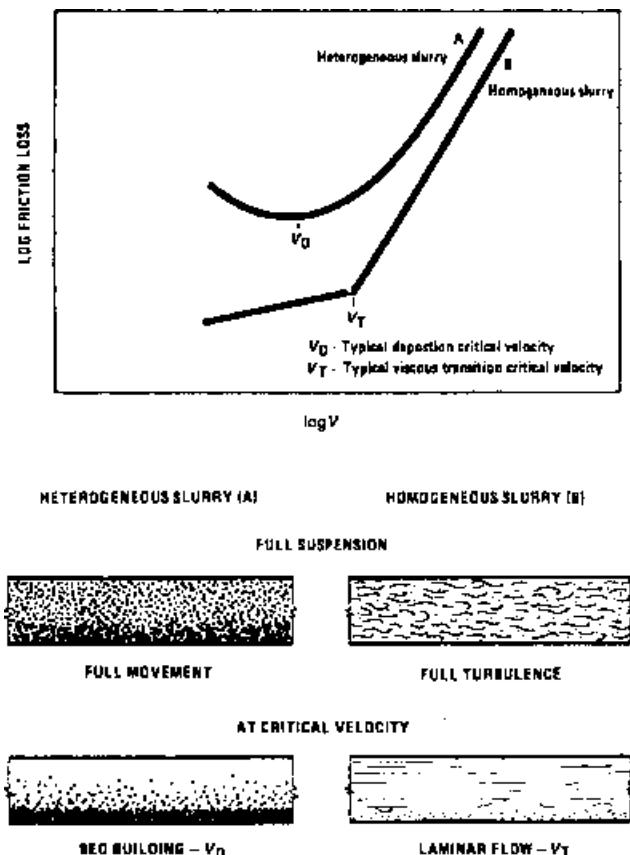
- Pseudoplastic
- Yield pseudoplastics.

Solving these classes of flow problems requires specific rheological data on the fluid, which is often not in the public literature, or requires laboratory determinations using a rotational or other type of viscometer. The results do not allow use of the usual Fanning or Moody friction charts and are beyond the scope of this chapter. Design literature is very limited, with some of the current available references being Sultan [65], Darby [5], Cheremisinoff, N.P. and Gupta [66], Perry et al. [1], Brodkey and Hershey [67], Darby and Chang [68], Darby and Nelson [69], and Durand et al. [70].

#### 4.73 SLURRY FLOW IN PROCESS PLANT PIPING

Most industrial process plants have from none to a few slurry flow lines to transport process fluids. The more common slurry lines discussed in the literature deal with long transmission lines for coal/water, mine tailings/water, limestone/water, wood pulp-fibers/water, gravel/water, and others. These lines usually can be expected to have flow characteristics somewhat different than in-plant process slurries. Considerable study has been made of the subject, with the result that the complexity of the variables makes correlation of all data difficult, especially when dealing with short transfer lines. For this reason, no single design method is summarized here, but rather reference is given to the methods that appear most promising (also see [71]).

Derammelaere and Wasp [72] present a design technique that ties into their classification of slurries as heterogeneous and homogeneous (Figures 4-61 and 4-62). This method uses the Fanning friction factor and conventional equations for pressure drop. The recommended design slurry velocities range from 4 to 7 ft/s. Pipe abrasion can be a problem for some types of solids when the velocity approaches 10 ft/s. For velocities below 4 ft/s there can be a tendency for solids to settle and create blockage and plugging of the line.



**Figure 4-61** Critical velocity characteristics depend on whether slurry is heterogeneous or homogeneous. (By permission from Derammelaere, R.H. and E.J. Wasp, "Fluid Flow, Slurry Systems and Pipelines", *Encyclopedia of Chemical Processing and Design*, J. McKetta, ed., Marcel Dekker Vol. 22, 1985 [72].)

The concentration of the solids as well as particle size and liquid and solid densities in the slurry determines the slurry rheology or viscosity. The two principal classifications are [72]:

1. Newtonian slurries are simple rheologically with a constant (property) viscosity, and can be treated as true fluids as long as the flowing velocity is sufficient to prevent the dropout of solids. For this type of slurry, the viscosity =  $\mu$ .
2. Bingham-plastic slurries require a shear stress diagram showing shear rate vs. shear stress for the slurry in order to determine the coefficient of rigidity (apparent viscosity),  $\eta$ , which is the slope of the line from the origin to the plot at a particular concentration. This is laboratory data requiring a rheometer. These are usually fine solids at high concentrations.

Derammelaere and Wasp [72] have two practical in-plant design examples worked out. The pressure drop design method of Turian and Yuan [30] is the development of the analysis of a major literature data review. The method categorizes slurry flow regimes similar in concept to the conventional multi-regime diagram for two-phase flow (Figure 4-63). Their friction factor correlations are specific to the calculated flow regime. See Figure 4-64 for one of four typical plots in the original reference.

Example calculations are included, and Figure 4-65 illustrates the effect of pipe size on the placement of the flow regime.

#### 4.74 PRESSURE DROP FOR FLASHING LIQUIDS

When a liquid is flowing near its saturation point (e.g., the equilibrium or boiling point) in a pipe line, decreased pressure will cause vaporization. The higher the pressure difference, the greater the vaporization resulting in flashing of the liquid. Steam condensate lines cause a two-phase flow condition, with hot condensate flowing to a lower pressure through short and long lines. For small lengths with low pressure drops, and the outlet end being a few pounds per square inch of the inlet, the flash will be assumed as a small percentage. Consequently, the line can be sized as an all liquid line. However, caution must be exercised as 5% flashing can develop an important impact on the pressure drop of the system [5].

Sizing of flashing steam condensate return lines requires techniques that calculate pressure drop of two-phase flow correlations. Many correlations have been presented in the literature [49, 50, 53, and 73]. Most flow patterns for steam condensate headers fall within the annular or dispersed region on the Baker map. Sometimes, they can fall within the slug flow region; however the flashed steam in steam condensate lines is less than 30% by weight.

Steam is the most common liquid that is flashed in process plants, but, of course, it is not the only one as many processes utilize flash operations of pure compounds as well as mixtures. Although this presentation is limited to steam, the principles apply to other materials.

Steam condensate systems often are used to generate lower pressure steam by flashing to a lower pressure. When this occurs, some steam is formed and some condensate remains, with the relative quantities depending upon the pressure conditions. Figure 4-66 is a typical situation.

Percent incoming condensate flashed to steam:

$$\% \text{ flash} = \frac{(h_1 - h_2) 100}{L_v} \quad (4-325)$$

where

$h_1$  = enthalpy of liquid at higher pressure, Btu/lb

$h_2$  = enthalpy of liquid at lower or flash pressure, Btu/lb

$L_v$  = latent heat of evaporation of steam at flash pressure, Btu/lb.

#### EXAMPLE 4-25

##### Calculation of Steam Condensate Flashing

There are 79,500 lb/h of 450 psig condensate flowing into a flash tank. The tank is to be held at 250 psig, generating steam at this pressure. Determine the quantity of steam produced.

Enthalpy of liquid at 450 psig = 441.1 Btu/lb

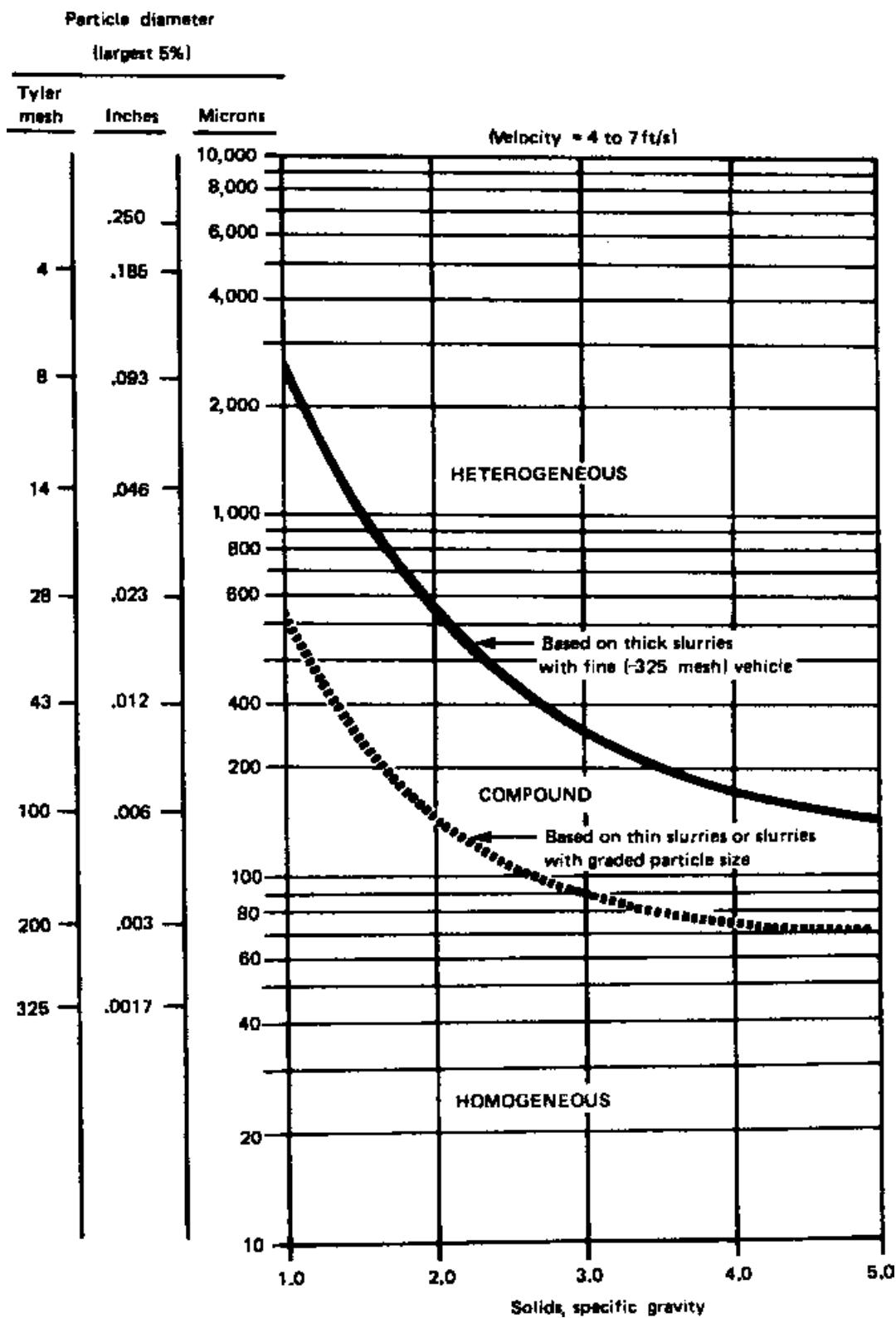
Enthalpy of liquid at 250 psig = 381.6 Btu/lb

Latent heat of vaporization at 250 psig = 820.1 Btu/lb.

$$\% \text{ flash into steam} = \frac{441.1 - 381.6}{820.1} (100) = 7.25\% \quad (4-325)$$

Steam formed =  $(0.0725)(79,500) = 5764 \text{ lb/h}$

Condensate formed =  $79,500 - 5764 = 73,736 \text{ lb/h}$ .



**Figure 4-62** Slurry flow regime (heterogeneous, homogeneous) is a function of soild's size and specific gravity. (By permission from Derammelaere, R.H. and E.J. Wasp, "Fluid Flow, Slurry Systems and Pipelines", *Encyclopedia of Chemical Processing and design*, J. McKetta, ed., Marcel Decker, Vol. 22, 1985 [72].)

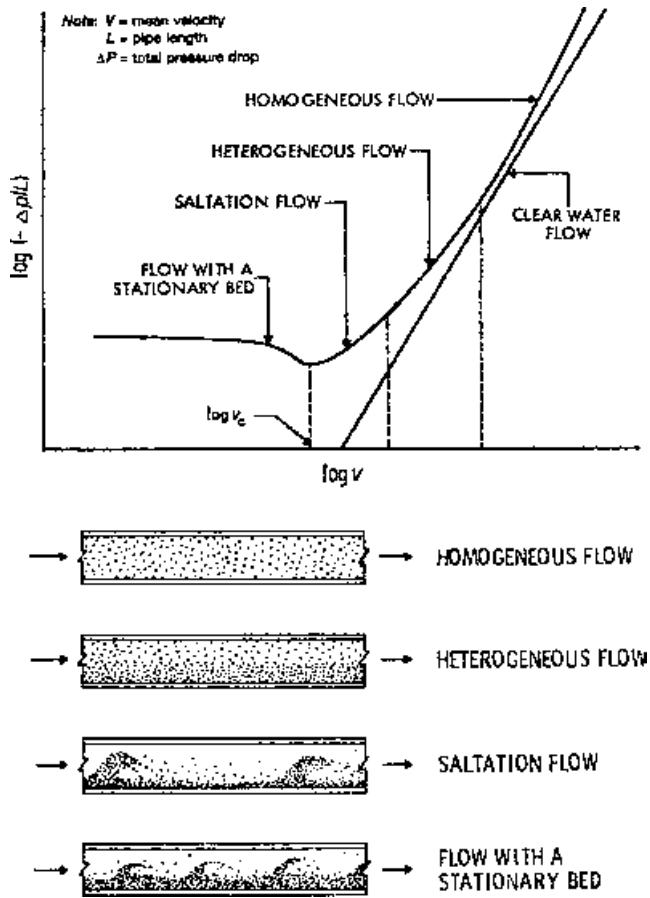


Figure 4-63 Representative plot of pressure drop for slurry flow. (By permission from Turian, R.M. and T.F. Yuan, "Flow of Slurries in Pipelines", *A.I.Ch.E.J.*, Vol. 23, 1977, p. 232-243.)

#### 4.75 SIZING CONDENSATE RETURN LINES

Steam condensate lines usually present a two-phase flow condition, with hot condensate flowing to a lower pressure through short and long lines. As the flow progresses down the pipe, the pressure falls and flashing of condensate into steam takes place continuously.

Calculation of condensate piping by two-phase flow techniques is recommended; however, the tedious work per line can often be reduced by using empirical methods and charts. Some of the best methods are proprietary and not available for publication; however, the Sarco method [74] has been used and found to be acceptable, provided no line less than  $1\frac{1}{2}$  in. is used regardless of the chart reading. Under some circumstances, which are too random to properly describe, the Sarco method may give results too small by possibly a half pipe size. Therefore, latitude is recommended in selecting either the flow rates or the pipe size.

#### 4.76 DESIGN PROCEDURE USING SARCO CHART [74]

- Establish upstream or steam pressure from which condensate is being produced and discharged into a return line through steam traps, or equivalent, psig.
- Establish the steam condensate load or rate in lb/h flow.
- Establish the pressure of the condensate return line, psig.

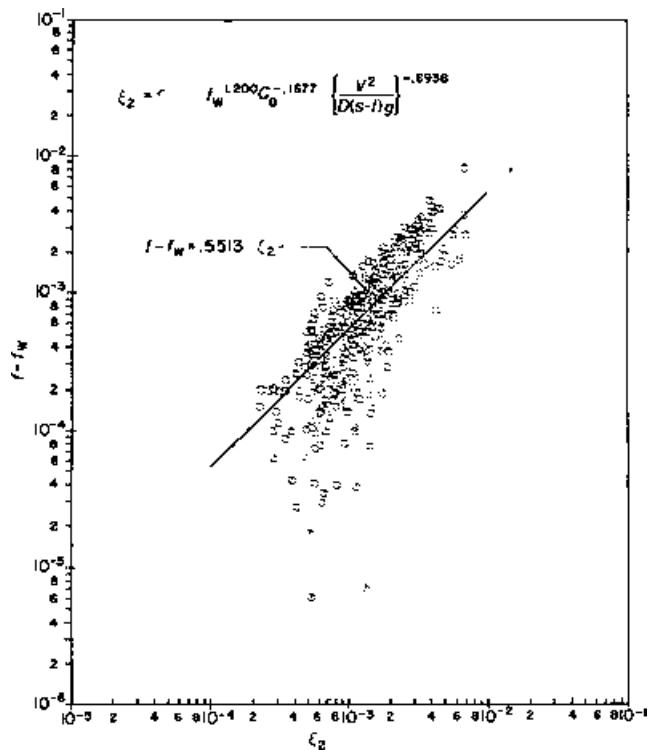
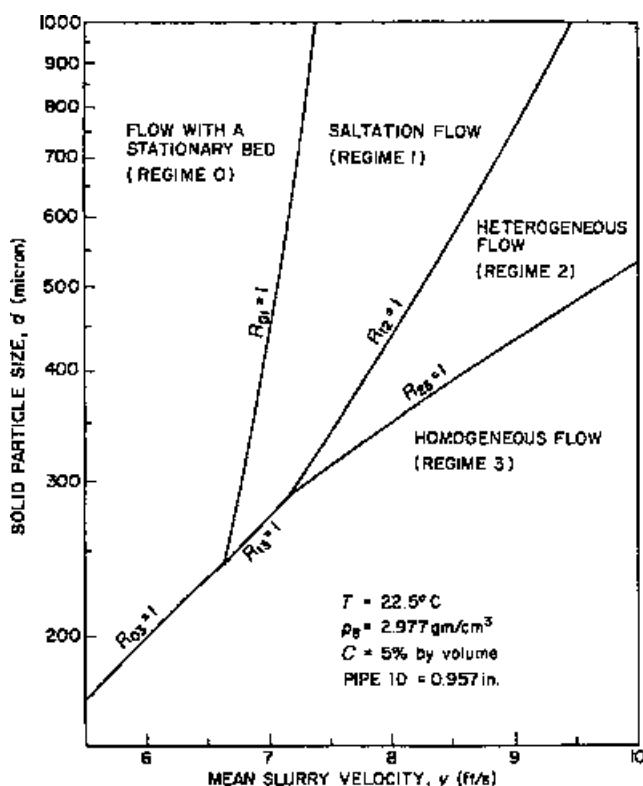


Figure 4-64 Friction factor correlation for slurry flow in heterogeneous flow regime. (By permission from Turian, R.M. and T.F. Yuan, "Flow of Slurries in Pipelines", *A.I.Ch.E.J.*, Vol. 23, 1977, pp. 232-243.)

- The method is based on an allowable 5000 ft/min velocity in the return line (mixture).
- Calculate load factor:
$$= \frac{5000(100)}{\text{Condensate Rate, lb/h}} = \frac{500,000}{C} \quad (4-326)$$
- Establish condensate receiver (or flash tank) pressure, psig.
- Referring to Figure 4-67, enter at steam pressure of (1) above, move horizontally to condensate receiver pressure of (6) above, and then up vertically to the "factor scale."
- Divide the load factor (step 1) by the value from the "factor scale" of (7) above, obtain ft/min/(100 lb/h load).
- Enter chart on horizontal velocity line, go vertically up to the steam pressure of (1) above, and read pipe size to the next largest size if the value falls between two pipe sizes.
- For pipe sizes larger than 3 in., follow the steps (1) thru (8) above. Then enter the vertical scale at the steam pressure of (1) above, and move to the 3-in. pipe size and down to the horizontal velocity scale.
- Divide the result of step (8) above by the result of step (10).
- Refer to the large pipe multipliers shown in the table on the chart, and select the pipe size whose factor is equal to or smaller than the result of step (11) above. This is the pipe size to use, provided a sufficient factor of safety has been incorporated in the data used for the selection of pipe size.
- Calculation of "factor scale" for receiver pressures different than those shown on chart:

$$\text{factor} = \frac{36.2(\bar{V})(h_p - h_r)}{L_v(h_p - 180)} \quad (4-327)$$



**Figure 4-65** Flow regime diagram for solid–water in 1-in. PVC pipe. (By permission from Turian, R.M. and T.F. Yuan, "Flow of Slurries in Pipelines", *A.I.Ch.E Journal*, Vol. 23, 1977, pp. 232–243.)

where

$$\begin{aligned}\bar{V} &= \text{specific volume of steam at return line pressure, ft}^3/\text{lb} \\ h_p &= \text{enthalpy of liquid at supply steam pressure, Btu/lb} \\ h_r &= \text{enthalpy of liquid at return line pressure, Btu/lb} \\ L_v &= \text{latent heat of evaporation at return line pressure, Btu/lb.}\end{aligned}$$

Use the factor so calculated just as if read from the chart, that is, in step (8) above.

## 4.77 FLOW THROUGH PACKED BEDS

Flow of fluids through packed beds of granular particles occurs frequently in chemical processes. Examples are flow through a fixed-bed catalytic reactor, flow through a filter cake, and flow through an absorption or adsorption column. An understanding of flow through packed beds is also important in the study of sedimentation and fluidization.

An essential factor that influences the design and operation of a dynamic catalytic or adsorption system is the energy loss (pressure drop). Factors determining the energy loss are many and investigators have made simplifying assumptions or analogies so that they could utilize some of the general equations. These equations represent the forces exerted by the fluids in motion (molecular, viscous, kinetic, static, etc.) to arrive at a useful expression correlating these factors.

Ergun [78] developed a useful pressure drop equation caused by simultaneous kinetic and viscous energy losses, and applicable to all types of flow. Ergun's equation relates the pressure drop per unit of bed depth to dryer or reactor system characteristics, such as velocity, fluid gravity, viscosity, particle size, shape,

### EXAMPLE 4-26

#### Sizing Steam Condensate Return Line

A 450 psig steam system discharges 9425 lb/h of condensate through traps into a return condensate line. The return header is to discharge into a flash tank held at 90 psig. The calculated total equivalent length of pipe, valves, and fittings is 600 ft.

Using the Sarco chart, Figure 4-67, determine the recommended line size for the return line.

1. Upstream steam pressure = 450 psig
2. Condensate load = 9425 lb/h
3. Return line pressure = 90 psig
4. Use the Sarco recommendation of 5000 ft/min
5. Load factor

$$= \frac{(5000)(100)}{9425} = 53.05$$

6. Receiver pressure = 90 psig
7. Refer to Figure 4-67 and note that required receiver pressure is not shown, so calculate "factor scale" by previous formula:

$$\begin{aligned}h_p &= 441 \text{ Btu/lb at } 450 \text{ psig} \\ h_r &= 302 \text{ Btu/lb at } 90 \text{ psig}\end{aligned}$$

$$L_v = 886 \text{ Btu/lb at } 90 \text{ psig}$$

$$\bar{V} = 4.232 \text{ ft}^3/\text{lb at } 90 \text{ psig.}$$

$$\text{"factor scale" value} = \frac{36.2(4.232)(441 - 302)}{886(441 - 180)} = 0.092 \quad (4-327)$$

$$8. \text{ Ft/min/100#/h} = \frac{53.05}{0.092} = 576.1$$

9. Read Chart: At 450 psig and 576.1, the line size shows just under 2 in. Recommended use 2 in.

The procedure for using the convenient chart Figure 4-68 [76] is, for example:

*Step 1:* Enter the figure at 600 psig below the insert near the right-hand side, and read down to the 200 psig end pressure.

*Step 2:* Proceed left horizontally across the chart to the intersection with

*Step 3:* The 1000 lb/h flow rate projected diagonally up from the bottom scale.

(continued)

**EXAMPLE 4-26—(continued)**

*Step 4:* Reading vertically up from this intersection, it can be seen that a 1-in. line will produce more than the allowed pressure drop, so a  $1\frac{1}{2}$ -in. size is chosen.

*Step 5:* Read left horizontally to a pressure drop of 0.28 psi/100 ft on the left-hand scale.

*Step 6:* Note the velocity given by this line as 16.5 ft/s, then proceed to the insert on the right, and read upward from 600 to 200 psig to find the velocity correction factor as 0.41.

*Step 7:* Multiply 0.41 by 16.5 to get a corrected velocity of 6.8 ft/s.

The comparison between this method and that of Dukler [73] and others gives good agreement for reasonably good cross section of flow regimes.

Because flashing steam-condensate lines represent two-phase flow, with the quantity of liquid phase depending on the system conditions, these can be designed following the previously described two-phase flow methods. Ruskin [76] assumes that a single homogeneous phase of fine liquid droplets is dispersed in the flashed vapor and the pressure drop is calculated using the Darcy equation:

$$\Delta P = \frac{0.00000336 f W^2}{\rho d^5}, \text{ psi} \quad (4-65)$$

Ruskin [76] developed a method for calculating pressure drop of flashing condensate. His method gave pressure drops comparable to those computed by the two-phase flow in good agreement with experimental data. The method employed here is based on a similar technique given by Ruskin. The pressure drop for flashing steam uses the average density of the resulting liquid-vapor mixture after flashing. In addition, the friction factor used is valid for complete turbulent flows in both commercial steel and wrought iron pipe. The pressure drop assumes that the vapor-liquid mixture throughout the condensate line is represented by mixture conditions near the end of the line. This assumption is valid since most condensate lines are sized for low pressure drop, with flashing occurring at the steam trap or valve close to the pipe entrance. If the condensate line is sized for a higher pressure drop, an iterative method must be used. For this case, the computations start at the end of the pipeline and proceed to the steam trap.

The method employed determines the following:

1. The amount of condensate flashed for any given condensate header from 15 to 140 psia. Initial steam pressure may vary from 40 to 165 psia.
2. The return condensate header temperature.
3. The pressure drop (psi/100 ft) of the steam condensate mixture in the return header.
4. The velocity of the steam condensate mixture and gives a warning message if the velocity is greater than 5000 ft/min, as this may present problems to the piping system.

**The Equations**

The following equations are used to determine the pressure drop for flashed condensate mixture [77].

$$WFRFL = B (\ln P_c)^2 - A \quad (4-328)$$

where

$$A = 0.00671 (\ln P_h)^{2.27} \quad (4-329)$$

$$B = e^x 10^{-4} + 0.0088 \quad (4-330)$$

and

$$X = 6.122 - \left( \frac{16.919}{\ln P_h} \right) \quad (4-331)$$

$$W_G = (WFRFL) (W) \quad (4-332)$$

$$W_L = W - W_G \quad (4-333)$$

$$TFL = 115.68 P_h^{0.226} \quad (4-334)$$

$$\rho_G = 0.0029 P_h^{0.938} \quad (4-335)$$

$$\rho_L = 60.827 - 0.078 P_h + 0.00048 P_h^2 - 0.0000013 P_h^3 \quad (4-336)$$

$$\rho_M = \frac{W_G + W_L}{\left( \frac{W_G}{\rho_G} + \frac{W_L}{\rho_L} \right)} \quad (4-337)$$

For fully turbulent flow

$$f = \frac{0.25}{\left[ -\log \left( \frac{0.000486}{d} \right) \right]^2} \quad (4-338)$$

where  $d$  = pipe diameter, in.

Pressure drop

$$\Delta P_T = \frac{0.000336 f W^2}{d^5 \rho_M} \quad (4-65)$$

$$v = \frac{3.054}{d^2} \left[ \frac{W_G}{\rho_G} + \frac{W_L}{\rho_L} \right] \quad (4-339)$$

If  $v \geq 5000$  ft/min, print a warning message as condensate may cause deterioration of the process pipeline.

where

$d$  = internal pipe diameter, in.

$f$  = friction factor, dimensionless

$P_c$  = steam condensate pressure before flashing, psia

$P_h$  = flashed condensate header pressure, psia

$v$  = velocity of flashed condensate mixture, ft/min

$W$  = total flow of mixture in condensate header, lb/h

$W_G$  = flashed steam flow rate, lb/h

$W_L$  = flashed condensate liquid flow rate, lb/h

$WFRFL$  = weight fraction of condensate flashed to vapor

$TFL$  = temperature of flashed condensate, °F

$\Delta P_T$  = pressure drop of flashed condensate mixture, psi/100 ft

$\rho_L$  = flashed condensate liquid density, lb/ft³

$\rho_G$  = flashed steam density, lb/ft³

$\rho_M$  = density of mixture (flashed condensate/steam), lb/ft³.

**EXAMPLE 4-27**

Determine the pressure drop for the 4-, 6-, and 8-in. (Sch. 40) condensate headers under the following conditions.

Flow rate, lb/h	10,000
Steam condensate pressure, psia	114.7
Header pressure, psia	14.7

*Solution*

For the 4-in. (Sch. 40) pipe size, ID = 4.026 in.

The weight fraction of the condensate is

$$\begin{aligned} X &= 6.122 - \left( \frac{16.919}{\ln P_h} \right) \\ &= 6.122 - \left( \frac{16.919}{\ln 14.7} \right) \\ &= -0.1726 \end{aligned} \quad (4-331)$$

$$\begin{aligned} A &= 0.00671 (\ln P_h)^{2.27} \\ &= 0.0632 \end{aligned} \quad (4-329)$$

$$\begin{aligned} B &= e^{(-0.1726)} (10^{-4}) + 0.0088 \\ &= 0.008884 \end{aligned} \quad (4-330)$$

The weight fraction of the condensate is

$$\begin{aligned} WFRFL &= B (\ln P_c)^2 - A \\ &= 0.008884 (\ln 114.7)^{2.27} - 0.0632 \\ &= 0.1365 \end{aligned} \quad (4-328)$$

$$\begin{aligned} W_G &= (WFRFL)(W) \\ &= 0.1365 \times 10,000 \\ &= 1365 \text{ lb/h} \end{aligned} \quad (4-332)$$

$$\begin{aligned} W_L &= W - W_G \\ &= 10,000 - 1365 \\ &= 8635 \text{ lb/hr} \end{aligned} \quad (4-333)$$

The temperature of the flashed condensate is

$$\begin{aligned} TFL &= 115.68 P_h^{0.226} \\ &= 212.4^\circ \text{F} \end{aligned} \quad (4-334)$$

The flashed steam density, condensate and density of the mixture are

$$\begin{aligned} \rho_G &= 0.0029 P_h^{0.938} \\ &= 0.0029 (14.7)^{0.938} \\ &= 0.0361 \text{ lb/ft}^3 \end{aligned} \quad (4-335)$$

$$\rho_L = 60.827 - 0.078 P_h + 0.00048 P_h^2 - 0.0000013 P_h^3 \quad (4-336)$$

$$\begin{aligned} \rho_L &= 60.827 - 0.078 (14.7) + 0.00048 (14.7)^2 \\ &\quad - 0.00000013 (14.7)^3 \\ &= 59.78 \text{ lb/ft}^3 \end{aligned}$$

$$\begin{aligned} \rho_M &= \frac{W_G + W_L}{\left( \frac{W_G}{\rho_G} + \frac{W_L}{\rho_L} \right)} \\ \rho_M &= \frac{(1365 + 8635)}{\left( \frac{1365}{0.0361} + \frac{8635}{59.78} \right)} \\ &= 0.2634 \text{ lb/ft}^3 \end{aligned} \quad (4-337)$$

Assuming that the flow through the line is turbulent:  
For fully turbulent flow

$$f = \frac{0.25}{\left[ -\log \left( \frac{0.000486}{d} \right) \right]^2} \quad (4-338)$$

$$\begin{aligned} f &= \frac{0.25}{\left[ -\log \left( \frac{0.000486}{4.026} \right) \right]^2} \\ &= 0.01628 \end{aligned}$$

Pressure drop of the steam condensate mixture in the return header:

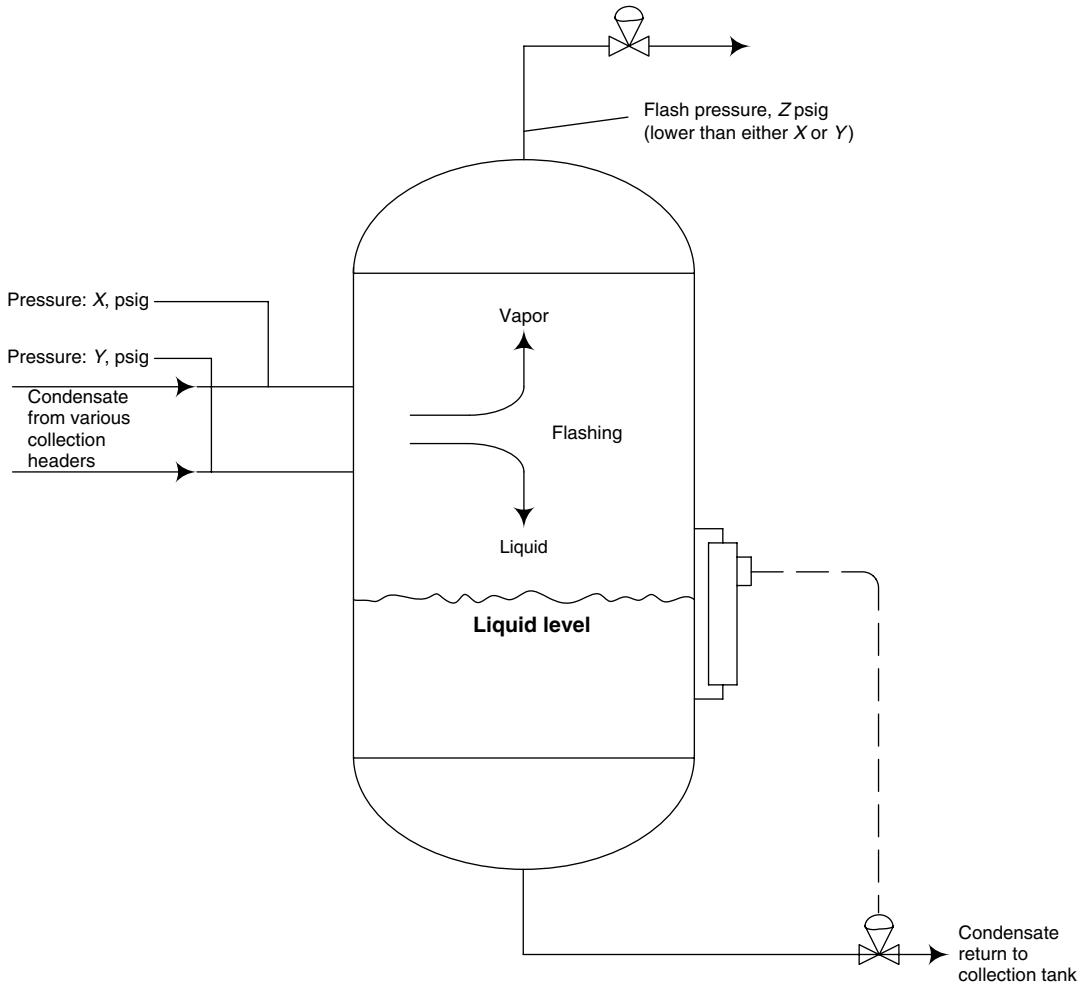
$$\Delta P_T = \frac{0.000336 f W^2}{d^5 \rho_M} = \frac{0.000336 (0.01628) (10,000)^2}{(4.026)^5 (0.2634)} = 1.964 \text{ psi/100 ft}$$

Velocity of the flashed condensate mixture is

$$\begin{aligned} v &= \frac{3.054}{d^2} \left[ \frac{W_G}{\rho_G} + \frac{W_L}{\rho_L} \right] \\ &= \frac{3.054}{(4.026)^2} \left[ \frac{1365}{0.0361} + \frac{8635}{59.78} \right] \\ &= 7154 \text{ ft/min} \end{aligned} \quad (4-340)$$

Since the velocity  $v \geq 5000$  ft/min, the condensate may cause deterioration of the 4-in. line.

The Excel spreadsheet (Example 4-27.xls) calculates the parameters for 6- and 8-in. Sch. 40 pipe sizes. Table 4-44 compares the results of 4-, 6-, and 8-in. pipe sizes.



**Figure 4-66** Typical steam condensate flashing operation.

surface of the granular solids, and void fraction. The original Ergun equation is

$$\frac{\Delta P}{L} g_c = \frac{150 (1 - \varepsilon)^2}{\varepsilon^3} \frac{\mu v}{D_p^2} + \frac{1.75 (1 - \varepsilon)}{\varepsilon^3} \frac{Gv}{D_p} \quad (4-341)$$

where

$\Delta P$  = pressure drop, psi ( $N/m^2$ )

$L$  = unit depth of packed bed, ft (m)

$g_c$  = dimensional constant,  $32.174 \text{ lb}_m \text{ft/lb}_f \text{s}^2$  ( $1 \text{ kg m/N s}^2$ )

$\mu$  = viscosity of fluid,  $\text{lb}/\text{ft h}$  ( $\text{kg}/\text{m s}$ )

$v$  = superficial fluid velocity,  $\text{ft}/\text{s}$  ( $\text{m}/\text{s}$ )

$D_p$  = effective particle diameter, ft (m)

$\varepsilon$  = porosity or void fraction of bed

$\rho$  = fluid density,  $\text{lb}/\text{ft}^3$  ( $\text{kg}/\text{m}^3$ )

$G$  = superficial mass velocity,  $\text{lb}/\text{h ft}^2$ .

Equation (4-341) gives the total energy loss in fixed beds as the sum of viscous energy loss (the first term on the right side of the equation) and the kinetic or turbulent energy loss (the second term on the right side of the equation). For gas systems, approximately 80% of the energy loss is dependent on turbulence and can be represented by the second term of Eq. (4-341). In liquid systems, the viscous term is the major factor.

For packed beds in either laminar or turbulent flow, the Ergun equation is

$$\frac{dP}{dl} = -\frac{\rho v_s^2}{D_p} \frac{(1 - \varepsilon)}{\varepsilon^3} \left\{ \frac{150 (1 - \varepsilon) \mu}{D_p \rho v_s} + 1.75 \right\} \quad (4-342)$$

where

$D_p$  = diameter of the packing. For non-spherical packing,  $D_p = 6(1 - \varepsilon)/\phi_s S$ , where  $S = S_o (1 - \varepsilon)$  and

$S_o = 6/D_p \phi_s$  where  $S_o$  is the surface area per unit volume of solid material in the bed and  $\phi_s = 1$  for a spherical particle. Use six times the ratio of volume to surface area of the packing as an effective  $D_p$

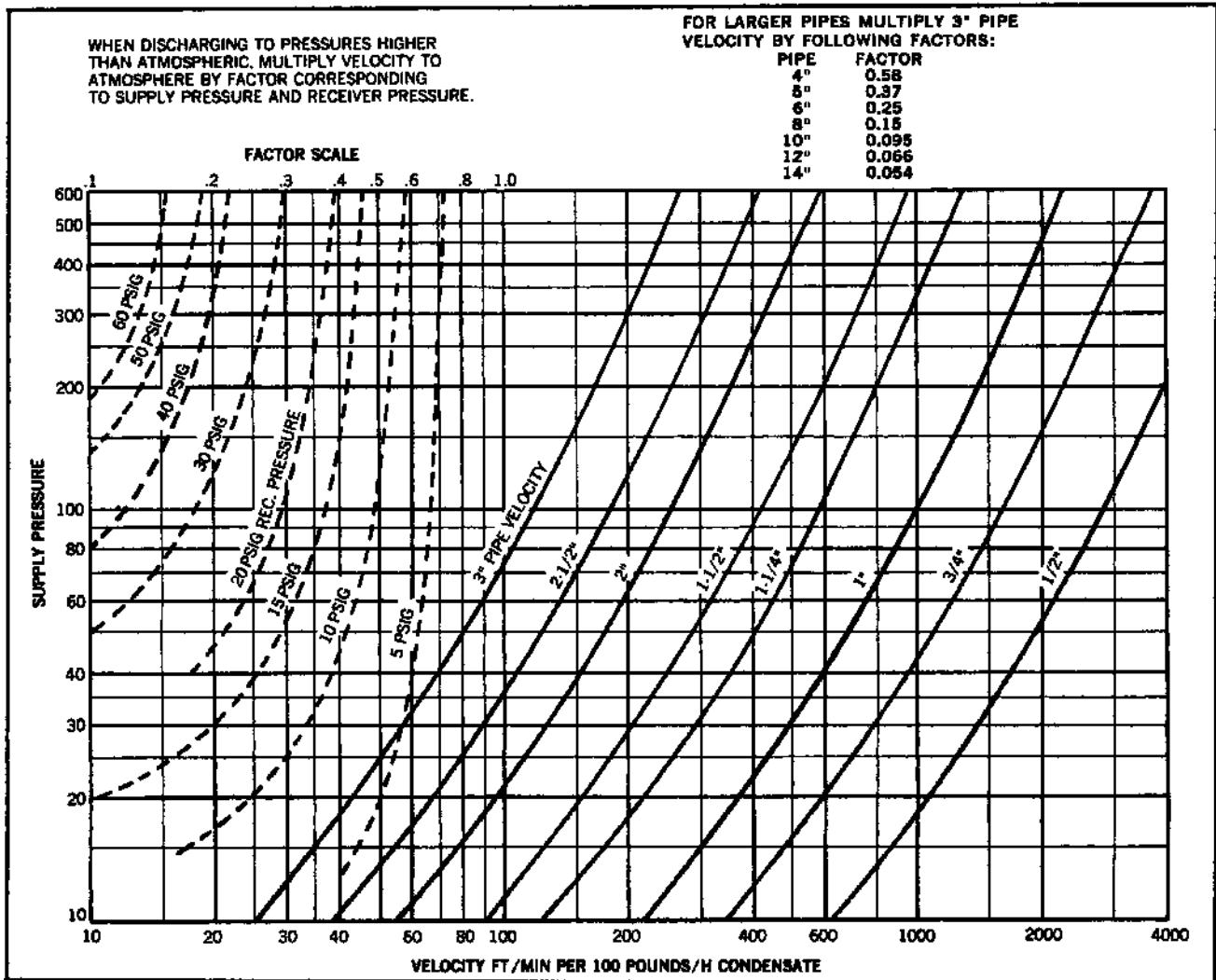
$v_s$  = superficial velocity when the tube is empty

$\rho$  = fluid density

$\phi_s$  = shape factor of the solid, defined as the quotient of the area of a sphere equivalent to the volume of the particle divided by the actual surface of the particle.

Equation (4-342) can be expressed in the form

$$-\frac{dP}{dl} = f \frac{\rho v_s^2}{g_c D_p} \quad (4-343)$$

**VELOCITY AT PIPE EXIT WHEN DISCHARGING CONDENSATE AT SATURATION TEMPERATURES FROM VARIOUS PRESSURES TO ATMOSPHERE AT A RATE OF 100 POUNDS/HR.**


**Figure 4-67** Sarco flashing steam condensate line sizing flow chart. (By permission from Spirax-Sarco, Inc., Allentown, PA [75].)

where the friction factor is

$$f = \frac{1 - \varepsilon}{\varepsilon^3} \left[ a + \frac{b(1 - \varepsilon)}{Re} \right] \quad (4-344)$$

with  $a = 1.75$ ,  $b = 150$ , and  $Re = \rho v_s D_p / \mu$

Handley and Hegg [79] derived a value with  $a = 1.24$  and  $b = 368$ . McDonald et al. [80] proposed that  $a = 1.8$  for smooth particles and 4.0 for rough particles and  $b = 180$ . Hicks [81] reviewed various  $\Delta P$  equations and inferred that the Ergun equation is limited at  $Re/(1 - \varepsilon) < 500$  and Handley and Hegg's equation to  $1000 < Re/(1 - \varepsilon) < 5000$ . He developed an equation for the friction factor for spheres, which is of the form

$$f = \frac{6.8(1 - \varepsilon)^{1.2}}{\varepsilon^3} Re^{-0.2} \quad (4-345)$$

which fits Ergun's and Handley and Hegg's data together with the results of Wentz and Thodos [82] at high Reynolds number. This shows that  $a$  and  $b$  are not true constants as stated by Tallmadge [83], who suggested that  $a = 1.75$  and  $b = 4.2 Re^{5/6}$ .

Gamane and Lannoy [84] developed the pressure drop through a fixed bed with randomly arranged spherical particles in terms of meters of fluid column ( $\Delta P/\delta$ ) as

$$\frac{\Delta P}{\delta} = \left[ 360 \frac{(1 - \varepsilon)}{Re} + 3.6 \right] \left( \frac{1 - \varepsilon}{\varepsilon^3} \right) \left( \frac{L}{D_p} \right) \left( \frac{v_s^2}{2g} \right) \quad (4-346)$$

and rewritten as

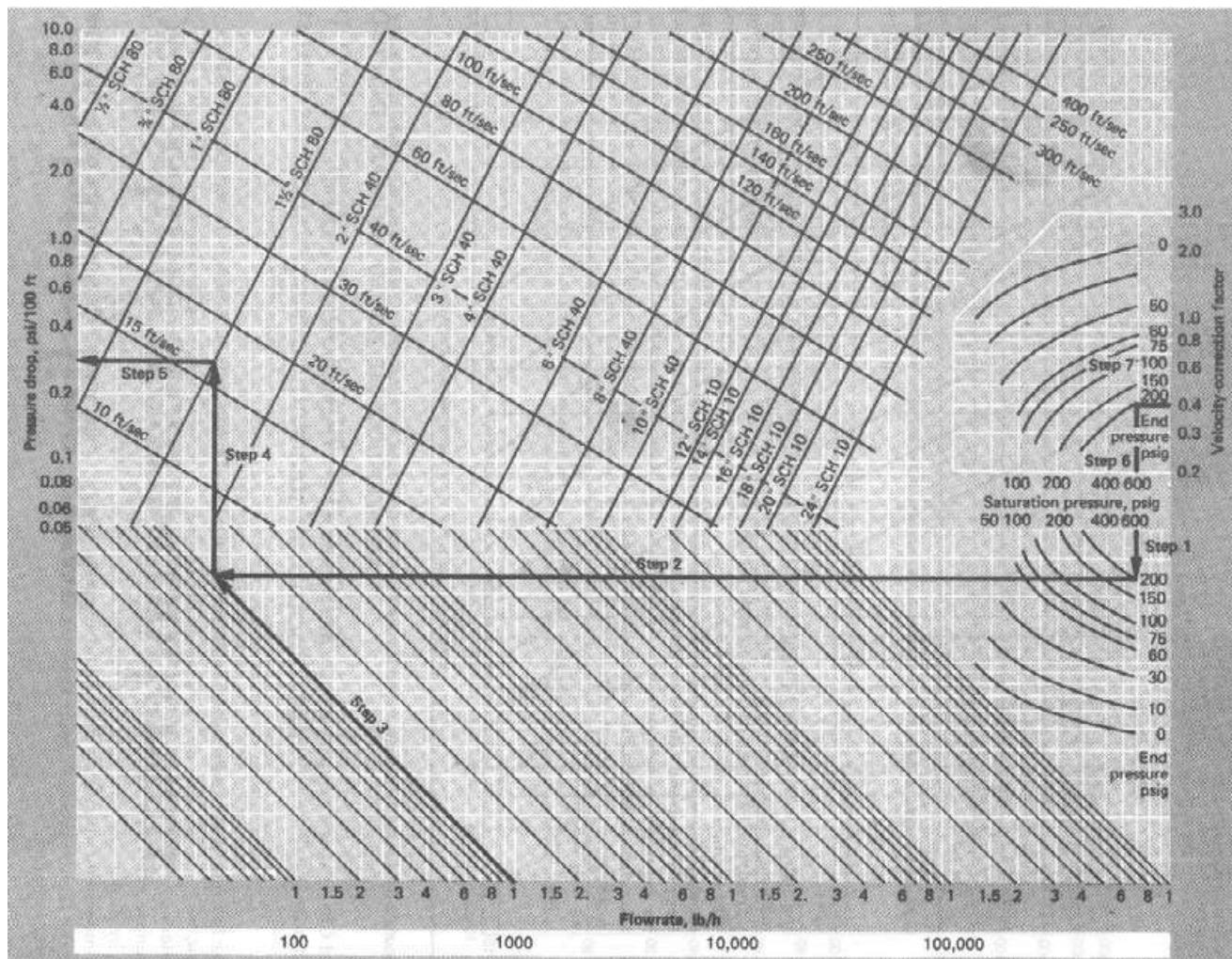
$$\frac{\Delta P}{\delta} = f \left( \frac{L}{D_p} \right) \frac{v_s^2}{2g}, \text{ m} \quad (4-347)$$

or

$$\Delta P = f \left( \frac{L}{D_p} \right) \frac{\rho v_s^2}{2}, \frac{\text{N}}{\text{m}^2} \quad (4-348)$$

$$\text{with } f = \left[ 360 \frac{(1 - \varepsilon)}{Re} + 3.6 \right] \left( \frac{1 - \varepsilon}{\varepsilon^3} \right) \quad (4-349)$$

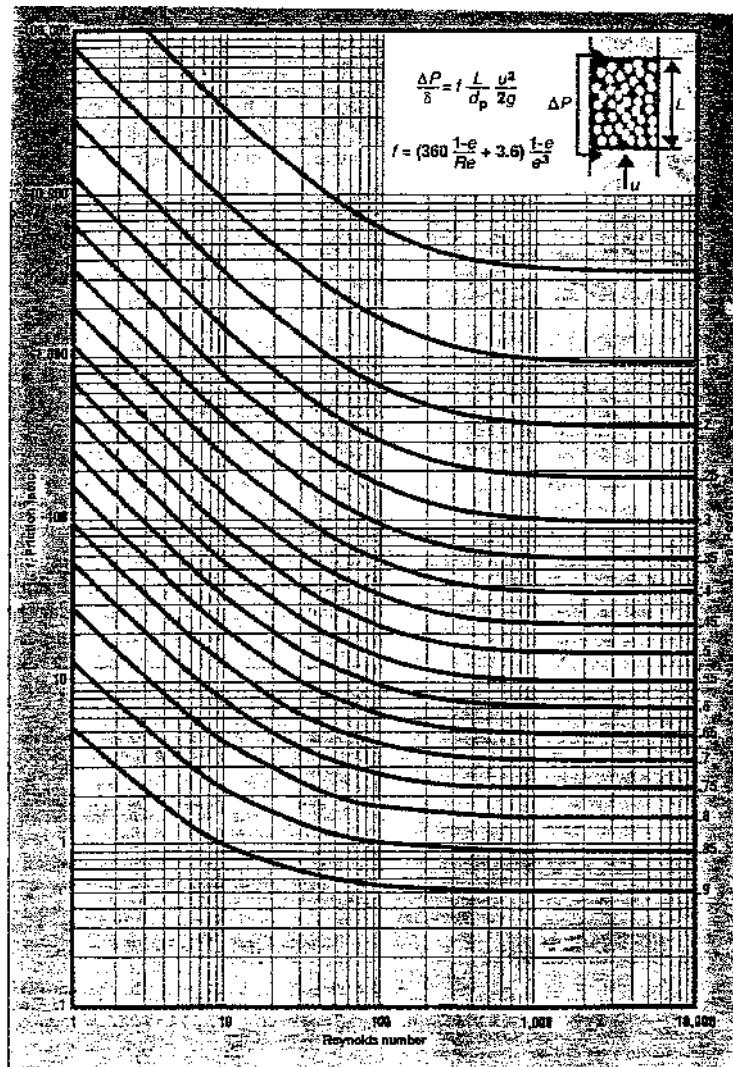
in the range  $1 < Re < 10,000$



**Figure 4-68** Flashing steam condensate line sizing chart. (By permission from Ruskin, R.P., "Calculating Line Sizes For Flashing Steam Condensate", *Chem. Eng.*, Aug 18, 1985, p. 101.)

**TABLE 4-44 Computer Results of Line Sizes for Flashing Steam Condensate of Example 4-27**

Pipe Internal Diameter (in.)	4.026	6.065	7.981
Total flow of mixture in condensate header, lb/h	10,000.0	10,000.0	10,000.0
Steam condensate pressure before flashing, psia	114.7	114.7	114.7
Flashed condensate header pressure, psia	14.7	14.7	14.7
Weight fraction of condensate flashed to vapor	0.135	0.135	0.135
Flashed steam flow rate, lb/h	1,346.0	1,346.0	1,346.0
Flashed condensate liquid flow rate, lb/h	8,654.0	8,654.0	8,654.0
Temperature of flashed condensate, °F	212.36	212.36	212.36
Flashed steam density, lb/ft <sup>3</sup>	0.036	0.036	0.036
Flashed condensate liquid density, lb/ft <sup>3</sup>	59.780	59.780	59.780
Density of mixture, lb/ft <sup>3</sup>	0.267	0.267	0.267
Friction factor	0.0163	0.0149	0.0141
Pressure drop of flashed condensate mixture, psi/100 ft	1.937	0.228	0.055
Velocity of flashed condensate mixture, ft/min	7,055.0	3,109.0	1,795.0
	Velocity is greater than 5,000 ft/min. Deterioration of the pipe line is possible.	Velocity is less than 5,000 ft/min. The Condensate header line will not deteriorate.	Velocity is less than 5000 ft/min. The Condensate header line will not deteriorate.



**Figure 4-69** To calculate friction factor, first locate the intersection of the Reynolds number and the porosity. (Source: Gammane N. and F. Lannoy, *Chem. Eng.*, Aug 1996, pp.123–124.)

where

$$Re = D_p v_s / \nu$$

$$\delta = \rho g = \text{specific weight } (\text{kg/m}^2 \text{s}^2)$$

$$D_p = \text{equivalent diameter } (\text{m})$$

$$\nu = \mu / \rho = \text{fluid kinematic viscosity } (\text{m}^2/\text{s})$$

$$f = \text{McDonald's friction factor.}$$

Figure 4-69 shows the plot of friction factor against Reynolds number and porosity. Gamane and Lannoy expressed that the diagram gives good results for randomly arranged spherical particles and for porosity between 0.2 and 0.47. However, for greater porosity, Figure 4-69 gives from 5 to 50% error according to the packing.

Equation (4-342) is also a good approximation for a fluidized bed reactor up to the minimum fluidizing condition. However, beyond this range, fluid dynamic factors are more complex than for the packed bed reactor. Among the parameters that influence the  $\Delta P$  in a fluidized bed reactor are the different types of two-phase flow, smooth fluidization, slugging or channeling, the particle size

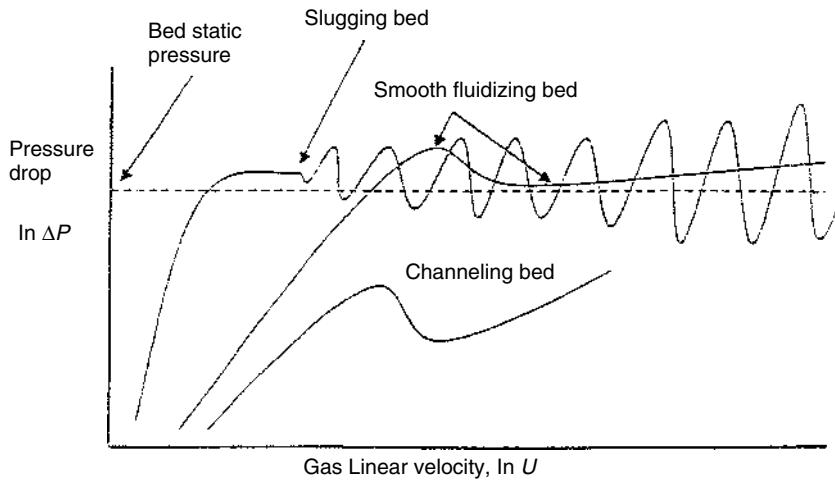
distribution, and the gas flow rate. After reaching a peak  $\Delta P$  at the point of minimum fluidization, the  $\Delta P$  of a smoothly fluidizing bed drops to a value that approximately corresponds to the static pressure of the bed and remains nearly constant with an increase in the gas flow rate until the entrainment point is reached. A slugging bed displays a wide fluctuation in the  $\Delta P$  beyond the point of minimum fluidization and a channeling bed exhibits a  $\Delta P$  far below the bed static pressure. Figure 4-70 shows the behavior of fluidized beds in these three operation modes.

$\Delta P$  for laminar flow in a circular tube is given by the Hagen–Poiseuille equation:

$$\frac{dP}{dl} = -8 \frac{\nu \mu}{R} \quad (4-350)$$

For turbulent flow,  $\Delta P$  is calculated from

$$\frac{dP}{dl} = -4 f_F \frac{1}{D} \frac{\rho v^2}{2} \quad (4-351)$$



**Figure 4-70** Relationship between  $\Delta P$  and gas linear velocity.

or

$$\frac{dP}{dl} = -f_F \left( \frac{\rho v^2}{R} \right) \quad (4-352)$$

or where the Darcy friction factor is four times the Fanning friction factor (e.g.,  $f_D = 4f_F$ )

$$\frac{dP}{dl} = -f_D \left( \frac{\rho v^2}{4R} \right) \quad (4-353)$$

The Fanning friction factor is given by  $f_F = 0.079 Re^{-0.25}$  and the Reynolds number is  $Re = \rho v D_p / 6\mu (1-\varepsilon)$  for a packed bed consisting of spherical particles and  $Re = \rho v D_p \phi_s / 6\mu (1-\varepsilon)$  for non-spherical particles. The values of  $\phi_s$  for other materials are provided by Perry et al. [1] and listed in the spreadsheet (PACKED.xls) package.

Industrial design problems often occur in tubular reactors that involved the simultaneous solution of  $\Delta P$ , energy and momentum balances.

### THE EQUATIONS

The original Ergun equation of the total energy loss can be rearranged as follows:

$$\Delta P = \frac{BL}{144} \frac{1-\varepsilon}{\varepsilon^3} \frac{G^2}{D_p g_c \rho} \left[ \frac{150 (2.419 \mu) (1-\varepsilon)}{D_p G} + 1.75 \right] \quad (4-354)$$

$\varepsilon$  = fraction void volume of bed,  $\text{ft}^3$  void/

$\text{ft}^3$  of packed tower volume.

$$\begin{aligned} &= \frac{\text{volume of void}}{\text{volume of bed}} \\ &= \frac{\left[ \left( \frac{\text{mass of bed}}{\rho_{\text{bed}}} \right) - \left( \frac{\text{mass of bed}}{\rho_{\text{catalyst}}} \right) \right]}{\text{mass of bed}} \end{aligned}$$

$$= \frac{\rho_c - \rho_b}{\rho_c} \quad (4-355)$$

where

$\rho_c$  = density of catalyst,  $\text{lb}/\text{ft}^3$

$\rho_b$  = density of packed bed,  $\text{lb}/\text{ft}^3$

$$\rho = \frac{M_w P}{10.73 Z T} \quad (4-356)$$

$$D_p = \frac{6 (1-\varepsilon)}{S} \quad (4-357)$$

$$S = \frac{(1-\varepsilon)}{V_p} A_p \quad (4-358)$$

For cylindrical particles

$$V_p = \frac{1}{1728} \left[ \frac{\pi}{4} (PD)^2 (PL) \right] \quad (4-359)$$

$$A_p = \frac{1}{144} \left[ \frac{\pi PD^2}{2} + \pi (PD) (PL) \right] \quad (4-360)$$

Reynolds number

$$Re = \frac{GD_p}{2.419 \mu (1-\varepsilon)} \quad (4-361)$$

Friction factor

For laminar flow with  $Re < 1$

$$f_p = \frac{150}{Re} \quad (4-362)$$

For intermediate Reynolds number

$$f_p = \frac{150}{Re} + 1.75 \quad (4-363)$$

For completely turbulent flow, it is assumed that  $f_p$  approaches a constant value for all packed beds with the same relative roughness. The constant is found by experiment to be 1.75 [85].

where

$A$  = cross-sectional area of bed,  $\text{ft}^2$

$A_p$  = area of particles,  $\text{ft}^2$

$BL$  = bed length, ft

$D_p$  = effective particle diameter, ft

$f_p$  = friction factor

$$g_c = \text{gravitational constant} = 4.17 \times 10^8 \left( \frac{\text{lb}_m}{\text{lb}_f} \cdot \frac{\text{ft}}{\text{h}^2} \right)$$

$$G = \text{superficial mass velocity} = \frac{W}{A} \left( \frac{\text{lb}}{\text{h ft}^2} \right)$$

MW = molecular weight of fluid

$N_{Re}$  = Reynolds number, dimensionless

$P$  = fluid pressure, psia

PD = particle diameter, in.

PL = particle length, in.

$S$  = packed bed surface area,  $\text{ft}^2/\text{ft}^3$  bed

$T$  = fluid temperature,  $^{\circ}\text{R}$

$V_p$  = volume of particles,  $\text{ft}^3$

$W$  = fluid flowrate,  $\text{lb/h}$

$Z$  = compressibility factor

$\Delta P$  = total pressure drop in packed bed,  $\text{lb/in.}^2$

$\varepsilon$  = fraction voids in packed bed

$\rho$  = density of fluid at flowing conditions,  $\text{lb}/\text{ft}^3$

$\mu$  = fluid viscosity, cP ( $1 \text{ cP} = 2.419 \text{ lb}/\text{ft h}$ ).

#### EXAMPLE 4-28

Calculate the pressure drop in a 60 ft length of 1½-in. (Sch. 40, ID = 1.610 in.) pipe packed with catalyst pellets ¼ in. in diameter when 104.4 lb/h of gas is passing through the bed. The temperature is constant along the length of pipe at 260°C. The void fraction is 45% and the properties of the gas are similar to those of air at this temperature. The entering pressure is 10 atm. The table below gives the required data.

Packed bed length, ft	60
Packed bed diameter, ft	0.134
Fluid flow rate, lb/h	104.4
Particle diameter, in.	0.25
Particle length, in.	0.25
Fraction voids in packed bed	0.45
Density of fluid at flowing conditions, $\text{lb}/\text{ft}^3$	0.413
Fluid viscosity, cP	0.0278
Gravitational constant, $(\text{lb}_m/\text{lb}_f)(\text{ft}/\text{h}^2)$	$4.17 \times 10^8$

Area of Particles is

$$\begin{aligned} A_p &= \frac{1}{144} \left[ \frac{\pi (\text{PD})^2}{2} + \pi (\text{PD})(\text{PL}) \right] \quad (4-360) \\ &= \frac{1}{144} \left[ \frac{\pi (0.25)^2}{2} + \pi (0.25)(0.25) \right] \\ &= 0.00204 \text{ ft}^2 \end{aligned}$$

Volume of cylindrical particles is

$$\begin{aligned} V_p &= \frac{1}{1728} \left[ \frac{\pi}{4} (\text{PD})^2 (\text{PL}) \right] \quad (4-359) \\ &= \frac{1}{1728} \left[ \frac{\pi}{4} (0.25)^2 (0.25) \right] \\ &= 7.1017 \times 10^{-6} \text{ ft}^3 \end{aligned}$$

Packed bed surface area is

$$\begin{aligned} S &= \frac{(1 - \varepsilon)}{V_p} A_p \quad (4-358) \\ &= \frac{(1 - 0.45)}{7.1017 \times 10^{-6}} (0.00204) \\ &= 158.4 \text{ ft}^2/\text{ft}^3 \text{ of bed.} \end{aligned}$$

Effective diameter is

$$\begin{aligned} D_p &= \frac{6 (1 - \varepsilon)}{S} \quad (4-257) \\ &= \frac{6 (1 - 0.45)}{158.4} \\ &= 0.0208 \text{ ft} \end{aligned}$$

*Solution*

The cross-sectional area of the bed

$$\begin{aligned} A &= \frac{\pi (\text{BD})^2}{4} = \frac{\pi (0.134)^2}{4} \\ &= 0.0141 \text{ ft}^2 \end{aligned}$$

Superficial mass velocity,  $W/A$

$$\begin{aligned} G &= W/A = 104.4/0.0141 \\ &= 7403 \text{ lb/h ft}^2 \end{aligned}$$

(continued)

**EXAMPLE 4-28—(continued)**

Fluid's Reynolds number

$$\begin{aligned} Re &= \frac{GD_p}{2.419\mu(1-\varepsilon)} \\ &= \frac{(7403)(0.0208)}{2.419(0.0278)(1-0.45)} \\ &= 4170 \end{aligned} \quad (4-361)$$

Friction factor

For laminar flow with  $Re < 1$ 

$$f_p = \frac{150}{Re} \quad (4-362)$$

For intermediate Reynolds number

$$\begin{aligned} f_p &= \frac{150}{Re} + 1.75 \\ &= \frac{150}{4170} + 1.75 \\ &= 1.7860 \end{aligned} \quad (4-363)$$

The developed spreadsheet (Example 4-28.xls) calculates the Reynolds number and pressure drop of varying particle lengths. The simulation exercise gives a pressure drop of 68.6 psi for a particle length of 0.25 in. Table 4-45 gives the Reynolds number, friction factor, and pressure drop of catalyst pellets of 0.25 in. and at different particle lengths. The spreadsheet results show that pressure drop in a packed bed depends on size and shape of the particles.

**TABLE 4-45 Pressure Drop in a 1½ in. Packed Bed**

Particle Length (in.)	Reynolds Number, $Re$	Friction Factor, $f_p$	Pressure Drop, $\Delta P$ (lb/in. <sup>2</sup> )
0.250	4170	1.7860	68.640
0.300	4415	1.7840	64.720
0.350	4609	1.7825	61.951
0.400	4766	1.7815	59.877
0.450	4895	1.7806	58.266

**EXAMPLE 4-29**

A bed with a height 2.5 m and an internal diameter of 0.035 m is packed with cylindrical particles of diameter  $D_p = 0.003$  m. The void fraction of the bed is  $\varepsilon = 0.33$ . A gas of density  $\rho = 0.9487$  kg/m<sup>3</sup> and dynamic viscosity  $\mu = 3.1 \times 10^{-5}$  Pa s flows through the bed with a superficial mass flow velocity of 1.4 kg/m<sup>2</sup>s. Determine the pressure drop in the bed.

*Solution*

From Ergun's Eq. (4-341)

$$-\frac{dP}{dl} = f \frac{\rho v_s^2}{g_c D_p} \quad (4-343)$$

where the friction factor  $f$  is

$$f = \left( \frac{1-\varepsilon}{\varepsilon^3} \right) \left[ a + \frac{b(1-\varepsilon)}{Re} \right] \quad (4-344)$$

$$a = 1.75 \text{ and } b = 150$$

$$\begin{aligned} v_s &= \frac{1.4}{0.9487} \left( \frac{\text{kg}}{\text{m}^2 \text{s}} \cdot \frac{\text{m}^3}{\text{kg}} \right) \\ &= 1.475 \text{ m/s} \end{aligned}$$

The Reynolds number,  $Re$ 

$$\begin{aligned} Re &= \frac{\rho v_s D_p}{\mu} \\ &= \left( \frac{0.9487 \times 1.475 \times 0.003}{3.1 \times 10^{-5}} \right) \left( \frac{\text{kg}}{\text{m}^3} \cdot \frac{\text{m}}{\text{s}} \cdot \frac{\text{m}}{\text{kg}} \right) \\ &= 135 \end{aligned}$$

The friction factor  $f$  from Eq. (4-344) is

$$\begin{aligned} f &= \left( \frac{1-0.33}{0.33^3} \right) \left[ 1.75 + \frac{150(1-0.33)}{135} \right] \\ &= 46.4 \end{aligned}$$

The pressure drop  $-\Delta P_t$  is

$$\begin{aligned} -\Delta P_t &= f \frac{\rho v_s^2 L}{g_c D_p}, \text{ N/m}^2 \\ -\Delta P_t &= 46.4 \frac{0.9487 \times 1.475^2 \times 2.5}{1 \times 0.003} = 79,809 \text{ N/m}^2 \\ &= 0.798 \text{ bar} \end{aligned}$$

McDonald et al. equation:  $a = 1.8$ ,  $b = 180$ ,  $f = 50.2$ , and  $-\Delta P_t = 86,368 \text{ N/m}^2 = 0.864 \text{ bar}$ Hicks equation:  $f = 43.9$  and  $-\Delta P_t = 75,509 \text{ N/m}^2 = 0.755 \text{ bar}$ Tallmadge equation:  $a = 1.75$ ,  $b = 4.2 Re^{5/6}$ ,  $f = 55.76$ , and  $-\Delta P_t = 95,908 \text{ N/m}^2 = 0.959 \text{ bar}$ Gamane and Lannoy equation:  $a = 3.6$ ,  $b = 360$ ,  $f = 100.43$ , and  $-\Delta P_t = 86,370 \text{ N/m}^2 = 0.864 \text{ bar}$ The Excel spreadsheet program (PACKED.xls) was developed to determine  $\Delta P_s$  of packed beds with known parameters.

**NOMENCLATURE**

$A$	Internal cross-section area for flow, $\text{ft}^2$ ; or area of orifice, nozzle, or pipe, $\text{ft}^2$	$(1/f)^{1/2}$	Gas transmission factor, or sometimes termed efficiency factor, see Table 4-30
$a$	Internal cross-section area for flow in pipe, $\text{in}^2$ .	$f_F$	Fanning friction factor
$a'$	Fractional opening of control valve, generally assumed at 60% = 0.60	$G$	Mass flow rate of gas phase, pounds per hour per square foot of total pipe cross-section area
$a_o$	Orifice area, $\text{in}^2$ .	$G'$	Mass rate, $\text{lbs/s}$ ( $\text{ft}^2$ cross section)
$a_w$	Velocity of propagation of elastic vibration in the discharge pipe $\text{ft/s} = 4660/(1 + K_{hs}B_r)^{1/2}$	$GPM$	Gallons per minute flow
$B$	Base pressure drop for control valve from manufacturer, $\text{psi}$	$g$	Acceleration of gravity, $32.2 \text{ ft/S}^2$
$B_r$	Ratio of pipe diameter (ID) to wall thickness	$H$	Total heat, $\text{Btu/lb}$
$C$	Condensate, $\text{lbs/h}$ (Eq. (4-326)); or for pipe, Williams and Hazen constant for pipe roughness (see Cameron Table 4-46 and Figure 4-28); or flow coefficient for sharp-edged orifices	$h$	Average height of all vertical rises (or hills) in two-phase pipe line, $\text{ft}$
$C'$	Flow coefficient for orifices and nozzles, which equals the discharge coefficient corrected for velocity of approach = $C_d/(1 - \beta_4)^{1/2}$	$h_{\text{or}, h}$	Static head loss, $\text{ft}$ of fluid flowing
$C'$	$C$ for Figures 4-19 and 4-20	$h_1$	Enthalpy of liquid at higher pressure, $\text{Btu/lb}$
$C' = c'$	Orifice flow coefficient	$h_2$	Enthalpy of liquid at lower or flash pressure, $\text{Btu/lb}$
$C_d$	Discharge coefficient for orifice and nozzles	$h_f = h_L$	Loss of static pressure head due to friction of fluid flow, $\text{ft}$ of liquid
$C_{D1}$	Diameter correction factor, vacuum flow, Figure 4-56	$h_p$	Enthalpy of liquid at supply steam pressure, $\text{Btu/lb}$
$C_{D2}$	Diameter correction factor, vacuum flow, Figure 4-56	$h_r$	Enthalpy of liquid at return line pressure, $\text{Btu/lb}$
$C_v$	Standard flow coefficient for valves; flow rate in $\text{gpm}$ for 60° F water with 1.0 psi pressure drop across the valve, = $Q[(p/62.4)(\Delta P)]^{1/2}$	$h_{L'}$	Head at orifice, $\text{ft}$ of liquid
$C'_v$	Valve coefficient of flow, full open, from manufacturer's tables	$h'_L$	Differential static head or pressure 10ss across flange taps when $C$ or $C'$ values come from Figure 4-17 or Figure 4-18, $\text{ft}$ of fluid
$C_{T1}$	Temperature correction factor, vacuum flow, Figure 4-56	$h_{wh}$	Maximum pressure developed by hydraulic shock, $\text{ft}$ of water (water hammer)
$C_{T2}$	Temperature correction factor, vacuum flow, Figure 4-56	$K$	Resistance coefficient, or velocity head loss in equation, $h_L = Kv^2/2g$
$C_1$	Discharge factor from chart in Figure 4-40a	$K_d$	Orifice or nozzle discharge coefficient
$C_2$	Size factor from Table 2-26, use with equation on Figure 4-40a	$K_{hs}$	Ratio of elastic modulus of water to that of the metal pipe material (water hammer)
$C_p/C_v$	Ratio of specific heat at constant pressure to that at constant volume = $k$	$k$	Ratio of specific heat, $c_p/c_v$
$D$	Inside diameter of pipe, $\text{ft}$	$L$	Pipe length, $\text{ft}$
$D_H$	Hydraulic diameter, $\text{ft}$	$L_e$	Equivalent length of line of one size referenced to another size, miles (or feet)
$d$	Inside diameter of pipe, $\text{in.} = d_i$	$L_{eq}$	Equivalent length of pipe plus equivalent length of fittings, valves, etc., $\text{ft}$
$d_e$	Equivalent or reference pipe diameter, $\text{in.}$	$L_m$	Length of pipe, miles
$d_H$	Hydraulic diameter, or equivalent diameter, $\text{in.}$	$L_v$	Latent heat of evaporation of steam at flash pressure, $\text{Btu/lb}$
$d_o$	Orifice diameter, or nozzle opening, $\text{in.}$	$I$	Horizontal distance from opening to point where flow stream has fallen one foot, $\text{in.}$
$d_\infty$	Diameter of a single line with the same delivery capacity as that of individual parallel lines $d_1$ and $d_2$ (lines of same length)	$M_w$	MW = molecular weight
$d_i$	Inside diameter of pipe, $\text{in.}$	$MR$	Universal gas constant
$E$	Gas transmission "efficiency" factor, varies with line size and surface internal condition of pipe	$n$	Number of vertical rises (or hills) in two-phase pipe line flow or
$F$	Factor in Babcock's steam flow equation	$n$	Polytropic exponent in polytropic gas P-V relationship
$F_D$	Friction pressure loss (total) at design basis, for a system, $\text{psi}$ , for process equipment and piping, but excluding the control valve	$P$	Pressure, $\text{psig}$ ; or, pressure drop, $P$ , pounds per square inch, Babcock Eq. (4-193)
$F_e$	Elevation factor for two-phase pipe line	$P_t$	Absolute pressure, torr
$F_M$	Friction pressure loss (total) at maximum flow basis, for a system, $\text{psi}$	$\Delta P_t$	Pressure drop, torr
$F_1$	Base friction factor, vacuum flow, Figure 4-56	$P'$	Pressure, psi absolute (psia)
$F_2$	Base friction factor, vacuum flow, Figure 4-56	$P_e$	Total pressure at lower end of system, $\text{psig}$
$f$	Friction factor, Moody or "regular" Fanning, see Note in Figure 4-5	$P_{br}$	Barometric pressure, $\text{psia}$
$f_T$	Turbulent friction factor, see Table 4-6	$P_S$	Total pressure upstream (higher) of system, $\text{psig}$
$f_g$	Moody or "regular" Fanning Friction for gas flow	$P_s$	Standard pressure for gas measurement, $\text{lbs/in.}^2$ absolute, $\text{psia}$
$f_{TP}$	Two-phase friction for wave flow	$p''$	Pressure, $\text{lbs}/\text{ft}^2$ absolute (in speed of sound equation, Eq. (4-122)), Note units.
		$p'$	Gauge pressure, $\text{psig}$
		$p_1$ or $P_1$	Initial pressure, $\text{in.}$ of mercury absolute, vacuum system

$\Delta P$	Pressure drop, $\text{lbs/in}^2$ . psi; or static loss for flowing fluid, psi	$v_m$	Mean velocity in pipe, at conditions of $V$ , ft/min
$\Delta P_c$	Pressure drop across a control valve, psi	$v_s$	Sonic (critical) velocity in compressible fluid, ft/s; or speed of sound, ft/s
$\Delta P_{vac}$	Pressure drop in vacuum system due to friction, in. water/100 ft pipe	$v_w$	Reduction in velocity, ft/s (actual flowing velocity, ft/s)
$\Delta P_{TPh}$	Total two-phase pressure drop for system involving horizontal and vertical pipe, psi per foot of length	$W$	Flow rate, lb/h
$\Delta P_{100}$	Pressure drop, psi/100 ft of pipe or equivalent	$W_m$	Mass flow rate of liquid phase, pounds per hour per square foot of total pipe cross-section area
$Q$	Flow rate, gallons per minute, gpm	$W_t$	Mass flow rate, lb/h/tube
$Q_b$	Flow rate, bbl/day	$w$	Flow rate, lb/min
$Q_D$	Design flow rate, gpm, or ACFM	$w_s$	Flow rate, lb/s; or sometimes, $W_s$
$Q_M$	Maximum flow rate, gpm, or ACFM	$x$	Fraction of initial line paralleled with new line
$q$	Flow rate at flowing conditions, $\text{ft}^3/\text{s}$	$Y$	Net expansion factor for compressible flow through orifices, nozzles, or pipe
$q_d$	Gas flow rate standard cubic feet per day, at 60° F and 14.7 psia (or 14.65 if indicated); or flow rate, $\text{ft}^3/\text{day}$ at base conditions of $T_s$ and $P_s$	$Z$	Compressibility factor for gases at average conditions, dimensionless. Omit for pressure under 100, psig
$q_{ds}$	Gas flow at designated standard conditions, $\text{ft}^3/\text{day}$ , cfD		
$q_h$	Gas flow rate, $\text{ft}^3/\text{h}$ , at 60° F and 14.4 psia		
$q'$	Gas flow, $\text{ft}^3/\text{sec}$ , at 14.7 psia and 60° F		
$q'_h$	Flow rate at standard conditions (14.7 psia and 60° F) $\text{ft}^3/\text{h}$ , SCFH		
$q_m$	Flow rate, $\text{ft}^3/\text{min}$		
$q'_m$	Free air, $\text{ft}^3/\text{min}$ at 60° F and 14.7 psia		
$R$	Individual gas constant = $MR/M = 1544/M$	$\Upsilon$	Ratio of internal diameter of smaller to large pipe sizes, or for orifices or nozzles, contractions or enlargements
$Re$	Reynolds number, see Figure 4-5	$\gamma$	Kinematic viscosity, $\text{ft}^2/\text{s}$
$R_H$	Hydraulic radius, ft	$\varepsilon$	Surface tension of liquid, dyn/cm
$R_c$	Ratio of compression at entrance of pipe, Figure 4-46	$\theta$	Roughness factor, effective height of pipe wall irregularities, ft
$r_c$	Critical pressure ratio = $P'_2/P'_1$	$\lambda$	Angles of divergence or convergence in enlargements or contractions in pipe systems, degrees
$S_g$	Specific gravity of gas relative to air (= ratio of molecular weight gas/29)	$\mu$	Homogeneous flow liquid ratio
$S^\circ$	Degrees of superheat in a steam condition, °F above saturated (not the actual temperature)	$\mu_e$	Absolute viscosity, cP
$s$	Steam quality as percent dryness, fractional molecular weight of gas	$\mu_g$	Absolute viscosity, lb (mass)/(ft s)
sp gr	molecular weight of air	$\mu_L$	Viscosity of gas or vapor phase, cP
sp gr	Specific gravity of fluid relative to water at same temperature	$\rho$	Viscosity of liquid phase, cP
$T$	Absolute Rankin temperature, $460 + t$ , °R	$\Sigma$	Density of fluid, $\text{lbs}/\text{ft}^3$ ; or $\text{lb/gal}$ ,
$T_s$	Standard temperature for gas measurement, °R = $460 + t$	$\Phi$	Summation of items
$T_1$	Average flowing temperature of gas, °R		Equations for $\Phi_{GTT}$ for two-phase pipe line flow
$t$	Temperature, °F		
$t_s$	Time interval required for the pressure wave to travel back and forth in a pipe, s		
$V$	Free air flow, $\text{ft}^3/\text{s}$ at 60° F and 14.7 psia		
$\bar{V}$	Specific volume of fluid, $\text{ft}^3/\text{lb}$		
$V$	Volume, $\text{ft}^3$		
$V_a$	Volume, $\text{ft}^3$		
$v$	Flow velocity (mean) or superficial velocity in pipe lines at flowing conditions for entire pipe cross section, $\text{ft/s}$ ; or reduction in velocity, $\text{ft/s}$ (water hammer)		

## GREEK SYMBOLS

$\beta$	Ratio of internal diameter of smaller to large pipe sizes, or for orifices or nozzles, contractions or enlargements
$\Upsilon$	Kinematic viscosity, $\text{ft}^2/\text{s}$
$\gamma$	Surface tension of liquid, dyn/cm
$\varepsilon$	Roughness factor, effective height of pipe wall irregularities, ft
$\theta$	Angles of divergence or convergence in enlargements or contractions in pipe systems, degrees
$\lambda$	Homogeneous flow liquid ratio
$\mu$	Absolute viscosity, cP
$\mu_e$	Absolute viscosity, lb (mass)/(ft s)
$\mu_g$	Viscosity of gas or vapor phase, cP
$\mu_L$	Viscosity of liquid phase, cP
$\rho$	Density of fluid, $\text{lbs}/\text{ft}^3$ ; or $\text{lb/gal}$ ,
$\Sigma$	Summation of items
$\Phi$	Equations for $\Phi_{GTT}$ for two-phase pipe line flow

## SUBSCRIPTS

0	Base condition for gas measurement
1	Initial or upstream or inlet condition, or $i_i$
2	Second or downstream or outlet condition
a	Initial capacity or first condition
b	New capacity or second condition
g	Gas
L	Liquid
vc	Gradual contraction
VE	Gradual enlargement

TABLE 4-46 Cameron Hydraulic Data

## Friction losses in pipes carrying water

Among the many empirical formulae for friction losses that have been proposed that of Williams and Hazen has been most widely used. In a convenient form it reads:

$$f = .2083 \left( \frac{100}{C} \right)^{1.85} q^{1.85}$$

in which

$f$  = friction head in ft of liquid per 100 ft of pipe (if desired in lb per sq in. multiply  $f \times .433 \times \text{sp gr}$ )

$d$  = inside dia of pipe in inches

$q$  = flow in gal per min

$C$  = constant accounting for surface roughness

This formula gives accurate values only when the kinematic viscosity of the liquid is about 1.1 centistokes or 31.5 SSU, which is the case with water at about 60°F. But the viscosity of water varies with the temperature from 1.8 at 32°F to .29 centistokes at 212°F. The tables are therefore subject to this error which may increase the friction loss as much as 20% at 32°F and decrease it as much as 20% at 212°F. Note that the tables may be used for any liquid having a viscosity of the same order as indicated above.

Values of  $C$  for various types of pipe are given below together with the corresponding multiplier which should apply to the tabulated values of the head loss,  $f$ , as given on pages 29 to 48.

TYPE OF PIPE	VALUES OF C		
	Range	Average value for good, clean, new pipe	Commonly used value for design purposes
High-best, smooth, well laid			
—	160-140	150	140
—	160-180	150	140
Low-poor or corroded	160-180	148	140
—	160-180	150	140
Cement—Asbestos.....	150	140	
Fibre.....	150	140	
Bituminous-enamel-lined iron or steel centrifugally applied.....	148	140	
Cement-lined iron or steel centrifugally applied.....	150	140	
Copper, brass, lead, tin or glass pipe and tubing.....	150-120	140	130
Wood-stave.....	145-110	120	110
Welded and seamless steel.....	150-80	140	100
Continuous-interior riveted steel (no projecting rivets or joints).....	—	139	100
Wrought-iron.....	150-80	130	100
Cast-iron.....	150-80	130	100
Tar-coated cast-iron.....	145-80	130	100
Girth-riveted steel (projecting rivets in girth seams only).....	—	130	100
Concrete.....	150-85	120	100
Full-riveted steel (projecting rivets in girth and horizontal seams).....	—	115	100
Vitrified.....	—	110	100
Spiral-riveted steel (flow with lap).....	—	110	100
Spiral-riveted steel (flow against lap).....	—	100	90
Corrugated steel.....	—	60	50
Value of C.....	180	140	180
Multiplier to correct tables.....	.47	.54	.63
	.71	.84	1.0
	1.22	1.59	1.93
	2.57		

Friction Losses In Pipe; C = 100  
5/8 Inch

FLOW U.S. gal per min	STANDARD WT STEEL			EXTRA STRONG STEEL		
	.369" inside dia			.315" inside dia		
	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
0.1	.565	.00	1.75	.884	.01	5.21
0.2	1.12	.02	6.51	1.77	.05	18.9
0.3	1.69	.04	15.4	2.65	.11	59.8
0.4	2.26	.06	22.8	3.54	.19	87.7
0.5	2.83	.12	34.4	4.42	.30	102
0.6	3.39	.18	48.2	5.33	.44	147
0.7	3.95	.24	62.1	6.20	.61	171
0.8	4.52	.32	72.4	7.08	.78	194
0.9	5.08	.40	82.1	7.96	.96	216
1.0	5.65	.50	92	8.84	1.21	239

## 5/8 Inch

FLOW U.S. gal per min	STANDARD WT STEEL			EXTRA STRONG STEEL		
	.364" inside dia			.302" inside dia		
	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
0.4	1.23	.02	5.22	1.79	.06	13.8
0.6	1.86	.05	11.1	2.69	.11	27.4
0.8	2.47	.09	18.8	3.69	.20	46.7
1.0	3.08	.15	26.5	4.48	.31	70.4
1.2	3.71	.21	33.9	5.38	.45	93.9
1.4	4.33	.29	42.8	6.27	.61	132
1.6	4.94	.38	67.4	7.17	.80	165
1.8	5.55	.48	84.4	8.07	1.01	200
2.0	6.17	.59	103	8.96	1.26	234
2.5	7.71	.92	155	11.2	1.95	336

## 3/8 Inch

FLOW U.S. gal per min	STANDARD WT STEEL			EXTRA STRONG STEEL		
	.498" inside dia			.423" inside dia		
	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
0.5	1.35	.03	4.30	1.88	.05	9.07
1.0	1.68	.04	8.60	2.28	.08	13.7
1.5	2.53	.10	12.8	3.48	.18	24.8
2.0	3.36	.15	18.4	4.57	.33	39.2
2.5	4.21	.28	25.4	5.71	.51	74.6
3.0	5.06	.40	39.5	6.85	.78	105
3.5	5.89	.54	56.0	8.00	.99	139
4.0	6.72	.70	74.8	9.14	1.30	178
4.5	8.41	1.10	104	11.4	2.0	225
5.0	10.1	1.65	134	13.7	2.9	377

(Source: By permission from G.V. Shaw and A.W. Loomis *Cameron Hydraulic Data*, 11th ed. Ingersoll-Rand Co., 1942 [7].)

TABLE 4-46—(continued)

Friction Losses In Pipe; C = 100 $\frac{1}{2}$ Inch										Friction Losses In Pipe; C = 100 1 Inch									
FLOW U S gal per min	Standard Wt Steel .622" inside dia			Extra Strong Steel .567" inside dia			Double Extra Strong Steel .253" inside dia			FLOW U S gal per min	Standard Wt Steel 1.049" inside dia			Extra Strong Steel .567" inside dia			Double Extra Strong Steel .399" inside dia		
	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft		Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
	0.5	.528	.00	.582	.686	.01	1.18	8.22	.16	47.2	1.06	1.11	.02	.595	.01	.490	2.28	.08	11.7
1.0	1.06	.08	2.10	1.37	.08	1.92	6.44	.64	170	1.58	1.44	.04	2.06	.03	1.74	3.42	.16	24.8	
2.0	2.11	.07	4.44	2.06	.07	3.28	9.56	1.45	361	2.57	2.74	.12	14.3	.08	2.24	4.66	.22	42.2	
4.0	4.22	.11	8.44	3.43	.13	21.6	16.1	4.03	928	5.64	5.43	.18	21.6			5.69	.50	65.8	
8.0	8.17	.16	16.8	4.11	.26	30.2				8.70	8.21	.21	11.7			9.11	.129	162	
16.0	16.28	.21	31.3	4.80	.36	40.2				16.43	16.4	.21	12.1			11.4	.20	232	
32.0	32.57	.28	57.8	5.48	.47	51.4				32.75	32.7	.24	25.4			13.7	.29	422	
64.0	64.75	.35	10.8	6.17	.59	52.0				65.28	65.2	.22	24.8			15.9	.39	849	
128.0	128.43	.43	21.4	6.86	.73	77.7				128.75	128.7	.14	53.0			18.3	.51	549	
256.0	256.81	.53	42.2	7.54	.98	92.7				257.0	257.0	.16	54.2			20.5	.65	592	
512.0	512.84	.63	67.8	8.23	1.05	109				512.75	512.7	.14	55.6			22.7	.80	532	
1024.0	1024.87	.73	67.8	8.91	1.23	126				1024.75	1024.7	.10	59.4			25.1	.98	532	
2048.0	2048.89	.86	76.8	9.60	1.43	145				2048.75	2048.7	.16	61.6			18.6			
4096.0	4096.92	.97	87.3	10.3	1.6	165				4096.75	4096.7	.16	67						
8192.0	8192.95	1.11	98.3	11.0	1.9	185				8192.75	8192.7	.14	78.5			2.4			
16384.0	16384.98	1.25	119	11.6	2.1	207				16384.75	16384.7	.14	83.4			23			
32768.0	32768.91	1.4	125	12.3	2.4	231				32768.75	32768.7	.16	85.6			28.8			
65536.0	65536.95	1.6	135	13.0	2.6	256				65536.75	65536.7	.16	92			28.7			
131072.0	131072.98	1.7	149	13.7	2.9	280				131072.75	131072.7	.16	100			20.1	6.2	295	
$\frac{3}{4}$ Inch										$\frac{1}{4}$ Inch									
FLOW U S gal per min	Standard Wt Steel .824" inside dia			Extra Strong Steel .742" inside dia			Double Extra Strong Steel .484" inside dia			FLOW U S gal per min	Standard Wt Steel 1.380" inside dia			Extra Strong Steel 1.378" inside dia			Double Extra Strong Steel .895" inside dia		
	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft		Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
	1.5	.903	.01	1.13	1.11	.02	1.98	8.25	.16	26.6	1.20	.02	.95	.01	.564	1.00	.02	.501	.06
2.0	1.51	.04	2.91	1.48	.03	2.21	4.34	.29	43.5	2.11	.06	2.85	.03	1.26	2.53	.10	6.98		
4.0	4.21	.07	6.42	2.68	.08	6.79	6.81	.66	92.3	4.22	.04	5.43	.04	1.51	1.74	.08	1.78		
8.0	8.41	.11	11.8	3.34	.17	14.4	9.77	1.48	195	8.71	.14	10.9	.18	2.41	3.88	.20	13.8		
16.0	16.21	.14	16.9	3.71	.21	17.4	11.4	1.48	238	16.41	.16	12.0	.26	2.41	3.88	.26	16.7		
32.0	32.41	.20	17.6	4.45	.31	24.5	13.0	2.6	333	32.71	.23	16.2	.36	3.43	5.14	.33	22.9		
64.0	64.41	.28	19.6	5.30	.43	32.6	16.2	3.6	443	64.71	.28	21.7	.43	4.00	6.16	.43	26.2		
128.0	128.43	.36	26.0	6.94	.68	41.7	17.4	4.7	567	128.75	.36	30.8	.50	10.0	14.7	.40	50.7		
256.0	256.45	.46	31.1	6.68	.69	51.3	19.5	5.9	704	256.75	.45	34.0	.50	13.1	17.2	.58	55.5		
512.0	512.56	.56	34.8	7.42	.86	52.0	21.7	7.3	836	512.75	.56	38.5	.59	15.7	19.7	.79	64.9		
1024.0	1024.58	.58	36.1	8.17	1.04	56.7	26.0	8.6	887	1024.75	.58	40.2	.67	18.7	22.4	1.03	72.0		
2048.0	2048.61	.81	53.0	9.51	1.28	58.3				2048.75	2048.7	.14	40.0			15.3	1.30	74.7	
4096.0	4096.72	7.82	95	61.5	9.63	1.44	102			4096.75	4096.7	.16	11.1			16.1	10.2	90.7	
8192.0	8192.84	8.48	110	70.9	11.7	1.7	117			8192.75	8192.7	.16	12.5			12.7	9.5	97	
16384.0	16384.93	9.63	114	70.9	11.9	2.2	150			16384.75	16384.7	.16	17.5			17.8	12.6	105	
32768.0	32768.98	10.8	118	71.2	12.4	2.2	157			32768.75	32768.7	.16	20.0			20.4	6.5	92	
65536.0	65536.98	12.0	126	72.6	14.8	3.4	227			65536.75	65536.7	.16	42.4			42.4			

(continued)

TABLE 4-46—(continued)

Friction Losses In Pipe; C = 100 1½ Inch										Friction Losses In Pipe; C = 100 2 Inch									
FLOW U S gal per min	Standard Wt Steel			Extra Strong Steel			Double Extra Strong Steel			FLOW U S gal per min	Standard Wt Steel			Extra Strong Steel			Double Extra Strong Steel		
	1.610" inside dia		Head loss ft per 100 ft	1.500" inside dia		Head loss ft per 100 ft	1.100" inside dia		Head loss ft per 100 ft		2.067" inside dia		Head loss ft per 100 ft	1.939" inside dia		Head loss ft per 100 ft	1.503" inside dia		Head loss ft per 100 ft
	Velocity ft per sec	Velocity head ft	Velocity ft per sec	Velocity head ft	Velocity ft per sec	Velocity head ft	Velocity head ft	Velocity head ft	Velocity head ft		Velocity ft per sec	Velocity head ft	Velocity ft per sec	Velocity head ft	Velocity head ft	Velocity ft per sec	Velocity head ft	Velocity ft per sec	
4	.63	.01	.267	.78	.01	.376	1.35	.03	1.70	6	.48	.00	.120	.84	.00	.163	.90	.01	.563
5	.79	.01	.403	.91	.01	.609	1.69	.04	2.07	7	.57	.01	.167	.66	.01	.229	1.09	.02	.769
6	.95	.01	.565	1.09	.02	.797	2.03	.06	2.49	8	.67	.01	.223	.76	.01	.304	1.27	.03	1.06
7	1.10	.03	.761	1.27	.03	1.05	2.36	.09	2.73	9	.77	.01	.286	.87	.01	.359	1.45	.03	1.34
8	1.26	.02	.952	1.45	.06	1.36	2.70	.11	3.14										1.67
9	1.42	.03	1.20	1.63	.04	1.69	3.04	.14	3.63										
10	1.58	.04	1.45	1.83	.05	2.06	3.38	.18	4.27										
12	1.89	.06	2.04	2.18	.07	2.87	4.05	.25	5.8										
14	2.21	.08	2.71	2.64	.10	3.32	4.73	.35	7.3										
16	2.58	.10	3.47	2.90	.13	4.89	5.40	.45	9.1										
18	2.84	.13	4.31	3.27	.17	6.08	6.08	.57	27.6										
20	3.15	.15	5.24	3.63	.20	7.39	6.76	.71	35.4										
22	3.47	.19	6.26	3.99	.25	8.82	7.42	.86	39.9										
24	3.78	.22	7.24	4.36	.30	10.4	9.10	1.02	46.8										
26	4.10	.26	8.51	4.72	.35	12.6	8.78	1.20	54.3										
28	4.41	.30	9.76	5.08	.40	13.5	9.45	1.39	62.3										
30	4.73	.35	11.1	5.45	.46	15.2	10.1	1.6	70.6										
32	5.04	.39	12.5	5.81	.52	17.6	10.8	1.8	79.8										
34	5.36	.45	14.9	6.17	.59	19.7	11.3	2.1	89.2										
36	5.67	.50	16.5	6.54	.66	21.9	12.8	2.3	99.2										
38	5.99	.56	17.2	6.90	.74	24.2	12.8	2.5	119										
40	6.30	.62	18.9	7.26	.82	25.7	13.5	2.8	121										
42	6.62	.68	20.7	7.63	.90	28.2	14.2	3.1	132										
44	6.93	.75	22.5	7.99	.99	30.8	14.9	3.5	144										
46	7.25	.82	24.5	8.35	1.08	34.5	15.6	3.8	156										
48	7.57	.89	27.1	8.72	1.18	37.5	16.3	4.1	169										
50	7.88	.97	28.6	9.08	1.28	40.3	16.9	4.4	182										
52	8.67	1.17	34.0	9.99	1.55	49.4	18.6	5.4	217										
54	9.46	1.39	40.0	10.9	1.8	56.4	20.3	6.4	236										
56	10.2	1.6	46.4	11.8	2.2	65.4	21.9	7.6	256										
58	11.0	1.9	53.2	12.7	2.5	76.0	23.6	8.7	329										
60	11.8	2.2	60.4	13.6	2.9	84.3	25.3	9.9	384										
62	12.6	2.5	68.1	14.5	3.3	94.1	27.0	11.3	456										
64	13.4	2.8	76.2	15.4	3.7	107	28.7	12.8	526										
66	14.2	3.1	84.7	16.8	4.1	119	30.4	14.4	549										
68	15.0	3.6	93.6	17.8	4.6	132	32.1	16.0	597										
70	15.8	3.9	103	18.2	5.1	145	33.8	17.8	657										
72	17.3	4.7	123	20.0	6.2	173													
74	18.9	5.6	144	21.8	7.4	203													
76	20.5	6.5	167	22.6	8.7	236													
78	22.1	7.6	190	25.4	10.0	271													
80	22.6	8.7	214	27.3	11.5	304													
82	23.2	9.9	240	29.0	12.1	346													
84	23.8	11.2	270	30.9	14.8	387													
86	24.4	12.5	306	32.7	16.6	431													

(continued)

TABLE 4-46—(continued)

Friction Losses In Pipe; C = 100 2½ Inch									Friction Losses in Pipe; C = 100 3 Inch										
FLOW U S gal per min	Standard Wt Steel			Extra Strong Steel			Double Extra Strong Steel			FLOW U S gal per min	Cast Iron			Std Wt Steel			Extra Strong Steel		
	2.469" inside dia		Head loss ft per 100 ft	2.823" inside dia		Head loss ft per 100 ft	1.771" inside dia		Head loss ft per 100 ft		3.0 inside dia		Head loss ft per 100 ft	3.068" inside dia		Head loss ft per 100 ft	3.900" inside dia		Head loss ft per 100 ft
	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft		Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
8	.54	.00	.126	.61	.01	.163	1.04	.03	.466	10	.48	.00	.053	.52	.00	.083	.77	.01	.256
10	.67	.01	.182	.76	.01	.244	1.30	.06	.714	18	.68	.01	.124	.78	.01	.16	1.16	.02	.533
12	.80	.01	.254	.91	.01	.342	1.56	.04	.928	20	.91	.01	.254	.87	.01	.227	1.04	.03	.924
14	.94	.01	.355	1.06	.03	.456	1.82	.05	1.70	25	1.13	.03	.355	1.09	.03	.344	1.30	.03	1.48
16	1.07	.03	.453	1.21	.03	.653	2.08	.07	2.18	30	1.36	.03	.553	1.30	.03	.481	1.56	.04	1.96
18	1.21	.03	.529	1.36	.08	.724	2.34	.09	2.71	40	1.59	.04	.714	1.82	.08	.842	2.70	.11	2.98
20	1.34	.03	.584	1.51	.04	.860	2.73	.12	3.50	45	1.82	.05	.914	2.08	.07	1.08	3.09	.15	3.33
22	1.47	.03	.626	1.67	.04	1.050	2.87	.18	3.93	50	2.04	.06	1.14	2.31	.09	1.34	3.47	.19	4.14
24	1.61	.04	.677	1.83	.06	1.259	3.13	.15	4.56	55	2.27	.08	1.35	2.50	.11	1.63	3.86	.23	5.05
26	1.74	.05	1.06	1.97	.06	1.433	3.29	.18	4.98	60	2.50	.10	1.54	2.86	.13	1.94	4.25	.28	5.97
28	1.88	.05	1.22	2.13	.07	1.64	3.45	.21	6.14	49	1.72	.12	1.94	2.60	.11	1.74	3.12	.15	7.05
30	2.01	.06	1.35	2.27	.08	1.86	4.00	.25	7.76	55	1.95	.14	2.24	2.82	.12	2.38	5.02	.19	8.89
32	2.15	.06	1.59	2.45	.11	2.08	4.56	.32	9.23	70	2.18	.16	2.57	3.04	.14	2.81	5.40	.45	9.95
34	2.28	.11	1.75	2.68	.14	2.37	5.21	.42	11.7	75	2.40	.18	2.92	3.36	.16	3.22	5.79	.52	10.7
36	2.42	.14	2.03	3.41	.18	3.96	6.86	.65	14.8	80	2.63	.20	3.80	3.47	.19	4.76	6.18	.59	12.0
38	2.56	.17	3.56	8.79	.22	4.79	6.81	.66	18.8	85	2.87	.23	4.69	3.69	.21	5.31	6.56	.67	13.4
40	2.69	.21	4.24	4.16	.27	5.71	7.16	.80	21.4	90	3.10	.26	5.21	4.91	.24	5.85	7.55	.75	17.5
42	4.02	.25	4.99	4.54	.38	6.22	7.81	.95	25.2	95	3.40	.28	5.92	5.50	.31	6.54	7.84	.84	18.1
44	4.45	.26	5.72	4.92	.38	6.79	8.47	1.11	29.5	100	3.64	.32	6.21	5.87	.31	7.72	9.33	.93	21.1
46	4.69	.34	6.04	5.30	.44	7.34	9.12	1.29	33.5	105	3.99	.39	6.53	6.33	.34	8.49	1.12	1.12	21.6
48	5.03	.39	7.05	5.68	.50	10.2	9.77	1.48	38.0	120	4.45	.46	6.98	8.21	.42	6.26	6.1	.23	25.4
50	5.26	.45	8.50	6.05	.57	11.4	10.4	1.7	42.8	130	5.90	.54	7.65	9.49	.49	7.26	7.71	.24	35.0
52	5.70	.50	9.51	6.43	.64	12.8	11.1	1.9	47.4	140	6.35	.62	8.28	8.87	.57	7.99	8.63	.28	42.8
54	6.03	.57	10.5	6.81	.72	13.9	11.7	2.1	52.5	150	6.81	.72	10.9	9.61	.66	8.56	9.21	.31	49.5
56	6.37	.63	11.7	7.19	.80	16.7	12.4	2.4	58.9	160	7.26	.83	11.9	9.78	.78	10.7	10.8	.34	56.0
58	6.70	.70	12.8	7.57	.89	17.3	13.0	2.6	64.7	170	8.45	.96	8.21	10.42	.82	8.85	9.27	.34	62.6
60	7.37	.84	15.5	8.38	1.08	21.8	14.3	3.2	71.7	180	9.08	.95	8.64	11.77	.95	10.8	10.8	1.6	75.4
62	8.04	1.00	18.0	9.08	1.28	24.2	15.6	3.8	77.7	190	9.99	1.05	9.55	12.49	1.04	11.5	11.8	1.8	82.0
64	8.71	1.13	20.9	9.84	1.30	26.1	16.9	4.4	84.7	200	10.9	1.18	10.4	13.77	1.17	12.5	12.5	2.1	88.5
66	9.38	1.27	23.9	10.6	1.7	32.2	18.3	5.1	102	210	11.8	2.2	29.2	11.3	2.0	26.6	12.6	2.4	94.4
68	10.0	1.6	27.3	11.3	2.0	36.7	19.5	6.9	137	220	12.7	2.5	33.5	12.2	2.3	30.0	14.6	3.8	122
70	10.7	1.8	30.7	12.1	2.8	41.2	20.8	6.7	164	230	13.6	2.9	38.9	13.0	2.6	34.1	15.6	3.8	130
72	11.4	2.0	34.3	12.9	2.6	45.1	22.2	7.7	173	240	14.5	3.8	42.8	13.9	3.0	38.4	17.1	5.5	136
74	12.1	2.3	36.1	13.6	2.9	45.3	23.4	8.5	192	250	15.4	3.7	47.9	14.8	3.4	43.0	17.7	6.7	142
76	12.7	2.5	32.1	14.4	3.2	56.7	24.7	9.5	212	260	16.3	4.1	53.3	15.6	3.8	47.8	19.7	7.4	149
78	13.4	2.8	46.5	15.1	3.5	63.3	26.1	10.6	232	270	17.3	4.6	59.9	16.6	4.3	51.8	21.8	8.4	154
80	14.7	3.4	50.6	16.7	4.3	74.7	28.7	12.8	276	280	18.3	5.1	64.7	17.4	4.7	58.8	24.8	10.4	162
82	16.1	4.0	54.4	18.2	5.1	87.8	31.3	15.2	327	290	19.1	5.7	64.5	18.2	5.1	63.5	26.2	12.7	170
84	17.4	4.7	78.3	19.7	6.0	101	38.9	17.9	379	300	20.9	6.8	83.8	20.0	6.2	76.3	31.2	15.1	178
86	18.8	6.5	82.3	21.2	7.0	116	36.5	20.7	406	310	21.8	7.4	87.8	21.2	7.4	80.9	36.5	19.6	186
88	20.1	6.3	96.1	22.7	8.0	129	29.1	23.8	464	320	21.8	7.4	96.7	20.8	6.7	81.3	21.4	214	196
90	23.5	8.6	129	26.5	10.9	125	45.6	32.3	527	330	22.7	8.0	97.5	21.7	7.3	98.6	23.3	226	206
92	26.8	11.2	167	30.2	14.8	101	32.1	23.2	541	340	22.7	8.0	105	22.9	8.2	91.1	34.0	18.0	215
94	30.2	14.3	250	34.1	18.1	279	32.1	27.9	560	350	22.7	8.0	109	23.2	8.9	96.5	36.5	19.6	225
96	33.5	17.4	252	37.9	22.3	340	32.1	24.2	579	360	22.7	8.0	117	23.2	12.4	107	37.1	21.4	235

(continued)

TABLE 4-46—(continued)

**Friction Losses In Pipe; C = 100**  
**3½ Inch**

FLOW U.S. gal per min	Cast Iron			Std Wt Steel			Extra Strong Steel			Double Extra Strong Steel		
	3.5 inside dia			3.548" inside dia			3.364" inside dia			2.728" inside dia		
	Ve- lo- city ft per sec	Ve- lo- city head ft	Head loss ft per 100 ft									
15	.50	.00	.061	.49	.00	.053	.54	.00	.075	.62	.01	.098
20	.67	.01	.120	.55	.01	.112	.72	.01	.145	1.10	.02	.403
25	.84	.01	.181	.61	.01	.168	.90	.01	.228	1.37	.03	.599
30	1.00	.02	.254	.77	.01	.237	1.08	.02	.308	1.65	.04	.893
35	1.17	.02	.357	1.14	.02	.316	1.26	.02	.409	1.93	.06	1.13
40	1.34	.03	.462	1.30	.03	.404	1.44	.03	.524	2.20	.08	1.46
45	1.51	.04	.577	1.46	.03	.505	1.68	.04	.655	2.47	.10	1.51
50	1.67	.04	.693	1.62	.04	.611	1.80	.04	.791	2.74	.12	1.59
55	2.01	.06	.914	1.95	.06	.856	2.17	.07	1.111	3.29	.17	2.07
60	2.24	.09	1.22	2.27	.08	1.14	2.53	.10	1.47	3.84	.23	2.49
65	2.63	.11	1.56	2.60	.11	1.46	2.89	.13	1.89	4.39	.30	3.23
70	3.01	.14	1.94	2.92	.13	1.61	3.25	.16	2.35	4.94	.38	3.67
75	3.33	.17	2.35	3.25	.16	2.20	3.61	.19	2.85	5.49	.47	4.21
80	3.65	.21	2.81	3.57	.20	2.63	3.97	.24	3.40	6.04	.57	4.83
85	4.02	.25	3.36	3.99	.25	3.08	4.33	.29	4.00	6.59	.67	5.41
90	4.35	.29	3.82	4.22	.28	3.53	4.69	.34	4.54	7.12	.79	6.05
95	4.68	.34	4.38	4.54	.32	4.10	5.05	.40	5.32	7.68	.92	7.01
100	5.02	.39	4.99	4.87	.37	4.57	5.41	.45	5.85	8.23	1.05	8.65
105	5.35	.44	5.61	5.19	.42	6.26	5.78	.53	6.81	8.78	1.20	10.79
110	5.69	.50	6.28	5.53	.47	6.35	6.14	.69	7.61	9.33	1.35	12.11
115	6.02	.56	6.98	5.85	.53	6.53	6.50	.66	8.46	9.88	1.52	22.9
120	6.35	.63	7.71	6.17	.59	7.32	6.85	.73	9.35	10.4	1.7	25.8
125	6.69	.70	8.46	6.50	.66	7.94	7.32	.81	10.4	11.0	1.9	34.6
130	7.85	.84	10.1	7.14	.79	9.47	7.94	.98	12.1	12.1	2.3	44.9
135	8.03	1.00	11.9	7.79	.94	11.1	8.66	1.17	14.4	12.2	2.7	39.9
140	8.70	1.18	13.8	8.44	1.11	12.9	9.28	1.37	16.7	14.3	3.2	46.3
145	9.36	1.36	15.8	9.09	1.28	14.1	10.1	1.5	19.2	14.4	5.1	53.1
150	10.0	1.55	18.9	9.74	1.47	16.5	10.8	1.8	21.8	16.5	4.2	60.4
155	10.7	1.8	20.5	10.4	1.7	18.6	11.5	2.1	24.5	17.6	5.3	67.0
160	11.4	2.0	22.6	11.0	1.9	21.2	12.3	2.4	27.4	18.7	5.4	76.1
165	12.0	2.2	25.2	11.7	2.1	25.5	13.0	2.6	30.5	19.8	6.1	84.6
170	12.7	2.5	27.8	12.8	2.4	26.0	13.7	2.9	33.7	20.9	6.8	95.0
175	13.4	2.8	30.5	13.0	2.6	28.6	14.4	3.2	37.1	22.0	7.5	107.0
180	14.0	3.1	32.6	13.6	2.9	31.3	15.2	3.6	40.6	23.0	8.2	112.5
185	14.7	3.4	35.6	14.3	3.2	34.1	15.9	3.9	44.2	24.1	9.0	125.0
190	15.4	3.7	37.6	14.9	3.6	37.1	16.6	4.3	48.3	26.3	9.9	135.0
195	16.1	4.0	42.2	16.6	3.8	40.1	17.3	4.7	51.9	26.8	10.7	144.0
200	16.7	4.5	46.2	16.3	4.1	43.3	18.1	5.1	56.8	27.4	11.7	158.0
205	18.4	5.3	50.1	17.9	5.0	51.6	19.9	6.2	59.2	30.2	14.3	165.0
210	20.1	6.3	54.7	19.5	5.9	56.6	21.7	7.3	78.6	32.9	16.8	218.0
215	21.8	7.4	56.1	21.1	6.9	79.2	23.5	8.6	91.8	35.7	19.8	262.0
220	22.4	8.5	57.4	22.7	8.0	89.6	25.3	9.4	104	38.4	23.9	269.0
225	22.1	9.8	57.4	24.4	9.3	91.0	27.1	11.4	119	41.1	26.4	279.0
230	26.8	11.2	59.1	26.0	10.5	103	28.9	13.0	151	45.9	30.0	371.0
235	28.4	12.5	59.5	27.6	11.8	115	30.7	14.6	150	46.7	33.9	414.0

The National Bureau of Standards has recommended the elimination of this pipe size.

**Friction Losses In Pipe; C = 100**  
**4 Inch**

FLOW U.S. gal per min	Cast Iron			Std Wt Steel			Extra Strong Steel			Double Extra Strong Steel		
	4.0 inside dia			4.026" inside dia			3.826" inside dia			3.152" inside dia		
	Ve- lo- city ft per sec	Ve- lo- city head ft	Head loss ft per 100 ft									
20	.51	.00	.063	.50	.00	.061	.56	.00	.078	.52	.01	.199
25	.77	.01	.122	.76	.01	.120	.84	.01	.164	.83	.02	.422
30	1.02	.03	.226	1.01	.02	.219	1.12	.02	.280	1.15	.04	.719
35	1.28	.03	.341	1.26	.03	.336	1.40	.03	.422	1.27	.07	1.07
40	1.53	.04	.477	1.51	.04	.463	1.67	.04	.573	1.47	.10	1.32
45	1.79	.05	.635	1.76	.05	.615	1.95	.06	.789	1.88	.18	2.02
50	2.04	.06	.813	2.02	.06	.785	2.23	.08	1.01	2.29	.17	2.46
55	2.30	.08	1.01	2.27	.08	.988	2.51	.10	1.26	3.70	.21	3.21
60	2.55	.10	1.23	2.53	.10	1.19	2.79	.12	1.63	4.11	.26	3.91
65	2.81	.12	1.47	2.77	.12	1.42	3.07	.15	1.82	4.83	.32	4.87
70	3.06	.15	1.72	3.02	.14	1.67	3.35	.17	2.14	4.94	.38	5.49
75	3.32	.17	2.00	3.28	.17	1.93	3.63	.20	2.48	5.35	.44	6.36
80	3.57	.20	2.29	3.23	.19	2.22	3.91	.24	2.84	6.76	.52	7.39
85	3.83	.23	2.61	3.78	.22	3.53	4.19	.27	3.24	6.17	.59	8.31
90	4.08	.26	2.93	4.03	.25	3.84	4.47	.31	3.84	6.58	.67	9.34
100	4.34	.29	3.28	4.29	.29	3.18	4.75	.35	4.07	6.99	.76	11.5
105	4.59	.33	3.54	4.54	.32	3.52	5.20	.39	4.52	7.40	.83	11.9
110	4.86	.37	3.80	4.79	.36	3.70	5.80	.44	5.00	7.82	.95	12.8
115	5.11	.41	4.33	5.05	.40	4.29	5.88	.50	5.20	8.23	1.05	14.1
120	5.62	.49	5.28	5.85	.48	5.12	6.14	.58	6.04	9.05	1.27	16.8
125	6.13	.56	6.21	6.06	.57	6.01	6.70	.67	7.70	9.87	1.51	19.1
130	6.64	.69	7.26	6.58	.67	6.97	7.26	.72	8.95	10.7	1.8	22.9
135	7.16	.79	8.25	7.06	.77	7.82	8.25	.85	10.2	11.5	2.1	25.5
140	7.66	.91	9.38	7.57	.89	8.93	9.38	.98	10.9	12.3	2.4	27.2
145	8.17	1.04	10.6	8.07	1.01	10.2	9.94	1.24	13.1	13.2	2.7	31.7
150	8.68	1.17	11.8	8.58	1.14	11.5	9.80	1.40	14.7	14.0	3.0	37.7
155	9.19	1.31	13.7	9.08	1.28	13.0	10.0	1.6	14.8	14.5	3.4	41.9
160	9.70	1.46	14.5	9.59	1.48	14.1	10.6	1.7	15.0	15.6	3.8	46.3
165	10.2	1.6	15.6	10.1	1.6	15.2	11.2	1.9	15.8	16.5	4.2	50.7
170	10.7	1.8	17.6	10.6	1.7	16.9	11.7	2.1	17.7	17.3	4.7	55.7
175	11.2	2.1	20.7	11.6	2.1	20.8	12.8	2.8	21.7	18.9	5.6	60.7
180	12.3	2.3	22.4	12.1	2.3	21.7	13.4	3.2				

TABLE 4-46—(continued)

**Friction Losses In Pipe; C = 100**  
**5 Inch**

FLOW U.S. gal per min	Cast Iron			Std Wt Steel			Extra Strong Steel			Double Extra Strong Steel		
	5.0 inside dia			5.047" inside dia			4.513" inside dia			4.063" inside dia		
	Velocity ft per sec	Velocity head ft per 100 ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft per 100 ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft per 100 ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft per 100 ft	Head loss ft per 100 ft
20	.49	.00	.046	.48	.00	.043	.68	.00	.054	.75	.01	.124
40	.68	.01	.076	.64	.01	.073	.71	.01	.072	1.00	.02	.212
60	.82	.01	.116	.80	.01	.110	.88	.01	.129	1.24	.02	.320
80	.98	.01	.161	.96	.01	.154	1.06	.02	.154	1.49	.03	.445
100	1.14	.02	.214	1.12	.02	.205	1.23	.02	.208	1.74	.05	.576
120	1.31	.03	.276	1.28	.03	.262	1.41	.03	.330	1.99	.06	.743
140	1.47	.03	.341	1.44	.03	.326	1.59	.04	.411	2.24	.08	.948
160	1.63	.04	.415	1.60	.04	.396	1.76	.05	.499	2.49	.10	1.15
180	1.76	.06	.481	1.92	.06	.565	2.11	.07	.700	2.98	.14	1.61
200	2.29	.08	.773	2.24	.08	.739	2.47	.09	.931	3.48	.19	2.15
220	2.61	.11	.990	2.55	.10	.946	2.82	.12	1.19	3.98	.25	2.75
240	2.34	.13	1.23	2.88	.13	1.15	3.17	.16	1.48	4.48	.31	3.41
260	3.27	.16	1.50	3.20	.16	1.43	3.62	.19	1.86	4.98	.39	4.16
280	3.59	.20	1.75	3.52	.20	1.78	3.88	.23	2.15	5.47	.47	4.95
300	3.92	.24	2.10	3.85	.23	2.00	4.93	.28	2.62	5.97	.55	5.92
320	4.25	.28	2.43	4.17	.27	2.32	4.58	.33	2.92	6.47	.65	6.75
340	4.58	.33	2.79	4.49	.31	2.65	4.94	.38	3.35	6.97	.76	7.74
360	4.90	.38	3.17	4.81	.36	3.03	5.29	.43	3.81	7.46	.87	8.79
380	5.23	.43	3.57	5.18	.41	3.41	5.64	.49	4.29	7.96	.98	9.91
400	5.56	.48	3.99	5.45	.46	3.81	5.99	.56	4.80	8.46	1.11	11.1
420	5.89	.64	4.44	5.77	.52	4.24	6.35	.63	5.34	8.96	1.25	12.3
440	6.22	.60	4.90	6.09	.58	4.68	6.70	.70	5.90	9.45	1.39	15.6
460	6.54	.66	5.39	6.41	.64	5.15	7.05	.77	6.49	9.95	1.54	15.0
480	6.87	.78	5.90	6.78	.70	5.64	7.40	.85	7.10	10.4	1.7	16.4
500	7.20	.81	6.43	7.05	.77	6.14	7.76	.94	7.74	10.9	1.8	17.8
520	7.53	.88	6.98	7.38	.85	6.67	8.11	1.02	8.40	11.4	2.0	19.4
540	7.85	.96	7.65	7.70	.92	7.29	8.46	1.11	9.09	11.9	2.2	21.0
560	8.17	1.04	8.15	8.02	1.00	7.79	8.82	1.21	9.81	12.4	2.4	22.6
580	8.49	1.26	8.72	8.63	1.21	9.28	9.70	1.46	11.7	13.7	2.9	25.6
600	9.80	1.49	11.7	9.62	1.49	10.9	10.6	1.7	13.7	14.9	8.5	25.7
620	10.6	1.7	13.2	10.4	1.7	12.6	11.5	2.1	15.9	16.2	4.1	35.8
640	11.4	2.0	15.2	11.2	1.9	12.8	12.4	2.4	18.3	17.4	4.7	42.2
660	12.2	2.4	17.2	12.0	2.2	16.5	13.2	2.7	20.8	18.7	5.4	47.9
680	13.1	2.7	19.4	12.8	2.5	18.6	14.1	3.1	23.4	19.9	6.2	54.8
700	13.9	3.0	21.7	13.6	2.9	20.8	15.0	3.5	26.2	21.1	6.9	60.4
720	14.7	3.4	24.2	14.4	3.2	21.1	15.9	3.9	29.1	22.4	7.8	67.1
740	15.5	3.7	26.7	15.2	3.6	25.6	16.7	4.3	32.2	23.6	8.7	75.9
760	16.3	4.1	29.4	16.0	4.0	28.1	17.6	4.8	35.4	24.9	9.6	81.6
780	18.0	5.0	35.0	17.6	4.8	33.5	19.4	5.8	42.2	27.4	11.7	97.3
800	19.6	6.0	41.1	19.2	5.7	35.3	21.1	6.9	45.6	29.8	18.8	114
820	21.2	7.0	47.7	20.8	6.7	45.6	22.9	8.2	57.4	32.3	16.2	133
840	22.9	8.1	54.7	22.4	7.8	52.2	24.7	9.5	55.2	34.8	18.8	162
860	24.5	9.3	62.2	24.0	9.0	59.2	26.4	10.8	74.8	37.3	21.6	175
880	26.1	10.6	70.1	25.6	10.2	66.9	28.2	12.4	84.3	39.8	24.6	195
900	27.8	12.0	78.4	27.2	11.6	74.9	30.0	14.0	94.4	42.8	27.8	210

**Friction Losses In Pipe; C = 100**  
**6 Inch**

FLOW U.S. gal per min	Cast Iron			Std Wt Steel			Extra Strong Steel			Double Extra Strong Steel		
	6.0 inside dia			6.065" inside dia			5.761" inside dia			4.897" inside dia		
	Velocity ft per sec	Velocity head ft per 100 ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft per 100 ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft per 100 ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft per 100 ft	Head loss ft per 100 ft
60	.67	.01	.047	.56	.00	.045	.62	.01	.055	.66	.01	.127
60	.68	.01	.046	.57	.01	.045	.74	.01	.061	1.02	.02	.178
70	.79	.01	.058	.78	.01	.054	.86	.01	.068	1.19	.02	.257
70	.91	.02	.113	.89	.01	.107	.98	.01	.138	1.36	.03	.314
70	1.03	.02	.141	1.00	.02	.133	1.11	.02	.171	1.53	.04	.373
100	1.13	.02	.171	1.11	.02	.162	1.23	.02	.208	1.70	.05	.469
120	1.36	.03	.239	1.83	.03	.227	1.48	.03	.292	2.04	.06	.643
140	1.59	.04	.318	1.56	.04	.302	1.72	.05	.388	2.38	.09	.856
160	1.82	.05	.406	1.78	.05	.387	1.97	.06	.497	2.72	.12	1.16
180	2.04	.06	.507	2.00	.06	.481	2.22	.08	.618	3.06	.15	1.36
200	2.27	.08	.616	2.22	.08	.584	2.46	.09	.751	3.40	.18	1.66
220	2.50	.10	.719	2.44	.09	.657	2.71	.11	.856	3.74	.22	1.97
240	2.72	.12	.813	2.67	.11	.719	2.96	.14	.953	4.08	.26	2.32
260	2.95	.14	1.00	2.89	.13	.950	3.20	.16	1.522	4.42	.30	2.69
280	3.18	.16	1.15	3.11	.15	.945	3.43	.19	1.40	4.77	.35	3.08
300	3.40	.18	1.30	3.38	.17	1.24	3.69	.21	1.55	5.11	.41	3.60
320	3.64	.21	1.47	3.56	.20	1.39	3.94	.24	1.79	5.45	.46	3.92
340	3.86	.23	1.64	3.78	.22	1.58	4.19	.27	2.00	5.79	.52	4.42
360	4.08	.26	1.83	4.00	.25	1.73	4.43	.31	2.28	6.13	.58	4.91
380	4.31	.29	2.02	4.22	.28	1.92	4.68	.34	2.46	6.47	.65	5.43
400	4.55	.32	2.22	4.44	.31	2.11	4.93	.38	2.71	6.81	.72	5.97
420	5.11	.41	2.76	5.00	.39	3.62	5.54	.48	3.36	7.66	.91	7.42
440	5.68	.50	3.36	5.56	.48	4.19	6.16	.59	4.09	8.51	1.13	9.02
460	6.26	.61	4.00	6.11	.58	4.88	6.77	.71	4.88	9.37	1.36	10.6
480	6.81	.72	4.70	6.66	.69	4.46	7.39	.85	5.73	10.3	1.6	12.6
500	7.38	.85	5.45	7.22	.81	5.17	8.00	.99	6.64	11.1	1.9	14.6
520	7.95	.98	6.25	7.78	.94	5.93	8.63	1.16	7.62	11.9	2.2	16.8
540	8.52	1.13	7.10	8.34	1.03	6.74	9.24	1.33	8.66	12.8	2.5	19.1
560	9.08	1.28	8.00	8.90	1.23	7.50	9.85	1.51	9.76	13.6	2.9	21.5
580	9.65	1.45	8.95	9.46	1.39	8.50	10.5	1.7	10.9	14.5	3.3	24.1
600	10.2	1.6	9.76	10.0	1.6	9.44	11.1	1.9	12.1	15.3	3.7	26.7
620	10.8	1.8	11.0	10.5	1.7	10.2	11.7	2.1	13.4	16.3	4.1	29.6
640	11.4	2.0	12.1	11.1	1.9	11.5	12.3	2.4	14.7	17.0	4.5	32.6
660	12.5	2.4	14.4	12.2	2.3	13.7	13.5	2.8	17.6	18.7	5.4	35.6
680	13.6	2.9	16.9	13.7	2.7	16.1	14.8	3.4	20.7	20.4	4.5	45.5
700	14.8	3.4	19.7	14.4	3.2	18.6	16.0	4.0	2			

TABLE 4-46—(continued)

**Friction Losses In Pipe; C = 100**  
**8 Inch**

FLOW U S gal per min	Cast Iron			Std Wt Steel			Extra Strong Steel			Double Extra Strong Steel		
	8.0 inside dia			7.981" inside dia			7.625" inside dia			6.375" inside dia		
	Velocity ft per sec	V- locity head ft per 100 ft	Head loss ft per 100 ft	Velocity ft per sec	V- locity head ft per 100 ft	Head loss ft per 100 ft	Velocity ft per sec	V- locity head ft per 100 ft	Head loss ft per 100 ft	Velocity ft per sec	V- locity head ft per 100 ft	Head loss ft per 100 ft
120	.88	.01	.069	.82	.01	.069	.91	.01	.067	1.12	.02	.143
140	.90	.01	.079	.90	.01	.079	.98	.01	.077	1.21	.02	.144
160	.95	.01	.089	.96	.01	.089	1.06	.02	.092	1.30	.03	.145
180	1.03	.02	.101	1.03	.02	.102	1.12	.03	.107	1.38	.03	.210
200	1.08	.02	.112	1.09	.02	.114	1.19	.03	.142	1.47	.03	.235
220	1.15	.02	.125	1.15	.02	.126	1.26	.03	.153	1.56	.04	.261
240	1.21	.02	.138	1.22	.02	.140	1.33	.03	.176	1.64	.04	.275
260	1.28	.03	.152	1.28	.03	.154	1.41	.03	.192	1.73	.06	.318
280	1.40	.03	.161	1.41	.03	.162	1.55	.04	.204	1.90	.06	.342
300	1.53	.04	.178	1.64	.04	.176	1.69	.04	.267	2.07	.07	.445
320	1.66	.04	.247	1.67	.04	.250	1.88	.05	.312	2.25	.08	.516
340	1.79	.05	.263	1.80	.05	.266	1.97	.06	.358	2.42	.09	.592
360	1.91	.06	.272	1.92	.06	.275	2.11	.07	.406	2.59	.10	.672
380	2.24	.08	.426	2.24	.08	.425	2.46	.09	.548	3.03	.14	.924
400	2.55	.10	.548	2.57	.10	.553	2.81	.12	.622	3.46	.19	1.14
420	2.87	.13	.681	2.88	.13	.682	3.16	.15	.760	3.89	.24	1.42
440	3.19	.16	.828	3.20	.16	.829	3.51	.19	.905	4.32	.29	1.73
460	3.51	.19	.987	3.52	.19	.989	3.86	.23	1.25	4.76	.35	2.06
480	3.83	.23	1.16	3.85	.23	1.17	4.22	.28	1.45	5.19	.42	2.42
500	4.15	.27	1.34	4.17	.27	1.36	4.57	.32	1.70	5.62	.49	2.78
520	4.47	.31	1.54	4.49	.31	1.54	4.92	.35	1.75	6.05	.57	3.22
540	4.79	.36	1.76	4.81	.36	1.77	5.27	.45	2.17	6.49	.65	3.56
560	5.11	.41	1.97	5.13	.41	1.99	5.62	.49	2.49	6.91	.74	4.13
580	5.43	.46	2.21	5.45	.46	2.23	5.97	.55	2.79	7.35	.84	4.62
600	5.75	.51	2.46	5.77	.52	2.48	6.33	.62	3.10	7.78	.94	5.13
620	6.06	.57	2.71	6.09	.58	2.74	6.67	.69	3.43	8.21	1.05	5.67
640	6.38	.63	2.98	6.41	.64	2.95	7.03	.77	3.77	8.64	1.16	6.24
660	6.70	.77	3.59	7.05	.77	3.60	7.88	.95	4.49	9.50	1.40	6.74
680	7.03	.77	3.59	7.05	.77	3.60	8.28	1.10	5.28	10.4	1.7	7.74
700	7.36	1.07	4.08	7.69	9.2	4.23	8.48	1.10	5.48	10.4	1.0	10.4
720	7.66	1.24	5.56	8.97	1.26	5.62	9.83	1.50	7.02	12.1	2.8	11.6
740	7.98	1.43	6.39	9.61	1.44	6.39	10.5	1.7	7.98	13.0	2.6	13.2
760	8.11	1.43	6.39	9.61	1.44	6.39	10.5	1.7	7.98	13.0	2.6	13.2
780	8.33	1.43	6.39	9.61	1.44	6.39	10.5	1.7	7.98	13.0	2.6	13.2
800	8.55	1.43	6.39	9.61	1.44	6.39	10.5	1.7	7.98	13.0	2.6	13.2
820	8.77	1.72	10.3	1.7	7.28	11.2	8.0	8.99	13.8	8.0	14.9	
840	10.2	1.6	8.12	11.8	2.1	8.96	12.6	2.5	11.5	15.6	2.8	16.5
860	11.5	2.1	8.85	11.8	2.1	8.96	14.1	3.1	15.6	18.6	3.8	20.5
880	12.8	2.6	10.8	15.8	2.5	10.9	14.1	3.1	15.6	17.3	4.7	22.6
900	14.1	3.1	12.8	14.1	3.1	13.0	15.8	3.7	16.6	19.0	5.6	26.0
920	15.3	3.6	15.1	15.4	3.7	15.2	16.9	4.4	19.2	20.7	6.7	31.5
940	16.6	4.3	17.6	16.7	4.3	17.7	18.3	5.2	22.1	23.5	7.9	36.5
960	17.9	5.0	20.0	18.0	5.0	20.3	19.7	6.0	25.3	24.2	9.1	41.9
980	19.1	5.7	22.8	19.2	5.7	23.0	21.1	6.9	28.8	25.9	10.4	47.6
1000	22.4	7.8	30.3	22.6	7.8	20.6	24.6	9.4	30.3	20.2	14.2	33.3
1020	25.6	10.2	35.6	25.6	10.2	37.2	28.1	12.3	45.0	34.6	18.6	61.0
1040	28.7	12.8	40.2	28.8	12.9	40.8	31.6	15.6	50.2	38.9	35.6	101
1060	31.9	15.8	48.6	32.0	15.9	49.3	35.1	19.1	74.0	43.2	39.0	122
1080	35.1	19.1	63.9	35.2	19.4	70.7	38.6	23.2	88.3	47.6	35.1	146

**Friction Losses In Pipe; C = 100**  
**10 Inch**

FLOW U S gal per min	Cast Iron			Standard Wt Steel			Extra Strong Steel		
	10.0" inside dia			10.02" inside dia			9.750" inside dia		
	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
180	.74	.01	.042	.78	.01	.042	.77	.01	.042
200	.82	.01	.051	.81	.01	.051	.86	.01	.051
220	.90	.01	.061	.89	.01	.061	.95	.01	.051
240	.98	.01	.072	.98	.01	.071	1.03	.02	.051
260	1.06	.02	.083	1.06	.02	.083	1.13	.02	.051
280	1.14	.02	.095	1.14	.02	.095	1.20	.02	.051
300	1.22	.03	.108	1.22	.03	.108	1.29	.03	.051
320	1.30	.03	.144	1.30	.03	.143	1.50	.04	.051
340	1.38	.04	.165	1.38	.04	.163	1.72	.06	.051
360	1.46	.04	.200	1.46	.04	.200	1.93	.06	.051
380	1.54	.05	.238	1.54	.05	.238	2.15	.07	.051
400	2.04	.08	.330	2.04	.08	.330	2.36	.09	.077
420	2.24	.09	.352	2.24	.09	.352	2.58	.10	.077
440	2.45	.11	.384	2.45	.11	.384	2.79	.12	.077
460	2.65	.11	.404	2.65	.11	.404	2.99	.12	.077
480	2.86	.13	.421	2.86	.13	.421	3.19	.14	.077
500	3.26	.17	.667	3.26	.17	.667	3.46	.19	.074
520	3.67	.21	.729	3.66	.21	.721	3.87	.23	.073
540	4.08	.26	.776	4.07	.26	.766	4.30	.24	.114
560	4.49	.31	1.171	4.49	.31	1.171	4.73	.35	.114
580	4.90	.37	1.41	4.89	.37	1.40	5.16	.41	.113
600	5.31	.44	1.64	5.30	.44	1.62	5.59	.49	.113
620	5.71	.51	1.88	5.70	.51	1.86	6.01	.56	.112
640	6.12	.58	2.13	6.10	.58	2.11	6.44	.64	.241
660	6.53	.65	2.40	6.51	.65	2.38	6.88	.74	.242
680	6.94	.75	2.69	6.92	.74	2.66	7.30	.83	.304
700	7.35	.84	2.99	7.32	.83	2.96	7.74	.93	.338
720	7.76	.94	3.30	7.73	.93	3.27	8.16	1.03	.374
740	8.16	1.03	3.53	8.14	1.03	3.50	8.60	1.15	.411
760	8.56	1.28	4.23	8.55	1.24	4.20	9.45	1.39	.450
780	9.00	1.49	5.09	9.76	1.48	5.04	10.3	1.6	.576
800	10.6	1.7	5.90	10.6	1.7	5.84	11.2	1.9	.677
820	11.4	2.0	6.77	11.4	2.0	6.70	12.0	2.2	.755
840	12.2	2.3	7.69	12.2	2.3	7.61	12.9	2.6	.879
860	12.9	2.7	8.66	12.8	2.7	8.58	13.8	3.0	.939
880	13.1	3.0	9.69	13.8	3.0	9.60	14.6	3.8	11.8
900	14.7	3.4	10.8	14.6	3.3	10.7	15.5	3.7	12.9
920	15.5	3.7	11.9	15.5	3.7	11.8	16.3	4.1	13.5
940	16.3	4.1	13.1	16.3	4.1	13.0	17.2	4.6	14.3
960	18.4	5.3	16.3	18.3	5.3	16.1	19.3	5.8	16.1
980	20.4	6.6	19.8	20.3	6.4	19.6	21.5	7.9	17.1
5000	22.4	7.8	23.6	22.4	7.8	23.4	23.6	8.7	26.7

TABLE 4-46—(continued)

Friction Losses In Pipe; C = 100 12 Inch										Friction Losses In Pipe; C = 100 14 Inch										Friction Losses In Pipe; C = 100 16 Inch									
FLOW U.S. gal per min	Cast Iron			Standard Wt Steel			Extra Strong Steel			FLOW U.S. gal per min	Cast Iron			Steel			FLOW U.S. gal per min	Cast Iron			FLOW U.S. gal per min	Cast Iron			Steel				
	12.0" inside dia			12.000" inside dia			11.750" inside dia				14.0" inside dia			13.25" inside dia				16.0" inside dia				15.25" inside dia							
	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft		Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft		Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft		Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft					
200	.67	.01	.921	.67	.01	.921	.69	.01	.923	500	.63	.01	.921	.70	.01	.928	600	.60	.01	.926	700	.56	.01	.936					
400	.71	.01	.922	.71	.01	.922	.74	.01	.926	1000	.64	.01	.936	.73	.01	.942	1200	.65	.01	.945	1400	.62	.01	.950					
600	.65	.01	.923	.65	.01	.923	.69	.01	.928	1500	1.04	.03	.954	1.16	.02	.971	1700	1.12	.02	.982	1900	1.23	.02	.987					
800	.59	.01	.924	.59	.01	.924	.63	.01	.929	2000	1.25	.03	.976	1.40	.03	.997	2200	1.28	.03	.998	2400	1.41	.03	.999					
1000	.53	.01	.925	.53	.01	.925	.57	.01	.930	2600	1.46	.03	.998	1.63	.04	.999	2800	1.44	.03	.999	3000	1.58	.04	.999					
1200	1.28	.03	.926	1.28	.03	.926	1.33	.03	.933	3200	1.67	.04	.999	1.80	.05	.999	3400	1.60	.04	.999	3600	1.76	.05	.999					
1400	1.42	.03	.927	1.42	.03	.927	1.48	.03	.938	3800	1.88	.05	.999	2.05	.07	.999	4000	1.92	.06	.999	4200	2.11	.07	.999					
1600	1.56	.03	.928	1.56	.03	.928	1.63	.03	.944	4400	2.09	.07	.999	2.23	.08	.999	4600	2.14	.08	.999	4800	2.24	.09	.999					
1800	1.70	.03	.929	1.70	.03	.929	1.77	.03	.950	5000	2.30	.08	.999	2.46	.10	.999	5200	2.56	.10	.999	5400	2.61	.12	.999					
2000	1.99	.06	.930	1.99	.06	.930	2.07	.07	.957	5600	2.50	.10	.999	2.79	.12	.999	5800	2.87	.13	.999	6000	3.16	.16	.999					
2500	2.27	.08	.935	2.27	.08	.935	2.37	.09	.964	6200	2.71	.11	.999	3.02	.14	.999	6400	3.19	.16	.999	6600	3.61	.19	.999					
3000	2.55	.10	.941	2.55	.10	.941	2.66	.11	.970	6800	2.92	.13	.999	3.26	.17	.999	7000	3.49	.20	.999	7200	3.86	.23	.999					
3500	2.84	.13	.946	2.84	.13	.946	2.96	.14	.976	7400	3.13	.15	.999	3.49	.19	.999	7600	4.79	.24	.999	7800	5.27	.28	.999					
4000	3.12	.15	.950	3.12	.15	.950	3.25	.15	.982	8000	3.34	.17	.999	3.72	.22	.999	8200	5.58	.35	.999	8400	6.18	.39	.999					
4500	3.41	.18	.951	3.41	.18	.951	3.55	.20	.984	8600	3.54	.19	.999	3.95	.34	.999	8800	6.38	.43	.999	9000	7.03	.49	.999					
5000	3.69	.21	.954	3.69	.21	.954	3.84	.23	.987	9200	3.75	.22	.999	4.19	.27	.999	9400	7.18	.49	.999	9600	7.91	.57	.999					
5500	3.98	.25	.957	3.98	.25	.957	4.14	.27	.991	9600	3.96	.24	.999	4.42	.30	.999	9800	7.98	.51	.999	10000	8.79	.62	.999					
6000	4.26	.28	.960	4.26	.28	.960	4.44	.31	.995	10200	4.17	.27	.999	4.65	.34	.999	10400	9.85	.63	.999	10600	10.5	.7	.999					
6500	4.55	.32	.962	4.55	.32	.962	4.78	.35	.999	10800	5.23	.42	.999	5.81	.52	.999	11000	11.2	.9	.999	11200	12.3	.94	.999					
7000	5.11	.41	1.23	5.11	.41	1.23	5.38	.44	1.36	11200	6.26	.61	1.50	6.39	.76	1.97	11400	12.8	2.5	4.73	11600	14.1	8.1	6.96					
7500	5.68	.50	1.50	5.68	.50	1.50	5.92	.54	1.66	11600	7.30	.68	1.99	8.18	1.03	2.60	11800	14.4	3.2	5.96	12000	15.8	3.9	7.63					
8000	6.25	.61	1.78	6.25	.61	1.78	6.51	.66	1.88	12000	8.34	1.08	2.55	9.81	1.86	3.32	12200	16.0	4.0	7.26	12400	17.6	4.8	8.16					
8500	6.81	.72	2.15	6.81	.72	2.15	7.10	.78	2.32	12400	11.4	1.37	3.17	10.5	2.17	4.15	12600	17.5	4.8	8.54	12800	19.3	5.8	10.9					
9000	7.38	.85	2.45	7.38	.85	2.45	7.69	.92	2.55	13000	10.4	1.7	3.95	11.6	2.1	5.03	13200	19.2	6.7	10.2	13400	21.1	6.9	12.8					
9500	7.95	.98	2.78	7.95	.98	2.78	8.28	1.07	3.09	13600	12.5	2.4	5.39	14.0	3.0	7.05	13800	20.8	6.7	11.8	14000	22.8	8.1	14.9					
10000	8.53	1.13	3.17	8.53	1.13	3.17	8.88	1.23	3.51	14000	14.6	2.3	7.17	16.3	4.1	9.38	14200	24.6	9.4	17.1	14400	25.5	10.7	18.2					
10500	9.10	1.54	4.21	9.10	1.54	4.21	10.3	1.6	4.67	14600	16.7	4.3	9.18	18.6	5.4	12.9	15000	24.0	9.0	15.3	15200	25.6	10.2	17.3					
11000	9.67	2.0	5.39	11.4	3.0	6.39	11.8	2.2	6.97	15000	18.8	5.5	11.4	20.9	6.8	14.9	15400	25.6	10.2	17.3	15800	28.1	12.3	21.8					
11500	10.24	2.5	6.79	12.8	2.5	6.79	13.3	2.7	7.43	16000	20.9	6.8	15.7	23.3	8.4	18.1	16200	28.7	12.8	21.5	16600	31.6	15.5	27.1					
12000	10.81	3.1	8.15	14.2	3.1	8.15	14.8	3.4	9.03	16400	23.0	8.2	16.7	26.6	10.2	21.6	16800	31.9	16.8	26.1	17200	35.1	19.1	33.0					
12500	11.38	3.72	15.6	8.8	9.72	16.8	4.1	10.8	17000	25.0	9.7	19.4	27.9	12.1	26.4	17400	35.1	19.1	31.2	17800	38.7	28.3	39.3						
13000	11.95	4.5	11.4	17.0	4.5	17.0	17.7	4.9	12.6	17400	27.1	11.4	22.5	30.2	14.2	29.5	17800	38.3	28.8	36.6	18200	42.2	27.7	42.9					
13500	12.42	5.2	13.2	18.4	5.3	18.2	19.2	5.7	14.7	18000	29.2	12.3	25.9	32.6	16.5	32.3	18400	38.9	24.7	38.6	18800	45.9	30.0	45.9					
14000	12.89	6.2	15.2	19.9	6.2	15.2	20.7	6.7	16.8	18600	31.3	14.2	27.4	34.9	18.9	35.4	19000	41.5	26.8	42.4	19400	48.7	33.6	48.9					
14500	13.36	7.1	17.3	21.3	7.1	17.3	22.2	7.7	19.1	19200	33.4	17.3	33.1	37.2	21.5	43.3	19600	44.7	31.1	48.7	20000	49.3	37.6	61.0					
15000	13.83	8.0	19.4	22.7	8.0	19.4	23.7	8.7	21.5	19800	35.4	19.5	37.0	39.5	34.2	48.4	20200	47.9	35.7	65.3	20600	53.7	43.2	69.8					
15500	14.30	9.1	21.7	24.2	9.1	21.7	25.1	9.8	24.1	20800	37.5	21.6	41.2	41.9	27.3	53.8	21200	51.1	40.5	62.3	21600	56.2	49.1	70.7					
16000	14.77	10.2	24.2	26.6	10.2	24.2	26.6	11.0	26.0	21800	41.7	27.0	50.0	46.5	33.6	55.4	22200	54.3	46	69.9	22600	59.8	55.6	69.3					
16500	15.24	11.3	26.7	27.0	11.3	26.7	28.1	12.3	29.5	22800	45.9	31.7	59.7	51.2	40.7	70.0	23200	57.5	51	77.5	23600	63.8	62.3	77.8					
17000	15.71	12.5	29.4	28.4	12.5	29.4	29.6	13.6	32.5	24800	49.0	38.9	70.1	55.8	48.4	91.6	25200	60.7	57	80.6	25600	66.8	69.3	80.8					
17500	16.18	15.1	35.0	31.2	15.1	35.0	32.5	16.4	38.5	26800	51.2	41.7	59.7	54.5	49.6	91.6	28200	66.8	69.3	80.6	28600	73.3	69.3	80.8					
18000	16.65	18.1	41.4	34.1	18.1	41.4	35.5	19.6	45.6	28800	53.4	44.7	60.0	58.0	53.8	91.6	30200	73.3	43.2	69.8	30600	80.6	69.3	80.8					
18500	17.12	21.2	47.7	36.9	21.2	47.7	38.4	22.9	52.5	30800	55.6	47.7	61.2	60.0	58.0	91.6	32200	73.3	40.5	69.8	32600	80.6	69.3	80.8					

TABLE 4-46—(continued)

Friction Losses In Pipe; C = 100												Friction Losses in Pipe; C = 100											
18 Inch						20 Inch						24 in. inside dia						30 in. inside dia					
FLOW U S gal per min	Cast Iron			Steel			FLOW U S gal per min	Cast Iron			Steel			Discharge in U S gallons per min	Velocity feet per sec	Velocity head in ft	Head loss in feet per 100 ft	Discharge in U S gallons per min	Velocity feet per sec	Velocity head in ft	Head loss in feet per 100 ft		
	18.0° inside dia			17.18° inside dia				20.0° inside dia			19.18° inside dia												
	Ve- loc- ity ft per sec	Ve- loc- ity ft per sec	Head loss ft per 100 ft	Ve- loc- ity ft per sec	Ve- loc- ity ft per sec	Head loss ft per 100 ft	Ve- loc- ity ft per sec	Ve- loc- ity ft per sec	Ve- loc- ity ft per sec	Head loss ft per 100 ft	Ve- loc- ity ft per sec	Ve- loc- ity ft per sec	Head loss ft per 100 ft	per min	per 24 hr	per min	per min	per 24 hr	per min	per min	per 100 ft		
500	.63	.01	.016	.69	.01	.020	500	.82	.01	.023	.89	.01	.028	500	506,000	.252	.00	.082	700	1,008,000	.323	.00	.002
1000	.72	.01	.022	.83	.01	.028	1000	1.02	.02	.035	1.11	.02	.042	1000	1,008,000	.495	.00	.087	1800	1,440,000	.446	.00	.004
1500	.89	.01	.030	.97	.01	.047	1500	1.23	.02	.048	1.33	.03	.059	1500	2,016,000	.999	.01	.104	2700	1,872,000	.589	.01	.068
2000	1.01	.02	.036	1.11	.02	.049	2000	1.43	.03	.044	1.55	.04	.079	2000	2,448,000	1.21	.02	.038	3600	2,448,000	.78	.01	.013
2500	1.13	.02	.048	1.25	.02	.060	2500	1.64	.04	.062	1.75	.05	.101	2500	2,880,000	.91	.01	.017	4200	2,880,000	.91	.01	.017
3000	1.26	.02	.068	1.38	.03	.072	3000	1.83	.06	.102	2.00	.06	.126	3000	3,356,000	1.43	.03	.051	4800	3,456,000	1.095	.02	.023
3500	1.38	.04	.087	1.66	.04	.101	3500	2.04	.06	.125	2.22	.06	.163	3500	3,846,000	1.70	.045	.071	5400	3,846,000	1.23	.02	.038
4000	1.78	.06	.106	1.94	.06	.135	4000	2.55	.10	.188	2.78	.12	.231	4000	4,464,000	1.92	.06	.089	5800	4,464,000	1.48	.08	.039
4500	2.03	.06	.138	2.21	.08	.173	4500	3.06	.15	.244	3.33	.17	.323	4500	4,896,000	2.21	.08	.114	6400	4,896,000	1.55	.04	.046
5000	2.27	.08	.171	2.49	.10	.215	5000	3.57	.20	.351	3.89	.24	.430	5000	5,472,000	2.41	.09	.136	6800	5,472,000	1.74	.06	.057
5500	2.52	.10	.203	2.77	.12	.261	5500	4.08	.26	.449	4.45	.31	.551	5500	6,048,000	2.7	.113	.167	7400	6,048,000	1.87	.06	.065
6000	2.16	.12	.214	3.46	.19	.374	6000	5.10	.40	.679	5.55	.48	.852	6000	6,480,000	2.99	.13	.200	7800	6,480,000	2.05	.07	.077
6500	3.78	.22	.440	4.15	.27	.565	6500	6.13	.58	.951	6.67	.69	1.17	6500	6,912,000	3.20	.16	.225	8200	6,912,000	2.19	.09	.101
7000	4.41	.30	.566	4.85	.37	.735	7000	7.15	.79	1.26	7.78	.94	1.55	7000	7,488,000	3.69	.21	.298	8800	7,488,000	2.61	.10	.112
7500	5.04	.39	.708	5.54	.49	.941	7500	8.17	1.0	1.62	8.89	1.3	1.98	7500	7,920,000	3.91	.24	.329	9200	8,496,000	2.69	.11	.127
8000	5.67	.50	.932	6.23	.60	1.17	8000	10.2	1.6	2.45	11.1	1.9	3.06	8000	8,496,000	4.19	.27	.377	9800	8,496,000	2.88	.12	.135
8500	6.30	.62	1.152	6.92	.74	1.42	8500	12.3	2.4	3.43	13.3	2.7	4.20	8500	9,336,000	4.41	.32	.413	10400	9,336,000	3.12	.15	.170
9000	7.56	.69	1.379	8.31	1.1	1.75	9000	14.3	3.2	4.66	15.5	3.7	6.51	9000	9,340,000	4.63	.38	.456	11000	10,944,000	3.47	.18	.243
9500	8.83	1.2	2.11	9.70	1.5	2.45	9500	15.8	3.6	5.18	16.7	4.8	6.34	9500	10,944,000	5.40	.50	.605	11600	12,960,000	4.42	.306	.319
10000	10.1	1.6	2.70	11.1	1.9	3.39	10000	16.3	4.1	5.84	17.8	4.9	7.16	10000	11,952,000	5.90	.54	.713	12200	13,968,000	4.64	.32	.328
10500	11.3	2.0	3.36	12.5	2.4	4.22	10500	18.4	5.3	7.26	20.0	6.2	8.26	10500	12,960,000	6.9	.74	.96	13800	15,968,000	5.01	.392	.401
11000	12.6	2.5	4.08	13.8	3.0	5.12	11000	20.4	6.5	8.52	22.3	7.7	10.8	11000	15,968,000	7.11	.79	.100	14400	17,280,000	5.47	.47	.473
11500	13.8	3.6	5.72	15.6	4.3	7.18	11500	22.5	7.9	10.5	24.4	9.3	12.9	11500	16,840,000	7.82	.95	.119	12500	18,000,000	5.10	.51	.51
12000	14.8	4.9	6.91	19.4	5.8	9.56	12000	24.5	9.3	12.4	26.7	11.1	15.1	12000	17,280,000	8.55	1.14	1.40	13000	18,720,000	5.94	.55	.55
12500	16.3	6.4	9.74	23.1	7.6	12.2	12500	25.5	10.1	13.3	27.8	12.0	16.3	12500	18,720,000	8.86	1.22	1.52	14000	20,160,000	6.4	.64	.63
13000	22.7	8.0	12.1	24.9	9.6	15.2	13000	26.6	11.0	14.3	28.9	13.0	17.6	13000	21,600,000	9.06	1.58	2.11	12500	18,000,000	5.70	.51	.51
13500	25.3	9.9	14.7	27.7	11.9	18.5	13500	28.6	13.7	16.4	31.1	15.0	20.1	13500	23,040,000	11.38	2.03	2.47	13000	20,160,000	6.4	.64	.63
14000	27.7	11.9	17.6	30.2	14.3	22.8	14000	30.6	14.6	18.7	33.3	17.3	22.4	14000	23,480,000	12.90	2.57	2.98	14500	23,480,000	9.12	1.30	1.32
14500	30.6	14.6	28.6	33.2	17.1	25.9	14500	32.6	16.5	21.0	35.6	19.7	25.8	14500	27,360,000	13.50	2.85	3.28	15000	32,900,000	10.09	1.89	1.78
15000	30.6	14.6	28.6	33.2	17.1	25.9	15000	34.7	18.7	23.6	37.8	22.2	28.9	15000	27,360,000	14.20	3.16	3.61	15500	34,500,000	12.75	2.46	2.27
15500	35.5	19.6	27.4	38.8	23.4	34.4	15500	35.7	19.8	24.8	38.9	23.5	30.4	15500	31,600,000	10.06	1.58	2.11	16000	28,920,000	8.20	1.05	1.09
16000	37.8	22.2	31.2	41.5	26.5	39.1	16000	36.8	21.0	26.2	40.0	24.9	32.1	16000	33,040,000	11.38	2.03	2.47	16500	27,560,000	8.67	1.17	1.11
16500	40.4	25.6	35.1	44.8	30.6	44.1	16500	38.8	23.4	28.9	42.2	27.7	35.4	16500	35,280,000	12.90	2.57	2.98	17000	28,800,000	9.12	1.30	1.32
17000	42.8	28.5	39.4	47.1	34.5	49.4	17000	40.8	25.9	31.8	44.5	30.8	39.0	17000	37,360,000	13.50	2.85	3.28	17500	34,500,000	10.09	1.89	1.78
17500	45.4	32.0	43.7	49.9	35.7	54.8	17500	45.9	32.7	39.5	50.0	38.9	48.5	17500	39,300,000	14.20	3.16	3.61	18000	30,320,000	12.75	2.46	2.27
18000	47.9	35.7	48.3	52.6	43.0	54.6	18000	51.0	40	48.0	55.5	47.9	58.9	18000	41,600,000	14.20	3.16	3.61	18500	37,360,000	12.75	2.46	2.27
18500	50.4	39.5	53.1	55.4	47.1	59.5	18500	56.1	49	57.3	61.1	58	70.3	18500	46,000,000	14.20	3.16	3.61	19000	37,360,000	12.75	2.46	2.27
19000	52.9	43	58.2	62.6	57.9	65.7	19000	61.3	58	67.2	66.7	69	82.3	19000	50,000,000	14.20	3.16	3.61	19500	37,360,000	12.75	2.46	2.27
19500	55.4	48	63.5	61.0	57.8	79.4	19500	66.4	68	78.1	72.2	81	95.7	19500	54,000,000	14.20	3.16	3.61	20000	37,360,000	12.75	2.46	2.27
20000	58.0	52	63.7	63.7	63	86.3	20000	71.5	79	89.5	77.8	94	110	20000	58,000,000	14.20	3.16	3.61	20500	37,360,000	12.75	2.46	2.27

Factor for correcting to other pipe sizes				Factor for correcting to other pipe sizes			
Dia in	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Dia in	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft




<tbl\_r cells="8" ix="4" maxcspan

TABLE 4-46—(continued)

Friction Losses in Pipe; C = 100								Friction Losses in Pipe; C = 100							
36 in. inside dia				42 in. inside dia				48 in. inside dia				54 in. inside dia			
Discharge in U.S. gallons		Veloc- ity feet per sec	Veloc- ity head in ft	Discharge in U.S. gallons		Veloc- ity feet per sec	Veloc- ity head in ft	Discharge in U.S. gallons		Veloc- ity feet per sec	Veloc- ity head in ft	Discharge in U.S. gallons		Veloc- ity feet per sec	Veloc- ity head in ft
per min	per 24 hr			per min	per 24 hr			per min	per 24 hr			per min	per 24 hr		
1400	2,016,000	.44	.00	2000	2,880,000	.46	.003	2600	2,880,000	.35	.002	3200	5,760,000	.55	.005
1700	2,448,000	.53	.00	2500	3,600,000	.58	.006	3100	5,760,000	.70	.006	3800	11,520,000	1.11	.019
2000	2,880,000	.63	.01	3300	4,320,000	.70	.007	3900	8,640,000	1.05	.017	4600	17,280,000	1.67	.043
2400	3,456,000	.75	.01	4000	5,040,000	.81	.010	4600	11,520,000	1.40	.030	5300	29,040,000	2.33	.077
2800	4,032,000	.88	.01	4800	5,760,000	.92	.013	5400	14,400,000	1.76	.048	6100	28,800,000	2.89	.123
3400	4,608,000	1.07	.02	5600	6,480,000	1.04	.016	6200	17,280,000	2.11	.069	6800	31,680,000	3.07	.146
4000	5,040,000	1.26	.02	6400	7,200,000	1.18	.021	7000	20,160,000	2.47	.095	7600	34,560,000	3.35	.174
4800	6,912,000	1.51	.04	7200	8,640,000	1.39	.030	7800	23,040,000	2.83	.134	8400	37,440,000	3.63	.208
5600	8,054,000	1.76	.05	8000	10,080,000	1.62	.041	8600	25,920,000	3.18	.166	9200	40,320,000	4.93	.238
6200	8,640,000	1.96	.06	8800	11,820,000	1.85	.053	9400	28,800,000	3.53	.198	10000	43,200,000	4.30	.274
7000	10,080,000	2.30	.07	9600	12,960,000	2.08	.067	10200	31,680,000	2.89	.235	10800	46,080,000	4.48	.311
7600	10,244,000	2.39	.09	10800	14,400,000	2.31	.083	11400	34,560,000	4.24	.279	12000	48,960,000	4.76	.383
8200	11,933,000	2.61	.10	12000	17,280,000	2.78	.120	12600	37,440,000	4.50	.329	13200	51,840,000	5.05	.396
9000	12,960,000	2.83	.12	13200	20,160,000	3.24	.163	13800	40,320,000	4.96	.388	14400	54,720,000	5.38	.440
9700	13,968,000	3.05	.14	14400	23,040,000	3.70	.212	15000	43,200,000	5.32	.440	15600	57,600,000	5.62	.490
10000	14,400,000	3.14	.16	15600	26,920,000	4.16	.269	16200	46,080,000	5.63	.500	16800	60,480,000	5.89	.539
11000	15,840,000	3.46	.19	16800	28,800,000	4.62	.331	17400	48,960,000	6.03	.565	18000	63,360,000	6.16	.590
12000	17,280,000	3.73	.22	17400	31,680,000	5.10	.404	18000	51,840,000	6.39	.632	18600	66,240,000	6.44	.642
12800	18,720,000	4.09	.26	18000	34,560,000	5.55	.477	18600	54,720,000	6.75	.708	19200	69,020,000	6.72	.701
14000	20,160,000	4.40	.30	18600	37,440,000	6.02	.561	19200	57,600,000	7.10	.782	19800	72,000,000	7.00	.760
15000	21,600,000	4.71	.34	19800	40,320,000	6.48	.651	20400	60,480,000	7.44	.859	21000	74,880,000	7.27	.820
16000	23,040,000	5.03	.39	20400	43,200,000	6.94	.748	21000	63,360,000	7.80	.943	21600	77,760,000	7.56	.889
18000	25,920,000	5.66	.50	21600	46,080,000	7.40	.850	22000	66,240,000	8.16	1.04	22600	80,640,000	7.84	.953
19000	27,360,000	5.98	.56	22000	48,960,000	7.86	.9	22600	69,020,000	8.51	1.15	23200	83,520,000	8.12	1.02
20000	28,800,000	6.30	.61	22600	51,840,000	8.33	1.07	23200	72,000,000	8.87	1.23	23800	86,400,000	8.40	1.10
21000	30,240,000	6.60	.67	23200	54,720,000	8.80	1.20	23800	74,880,000	9.22	1.32	24400	89,280,000	8.68	1.17
22000	31,680,000	6.92	.74	24400	57,600,000	9.26	1.32	24800	77,760,000	9.58	1.42	25400	92,160,000	9.56	1.25
23000	33,120,000	7.24	.81	24800	60,480,000	9.72	1.46	25400	80,640,000	9.94	1.54	26000	95,040,000	9.34	1.32
24000	34,560,000	7.55	.88	25400	63,360,000	10.18	1.61	26000	83,520,000	10.28	1.64	26600	97,920,000	9.52	1.40
24000	37,440,000	8.18	1.04	26000	66,240,000	10.63	1.76	26600	86,400,000	10.63	1.76	27200	100,960,000	9.80	1.49
25000	40,320,000	8.80	1.20	27200	69,020,000	11.10	1.92	27800	89,280,000	10.99	1.87	28400	102,680,000	10.07	1.67
26000	42,200,000	9.44	1.38	27800	72,000,000	11.58	2.08	28400	92,160,000	11.34	1.87	29000	106,560,000	10.35	1.66
27000	45,960,000	10.70	1.77	29000	74,880,000	12.01	2.25	29600	109,440,000	11.70	2.12	30200	112,320,000	10.62	1.76
28000	54,720,000	11.95	2.30	29600	77,760,000	13.49	2.41	30200	115,300,000	12.05	2.35	30800	118,200,000	10.91	1.85
29000	60,480,000	13.20	2.70	30800	80,640,000	12.93	2.60	30800	124.1	3.39	2.58	31400	115,300,000	11.19	1.94

## Factor for correcting to other pipe sizes

Dia in	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
35	1.068	1.119	1.147
34	1.121	1.267	1.221
33	1.190	1.416	1.227
32	1.264	1.602	1.774

## Factor for correcting to other pipe sizes

Dia in	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
41	1.049	1.101	1.124
40	1.102	1.216	1.268
39	1.160	1.345	1.324
38	1.232	1.492	1.427

## Factor for correcting to other pipe sizes

Dia in	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
47	1.043	1.089	1.105
46	1.089	1.186	1.238
45	1.138	1.295	1.352
44	1.190	1.416	1.427

## Factor for correcting to other pipe sizes

Dia in	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
53	1.038	1.078	1.096
52	1.078	1.163	1.202
51	1.121	1.257	1.321
50	1.166	1.360	1.422

(continued)

TABLE 4-46—(continued)

## Friction Losses in Pipe; C = 100

68 in. Inside dia				72 in. inside dia			
Discharge in U.S. gallons		Velocity feet per sec	Head loss in feet per 100 ft	Discharge in U.S. gallons		Velocity feet per sec	Head loss in feet per 100 ft
per min	per 24 hr			per min	per 24 hr		
5000	7,200,000	.56	.005	10000	14,400,000	.78	.009
12000	14,400,000	1.13	.019	20000	28,800,000	1.57	.038
18000	21,600,000	1.70	.048	30000	43,200,000	1.97	.060
25000	28,800,000	2.26	.079	40000	58,200,000	2.36	.086
35000	36,000,000	2.88	.124	50000	60,400,000	2.76	.118
40000	43,200,000	3.40	.179	40000	57,600,000	3.16	.154
45000	46,800,000	3.68	.205	50000	64,800,000	3.54	.194
50000	48,960,000	3.86	.220	60000	72,000,000	3.94	.240
55000	51,840,000	4.09	.269	70000	74,880,000	4.39	.269
58000	54,720,000	4.33	.300	80000	77,760,000	4.28	.280
60000	57,600,000	4.55	.320	60000	80,560,000	4.41	.302
62000	60,480,000	4.78	.354	70000	83,320,000	4.57	.324
65000	63,360,000	5.00	.387	80000	86,400,000	4.78	.347
68000	66,240,000	5.23	.423	90000	89,280,000	4.98	.370
70000	69,020,000	5.45	.460	100000	92,160,000	5.04	.384
72000	72,000,000	5.68	.500	60000	95,040,000	5.20	.420
75000	74,880,000	5.90	.540	70000	97,920,000	5.36	.447
78000	77,760,000	6.13	.582	80000	100,800,000	5.51	.473
80000	80,640,000	6.35	.626	90000	103,680,000	5.67	.499
82000	83,520,000	6.53	.672	100000	106,560,000	5.83	.523
84000	86,400,000	6.71	.720	70000	109,440,000	5.99	.558
86000	89,280,000	7.03	.768	80000	112,320,000	6.15	.588
88000	92,160,000	7.25	.819	90000	115,200,000	6.31	.620
90000	95,040,000	7.49	.870	100000	118,080,000	6.46	.650
92000	97,920,000	7.73	.925	80000	120,960,000	6.62	.680
70000	100,800,000	7.95	.980	40000	123,840,000	6.78	.712
72000	103,680,000	8.17	1.04	50000	126,720,000	6.93	.746
74000	106,560,000	8.40	1.10	60000	129,600,000	7.09	.780
76000	109,440,000	8.63	1.16	70000	135,580,000	7.49	.870
78000	112,320,000	8.86	1.32	80000	144,000,000	7.88	.960
80000	115,200,000	9.06	1.38	100000	151,200,000	8.28	1.06
82000	118,080,000	9.44	1.44	90000	158,400,000	8.67	1.16
84000	119,960,000	10.30	1.61	110000	165,600,000	9.05	1.27
86000	126,800,000	10.78	1.80	120000	172,800,000	9.48	1.38
88000	144,000,000	11.85	2.00	130000	180,000,000	9.83	1.51

Factor for correcting to other pipe sizes				Factor for correcting to other pipe sizes			
Dia in	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft	Dia in	Velocity ft per sec	Velocity head ft	Head loss ft per 100 ft
59	1.034	1.070	1.005	79	1.058	1.119	1.147
58	1.070	1.145	1.173	68	1.121	1.287	1.318
57	1.108	1.228	1.204	66	1.190	1.416	1.454
56	1.149	1.318	1.359	64	1.366	1.602	1.774

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## SOFTWARE FOR CALCULATING PRESSURE DROP

The following software in DOS for calculating frictional pressure losses in relationship to fluid flow for incompressible, compressible and two-phase flows are

PIPECAL	Pressure drop calculation for incompressible fluids (Imperial Units)
PIPESI	Pressure drop calculation for incompressible fluids (SI units).
COMPRES	Pressure drop calculation for compressible fluids
TWOPHASE	Pressure drop calculation for two-phase (gas-liquid) fluids.

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# 5

## PUMPING OF LIQUIDS

**P**umping of liquids is almost universal in chemical and petrochemical processes. The many different materials being processed require close attention to selection of materials of construction of the various pump parts, shaft sealing, and the hydraulics of the individual problems. A wide variety of pump types have been developed to satisfy the many special conditions found in chemical plant systems; however, since all of these cannot be discussed here, the omission of some does not mean that they may not be suitable for a service. In general, the final pump selection and performance details are recommended by the manufacturers to meet the conditions specified by the process design engineer. It is important that the designer of the process system be completely familiar with the action of each pump offered for a service in order that such items as control instruments and valves may be properly evaluated in the full knowledge of the system.

This chapter presents information on rating, sizing, and specifying process pumps. The emphasis will be on centrifugal pumps, which are by far the most widely used in the process industries; however, applications of other types of pumps will also be discussed (Table 5-1). The chapter further provides the Excel spreadsheets for rating/sizing a centrifugal pump (Figure 5-61b), and to determine pump efficiency, hydraulic brake horsepower, and actual brake horsepower for pumps at varying flow rate and developed head.

To properly accomplish a good and thorough rating/sizing of a centrifugal pump, the plant system designer should at a minimum do the following.

1. Understand the fundamentals of performance of the pump itself
2. Understand the mechanical details required for a pump to function properly in a system.
3. Calculate the friction and any other pressure losses for each "side" of the pump, suction, and discharge (see Chapter 4).
4. Determine the suction side and discharge side heads for the mechanical system connecting to the pump.
5. Determine the important available net positive suction head ( $NPSH_A$ ) for the pump suction side mechanical system, and compare this to the manufacturer's required net positive suction head ( $NPSH_R$ ) by the pump itself. This requires that the designer makes a tentative actual pump selection of one or more manufacturers in order to use actual numbers.

6. Make allowable corrections to the pump's required  $NPSH$  (using charts where applicable) and compare with the available  $NPSH$ . The available must always be several feet (mm) greater than the corrected required.
7. Make fluid viscosity corrections to the required performance if the fluid is more viscous than water.
8. Examine specific speed index, particularly if it can be anticipated that future changes in the system may be required.
9. If fluid being pumped is at elevated temperature (usually above  $\pm 90^\circ F$  ( $\pm 32.2^\circ C$ )), check temperature rise in the pump and the minimum flow required through the pump.
10. Make pump brake horsepower corrections for fluids with a specific gravity different from water. Select actual driver (electric motor, usually) horsepower in order that horsepower losses between the driver and the pump shaft will still provide sufficient power to meet the pump's input shaft requirements.
11. If the pump has some unique specialty service or requirements, recognize these in the final sizing and selection. Consult a reliable manufacturer that produces pumps for the type of service and applications and have them verify the analysis of your system's application(s).

The centrifugal pump (Table 5-2) develops its pressure by centrifugal force on the liquid passing through the pump and is generally applicable to high capacity, low to medium head installations. In order to satisfy pump discharge head (or pressure) requirements the unit may be a multistage (multiple impellers) instead of a single stage [1]. The conditions of pumping water vs. pumping hot light hydrocarbons require considerably different evaluation in pump design features for satisfactory operation, safety, and maintenance.

The inline centrifugal process pump (Figure 5-1) is relatively new to general applications; however, it is finding many applications where space and installation costs are important. Each application must be carefully evaluated, as there are three basic types of pump construction to consider. Generally, for many applications the dimensions have been standardized through the American Voluntary Standard, American National Standards Institute (ANSI), or American Petroleum Institute (API)-610. The performance curves are typical of single stage centrifugal pumps.

The turbine is a special type of centrifugal pump (Figure 5-2) and has limited special purpose applications.

**TABLE 5-1 General Types or Classification of Pumps**

Centrifugal	Rotary	Reciprocating
Centrifugal	Cam	Piston
Propeller	Screw	Plunger
Mixed Flow	Gear	Diaphragm
Peripheral	Vane	
Turbine	Lobe	
Radical Flow	Piston	
Axial Flow	Flexible Rotor	

Note: All types will not be treated in detail, but consideration of their particular features is important in many situations.

**TABLE 5-2 Approximate Capacity-Head Ranges for Centrifugal Pumps**

Type	Maximum GPM*	Maximum Head* (ft)	Figure Number†
Single Stage (H)	600	225	5-12 and 5-13
Canned, Sealless	2000	650	5-52
Canned, Sealless	1000	800	5-53
Single Stage (V)	>150	250+	5-1
Double Suction			
Single Stage (H)	15,000	300	5-54 and 5-55
Multistage (H)	3,000	5,000	5-56-5-58
Single and Multistage (V)			
Mixed Flow (V)	100,000	75	5-59
Axial Flow (V)	100,000	25	5-60
Centrifugal (V)	400±	5,750	5-61-5-63 and 5-2

\* Not necessary at same point. (H) = Horizontal (V) = Vertical

† See Appendix for Figure 5-52-5-63

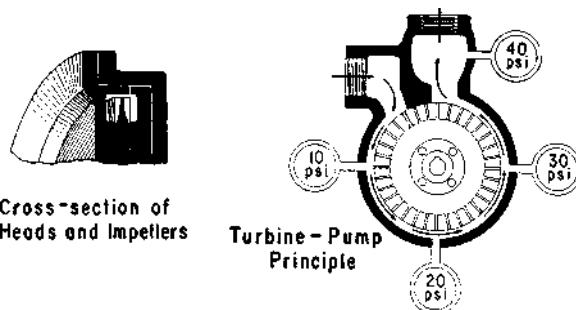


Figure 5-2 Turbine pump. (Courtesy of Roth Pump Co.)

## 5.1 PUMP DESIGN STANDARDIZATION

Certain pump designs have been standardized to aid manufacturer's problems, and to allow the owners to take advantage of standardization of parts and dimensions, and consequently maintain a more useful inventory. The standards are sponsored through ANSI; however, many manufacturers also produce to the API standards and their own proprietary standards. These are special pumps that do not conform to all the standards, but are designed to accomplish specific pumping services.

The primary pump types for the chemical industry for horizontal and vertical inline applications have been standardized in ANSI B-123, ANSI Std # B73.1M for horizontal end suction centrifugal pumps, and ANSI B73.2M for vertical inline centrifugal pumps. The standards are in a continuous process of upgrading to suit requirements of industry and the manufacturers. The API-610 standard is primarily a heavy-duty application, such as is used for the refinery and chemical industry requirements. This is the only true world pump [2] standard, although the International Organization for Standardization (ISO) is studying such an improved design [3].

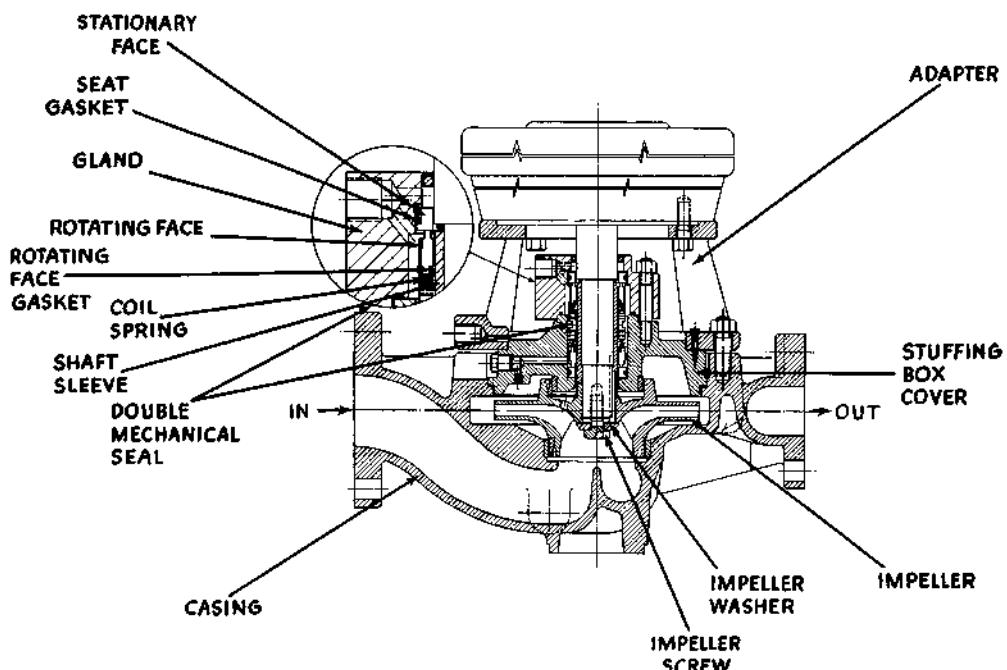


Figure 5-1 Cross-sectional view of a vertical inline pump. (By permission from Knoll, H. and S. Tinney, *Hydroc. proc.*, May 1971, p. 131 and Goulds Pump, Inc. Mechanical seal and seal venting details courtesy Borg-Warner.)

The standards are important because they allow the dimensional interchangeability of pumps and shaft packing of different manufacturers, but only as long as the manufacturers conform to the standard.

## 5.2 BASIC PARTS OF A CENTRIFUGAL PUMP

Table 5-3 is a quick reference to the function of the basic parts.

### IMPELLERS

The three common types of impellers that impart the main energy to the liquid for process applications are (Figure 5-3) as follows.

1. fully enclosed – used for high head, high pressure applications;
2. semi-enclosed – used for general purpose applications, has open vane tips at entrance to break up suspended particles and prevent clogging;
3. open – used for low heads, suspended solids applications, very small flows.

Small radial vanes are usually provided on back shroud or plate of impeller to reduce the pressure on the stuffing box, and to prevent suspended solids from entering the backside and possibly causing clogging.

The working or pumping vanes are backward in form relative to the impeller rotation. These impellers are available in nearly

any material of construction as well as as rubber, rubber-lined, glass-lined, and plastic. The lined impellers are of the open or semi-open type.

### CASING

The casing may be constructed of a wide variety of metals, as well as may be lined to correspond to the material of the impeller. Operating pressures go to about 5000 psi for the forged or cast steel barrel-type designs. However, the usual process application is in the 75–1000 psi range, the latter being in light hydrocarbon and similar high vapor pressure systems.

The removal of the casing parts is necessary for access to the impeller and often to the packing or seals. Some designs are conveniently arranged to allow dismantling the casing without removing the piping connections. There are proposed construction standards being considered which will allow easy maintenance of many of the types now being offered in a non-standard fashion.

### SHAFT

Care should be given in selecting the shaft material. It must be resistant to the corrosive action of the process fluids, yet possess good strength characteristics for design. For some designs it is preferable to use a shaft sleeve of the proper corrosion-resistant material over the preferred structural shaft material. These sleeves may be metal, ceramic, rubber, and so on, as illustrated in Figure 5-4.

**TABLE 5-3 Basic Parts of a Centrifugal Pump**

Part	Purpose
Impeller	Imparts velocity to the liquid, resulting from centrifugal force as the impeller is rotated.
Casing	Gives direction to the flow from the impeller and converts this velocity energy into pressure energy which is usually measured in feet of head.
Shaft	Transmits power from the driver to the impeller.
Stuffing box	This is a means of throttling the leakage which would otherwise occur at the point of entry of the shaft into the casing; Usually not a separate part, but rather made up of a group of small details, as "A"–"D."
(a) Packing	This is the most common means of throttling the leakage between the inside and the outside of the casing.
(b) Gland	Used to position and adjust the packing pressure.
(c) Seal gage (also called water-seal or lantern ring)	Provides passage to distribute the sealing medium uniformly around the portion of the shaft that passes through the stuffing box. This is very essential when suction lift conditions prevail to seal against in-leakage of air.
(d) Mechanical seal	Provides a mechanical sealing arrangement that takes the place of the packing. Basically it has one surface rotating with the shaft and one stationary face. The minutely close clearance between these two faces prevents leakage of liquid out or air in.
Shaft sleeve	Protects the shaft where it passes through the staffing box. Usually used in pumps with packing but often eliminated if mechanical seals are employed.
Wearing rings	Keeps internal recirculation down to a minimum. Having these rings as replaceable wearing surfaces permits renewal of clearances to keep pump efficiencies high. On small types only one ring is used in the casing and on larger sizes, composition rings are used in the casing and on the impeller.
Wearing plates	With open type impellers or end clearance wearing fits, these perform the same purpose as wearing rings do with radial clearances.
Bearings	Accurately locate shaft and carry radial and thrust loads.
Frame	Used to mount unit rigidly and support bearings. In most single suction pumps this is a separate piece. In many double suction pumps, the support is through feet cast as part of the casing. In some special suction pumps, the feet are also part of the casing and the bearing assembly is overhung. With close coupled single suction types, this support is provided by the motor or by special supporting adapters.
Coupling	Connects the pump to the driver.



Enclosed single-suction impeller with sealing on suction and back sides. (Courtesy The Deming Co.)



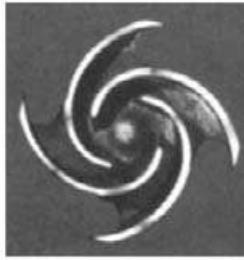
Enclosed double-suction impeller with sealing rings on both sides. (Courtesy The Deming Co.)



Mixed flow semi-enclosed impeller. (Courtesy The Deming Co.)



Semi-open or semi-enclosed impeller. (Courtesy Goulds Pumps Inc.)



Front



Back

**Figure 5-3** Impeller types. Open impeller for corrosive or abrasive slurries and solids. (Courtesy of Goulds Pumps, Inc.)

## BEARINGS

The bearings must be adequate to handle the shaft loads without excessive wear, provided lubrication is maintained. Usually this is not a point of real question provided the manufacturer has had experience in the type of loads imposed by the service conditions, and the responsibility for adequate design must be his/hers.

In all cases, the bearings should be of the outboard type, that is, not in the process fluid, unless special conditions prevail to make this situation acceptable.

## PACKING AND SEALS ON ROTATING SHAFT

Conventional soft or metallic packing in a stuffing box (Figure 5-5) is satisfactory for many low pressure, non-corrosive fluid systems. Special packings such as Teflon or mechanical seals are commonly used for corrosive fluids, since there can be leakage through the packing along the rotating shaft. However, for these conditions a mechanical seal is preferred. When the pressure becomes high (above about 50 psig (3.4 barg)) or the fluid is corrosive, additional means of sealing the shaft must be provided. Particular care must be taken in handling and using the mechanical seals, and these special instructions should be obtained from the seal manufacturer

[4]. Generally, it is not wise to have the mechanical seal installed at the pump factory, as the slightest amount of grit on the faces can cause permanent damage or destruction on only one or two revolutions at pump speed. The seals should be inspected and cleaned immediately prior to initial start-up.

A mechanical seal system (Figures 5-6a [5] and b [6]) contains a rotating element attached to the rotating shaft by set screws (or a clamp) that turns against a stationary unit set in the gland housing. The necessary continuous contact between the seal faces (see Figure 5-6a) is maintained by hydraulic pressure in the pump from the fluid being pumped and by the mechanical loading with springs or bellows. To seal the mechanical seal elements to the rotating shaft to prevent leakage along the shaft, two basic types of seals are used: (a) pusher type using springs and seal "O" rings, wedge rings, and so on; and (b) non-pusher type using some form of bellows of elastomer or metal [7] (Table 5-4).

The matching contact rubbing faces are made of dissimilar materials, precision finished to a mirror-like flat surface. There is little friction between these, and hence they form a seal that is practically fluid tight. The rubbing materials may be some combination of low friction carbon, ceramics (aluminum oxide, silicon carbide), and/or tungsten. The choice of materials will depend on the service, as will the selection of the materials of construction for the other

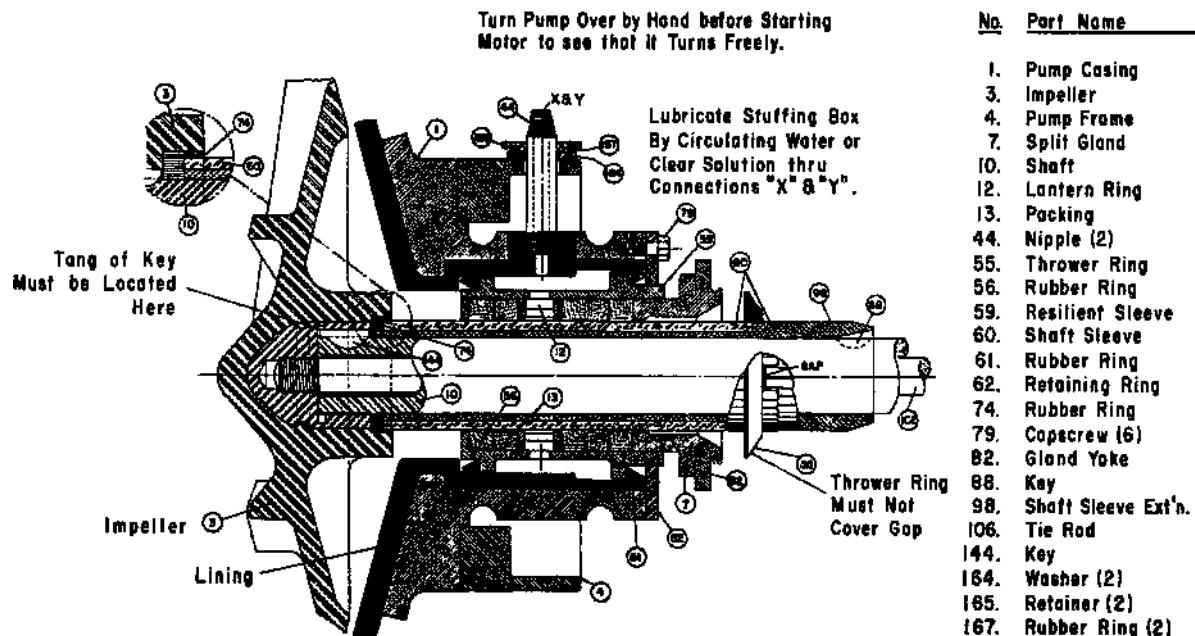


Figure 5-4 Stuffing box details lined pump with porcelain or teflon® shaft sleeve. (Courtesy of Dorr-Oliver, Inc.)

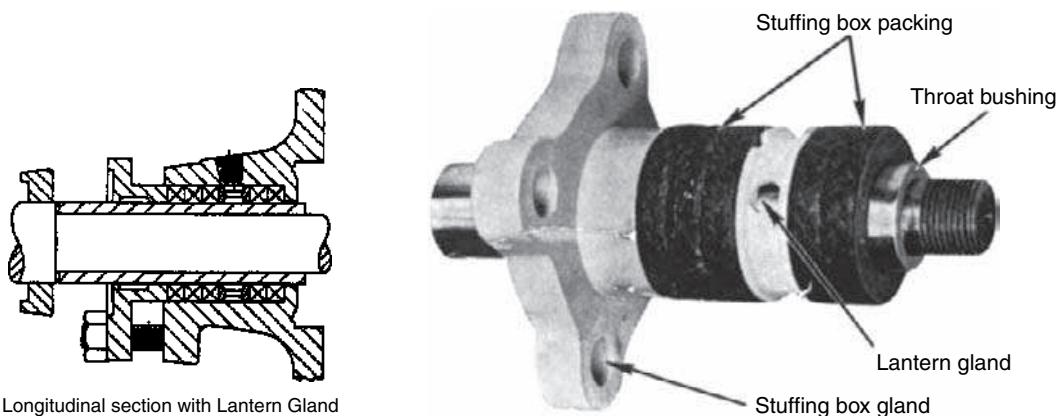


Figure 5-5 Packed stuffing box. (Courtesy of Dean Brothers Pumps, Inc.)

components, such as springs, "O" rings, other seal rings, and even the housing. The designer should consult the seal manufacturers for details of application not possible to include here.

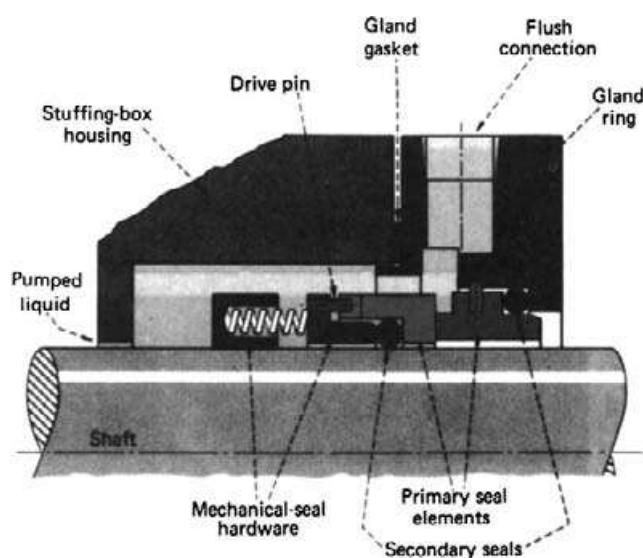
The "single" mechanical seal is made of a rotating element fixed to the shaft (or shaft sleeve), and a stationary element fixed to the pump casing [6].

The "double" seal is for severe sealing problems where out-leakage to the environment cannot be tolerated and must be controlled (Figures 5-7a and b). Depending upon the fluid's characteristics, the vent between the double seals (Figures 5-7c and d) may be purged with process liquid, or a different liquid or oil, or it may be connected to a seal pot and vent collection to prevent leakage to the air/environment. There are techniques for testing for leakage of the inner seal by measuring the vent space pressure through the seal liquid surge port. This should be essentially atmospheric (depending on the vent system back pressure). This allows detection before the leakage breaks through the outer seal.

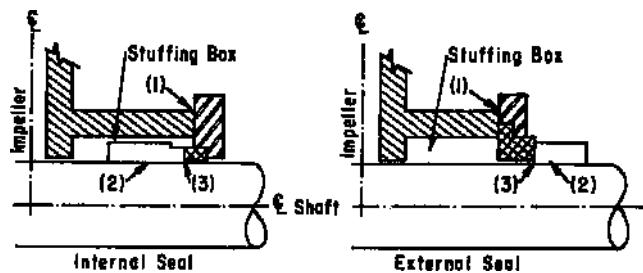
Figure 5-8 illustrates a seal installed in a conventional stuffing box with cooling liquid flow path. Figures 5-6a, 5-6b, 5-8-5-10 identify the fundamentals of mechanical seals, even though there are many specific designs and details. These various designs are attempts to correct operational problems or seal weaknesses when used under various conditions in the wide variety of process fluids.

The average unbalanced external seal is good for pressures of about 30 psig (2.1 barg), while the balanced design will handle 150 psig (10.3 barg). Special designs will handle much higher pressures. Actually the maximum operating process pressures are a function of the shaft speed and diameter for a given seal design fluid and fluid temperature.

Figure 5-11 is an outside balanced seal designed for vacuum to 150 psig and -40 to +400°F (see Table 5-4). The process fluid must be free of solids (as for practically all mechanical seals) and must not attack the material of the O-ring shaft packing. Many other designs are available, and the manufacturers should be consulted for advice on specific sealing problems.



**Figure 5-6a** Basic components of all mechanical seals. (By permission from Adams, W.H., *Chem. Eng.*, Feb 7, 1983, p. 48.)



**Figure 5-6b** The three sealing points in mechanical seals. (By permission from Sniffen, T.J., *Power and Fluids*, Winter 1958, Worthington Corp.)

### 5.3 CENTRIFUGAL PUMP SELECTION

The centrifugal pump is a versatile unit in the process plant, since its ease of control, non-pulsing flow; pressure-limiting operation fits many small and large flow systems.

Generally, the centrifugal pump has the following characteristics.

1. wide capacity, pressure, and fluid characteristics range;
2. easily adapted to direct motor, V-belt, or other drive;
3. relatively small ground area requirements;
4. relatively low cost;
5. difficult to obtain very low flows at moderate to high pressures;
6. develops turbulent conditions in fluids;
7. turbine type: (a) offers very high heads at low flows, (b) self-priming, (c) limited to very clean, non-abrasive fluids with limited physical properties, (d) clearances can be problem on assembly and maintenance.

#### SINGLE-STAGE (SINGLE IMPELLER) PUMPS

This type of pump (Figures 5-11, 5-12, and 5-13) is the workhorse of the chemical and petrochemical industries.

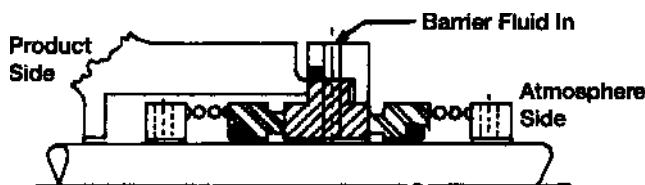
It also serves important functions in petroleum refining and almost every industry handling fluids and slurries. Although the performance characteristics may vary for specific applications, the general fundamental features are the same especially for manufacturers who standardize to some extent through the Hydraulic Institute [8] and ANSI.

Figure 5-14 indicates the relative relationship for three of the centrifugal type pumps, with curves labeled "centrifugal" referring to the usual process (open or enclosed impeller) type unit. A similar set of curves is shown in Figure 5-15 for the turbine unit. Note that the flat head curve of the centrifugal unit has advantages for many process systems, giving fairly constant head over a wide range of flow. For some systems where changes in flow must be reflected by pressure changes, the turbine characteristic is preferred. The centrifugal impeller provides an ever-rising horsepower requirement with increasing flow, while the horsepower of the turbine pump falls off with increasing flow (and decreasing head); hence it is "overloading" at low flows and must be operated with ample horsepower for these conditions.

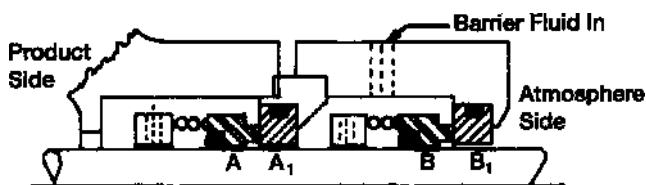
**TABLE 5-4 Requirements for Mechanical Seal Installations**

Feature	Description	Remarks	Figure Number*
Cooling	Water-jacketed stuffing box	Liquid must dead-end in stuffing box	5-64
Cooling	Gland plate	Efficient to cool contact faces	5-65
Lubrication	Dead-End	Good under vacuum, mild abrasives (metal–metal), dry seals	5-66
Lubrication	Circulating	Good cooling of contact faces	5-67
Flushing	Inside seal	Good for volatile liquids, solutions tending to crystallize, steam	5-68
Flushing	Outside seal	Heating to prevent solidification	5-69
Quenching	Outside seals (only)	For oxidizing and corrosive liquids, seal liquid washes process fluid, for high temp	5-70
Vent and drain	Inside seal	Safety feature, for venting to flare, draining	5-71
Flushing	Double	Requires circulation system	5-7a
Flushing	Tandem	Requires circulation system	5-7b
Two Rotary	Double	For improved sealing	5-7c
Four Rotary	Double tandem	For improved sealing	5-7d

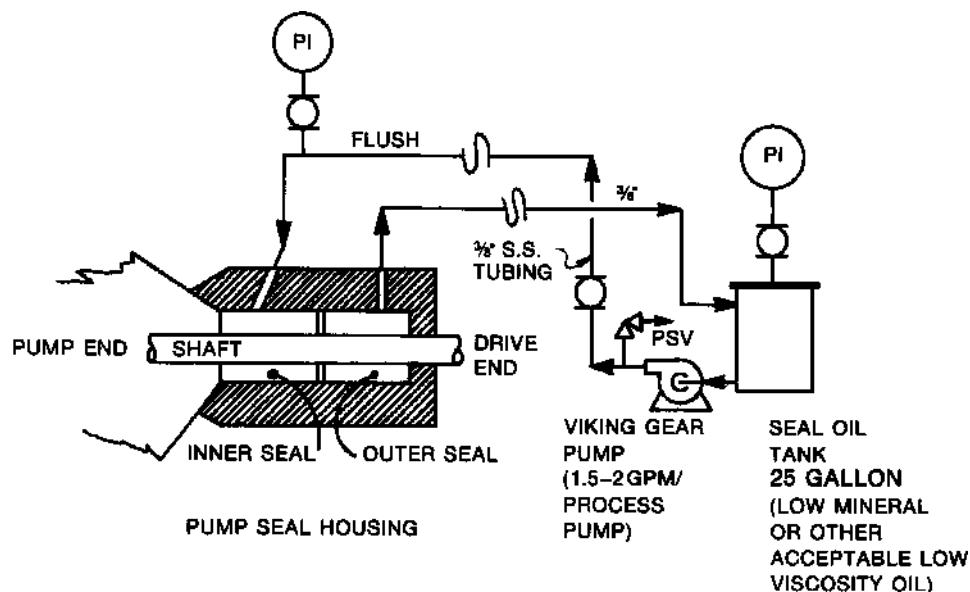
\* See Appendix for Figure 5-64–5-71



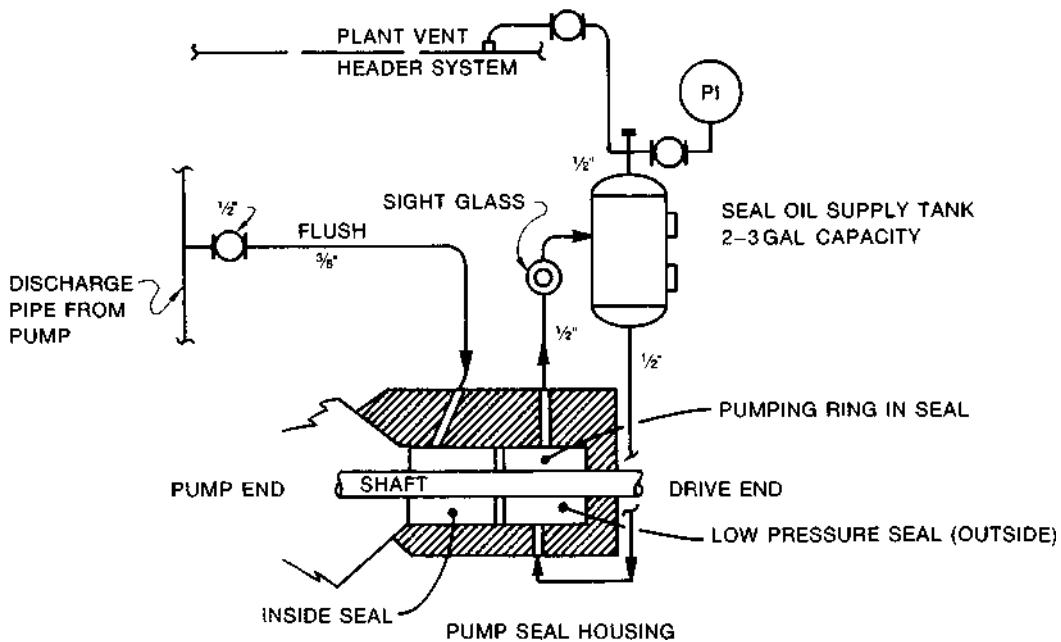
**Figure 5-7a** Double mechanical seal, two rotary elements against common stationary. (By permission from Fischer, E.E., *Chem. Process.*, Oct 1983 [7].)



**Figure 5-7b** Tandem double seal. (By permission from Fischer, E.E., *Chem. Process.*, Oct 1983 [24].)



**Figure 5-7c** Typical seal flush arrangement for double mechanical seals.



**Figure 5-7d** Typical seal flush arrangement for tandem mechanical seals.

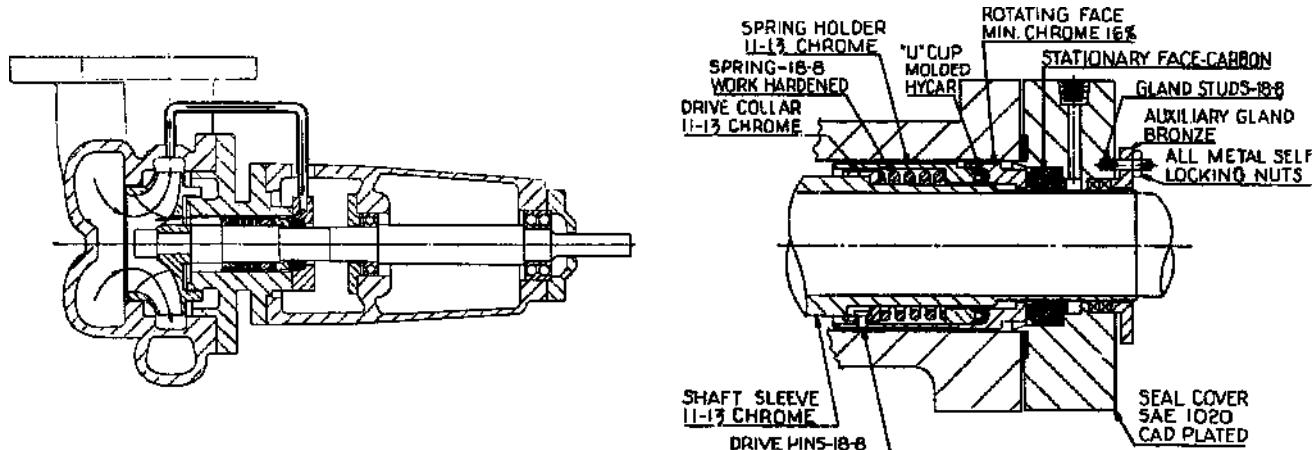


Figure 5-8 Typical single mechanical seal inside pump stuffing box. (Courtesy of Borg-Warner Co.)

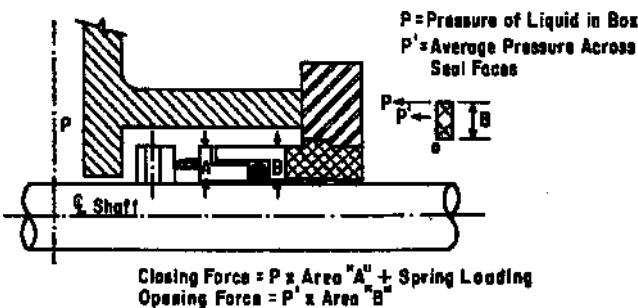


Figure 5-9 Area relationship for unbalanced seal construction. (By permission from Sniffen, T.J., *Power and Fluids*, Winter 1958, Worthington Corp.)

The effects of impeller shape for the usual centrifugal process pump performance are given in Figure 5-16. The only part the process designer can play is in the selection of a manufacturer's performance curve to fit the control requirements of the system. If the curve is too steep, select an impeller of necessary basic characteristics to move the curve in the proper direction, providing the manufacturer has an impeller pattern to fit that pump casing, and with the improved physical dimensions. This may require changing the make of pump to obtain the necessary range and characteristic.

For conditions of (1) high suction side (or inlet) friction loss, from suction piping calculations or (2) low  $NPSH_A$  (10 ft or less), a large open eye on the impeller inlet is necessary to keep the inlet velocity low. Net Positive Suction Head is discussed in a later section. The manufacturer should be given the conditions in order to properly appraise this situation.

In most instances the manufacturer has a series of impellers to use in one standard casing size. The impeller may be trimmed to proper diameter to meet head requirements and yet stay within the power range of a specified driver. It is not necessary to place a full size impeller in a casing unless the system requires this performance. It is good to know when larger impellers can be placed in the casing, and what their anticipated performance might be in order to adequately plan for future uses and changing loads on the pump.

Although the previous discussion has pertained to single impellers, the principles are the same for the multistage units (impellers in series in the casing) and the casing with double

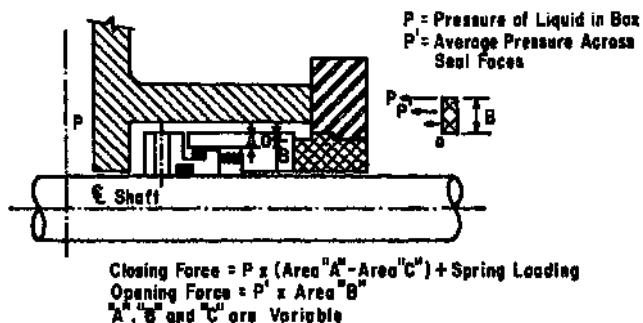


Figure 5-10 Area relationship for balanced seal construction. (By permission from Sniffen, T.J., *Power and Fluids*, Winter 1958, Worthington Corp.)

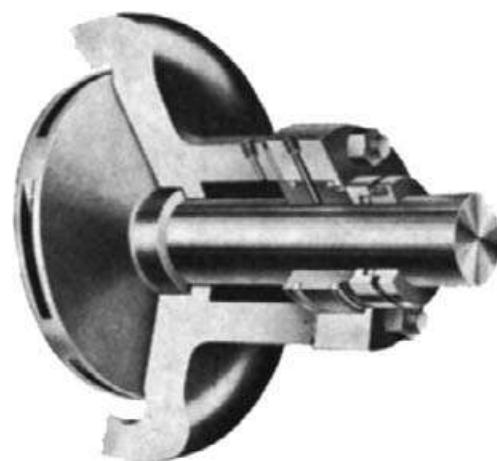
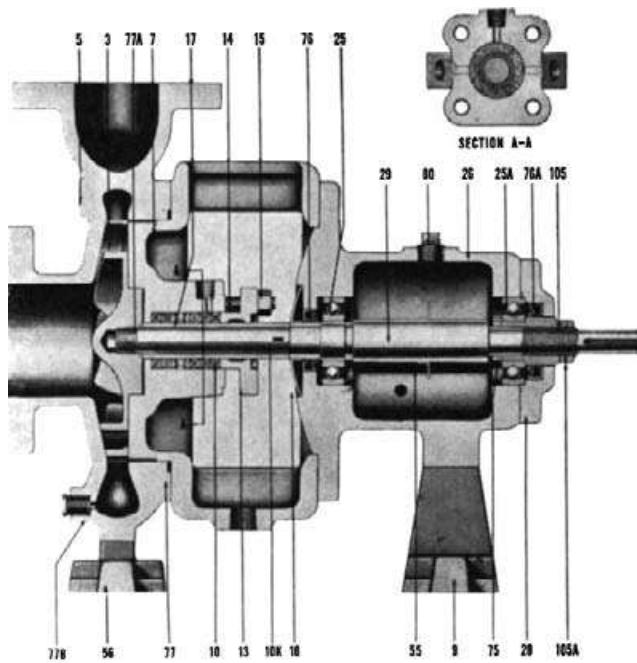


Figure 5-11 Single outside balanced seal. (Courtesy of Durametallic Corp.)

inlets. The latter pump is used for the higher flows, usually above 500 gpm, and this design serves to balance the inlet liquid load as it enters the impeller, or first stage (if more than one) from two sides instead of one as in the single impeller. The double suction



**Figure 5-12** General service centrifugal pump. (Courtesy of Dean Brothers Pumps, Inc.)

3	Impeller	26	Bearing Housing
5	Casing	28	Bearing End Cover
7	Back Head Cradle	29	Pump Shaft
9	Bearing Housing Foot	55	Oil Disc. (Flinger)
10	Shaft Sleeve	56	Casing Foot
10K	Shaft Sleeve Key	75	Retaining Ring
13	Stuffing Box Gland	76	Oil Seal - Front
14	Stuffing Box Gland Stud	76A	Oil Seal - Rear
15	Stuffing Box Gland Stud Nut	77	Gasket - Casing
17	Seal Cage	77A	Gasket - Sleeve
18	Splash Collar	77B	Gasket - Drain Plug
25	Shaft Bearing - Radial	80	Oil Vent
25A	Shaft Bearing - Thrust	105	Shaft Adjusting Sleeve
		105A	Sleeve Lock Nut

pump has the liquid passages as a part of the casing, with still only one external suction piping connection.

The axial and mixed flow impellers are used primarily for very high capacities at relatively low heads as shown in Table 5-2. They are usually applied to services such as water distribution to a large system, waste water disposal, recirculating large process liquor flows, and so on.

Many applications can be handled by either a horizontal or a vertical pump. In the range usually associated with process plants and the associated services, Tables 5-5 and 5-6 are helpful guides in making the selection [9].

#### PUMPS IN SERIES

Sometimes it is advantageous or economical to use two or more pumps in series (one pump into and through the other) to reach the desired discharge pressure. In this situation the capacity is limited by the smaller capacity of any one of the pumps (if they are different) at its speed of operation. The total discharge pressure of the last pump is the sum of the individual discharge pressures of the individual pumps. For identical pumps, the capacity is that of one pump, and the discharge pressure of the last pump is the sum

of the individual heads of each pump acting as a single unit. Thus, for two identical pumps the discharge head is twice that of the rated pressure of one pump at the designated flow rate (Figure 5-17). The pump casing of each stage (particularly the last) must be of sufficient pressure rating to withstand the developed pressure.

#### PUMPS IN PARALLEL

Pumps are operated in parallel to divide the load between two (or more) smaller pumps rather than a single large one, or to provide additional capacity in a system on short notice, or for many other related reasons. Figure 5-17 illustrates the operational curve of two identical pumps in parallel, each pump handling one half the capacity at the system head conditions. In the parallel arrangement of two or more pumps of the same or different characteristic curves, the capacities of each pump are added, at the head of the system, to obtain the delivery flow of the pump system. Each pump does not have to carry the same flow; but it will operate on its own characteristic curve, and must deliver the required head. At a common tie point on the discharge of all the pumps, the head will be the same for each pump, regardless of its flow.

The characteristic curves of each pump must be continuously rising (right to left) as shown for the single pump of Figure 5-17, otherwise with drooping or looped curves they may be two flow conditions for any one head and the pumps would "hunt" back and forth with no means to become stabilized.

Figures 5-18a-c represent typical and actual performance curves showing discharge total head (head pressure at pump outlet connection for any fluid), required minimum water horsepower (for pumping water), and capacity or pumping volume of the pump (for any fluid) for several impeller diameters that would fit the same case (housing). Additionally, the important  $NPSH_R$  characteristics of the pump's design, impeller entrance opening and diameter, and the hydraulic operating efficiency of the pump at the fixed designated speed of the performance curves are shown on the chart. All of this performance is for one specific impeller diameter of the fixed rotating speed (rpm), and the fixed impeller design pattern proprietary to the manufacturer (number, shape and spacing of vanes, and wrap or curvature of vanes).

Note that Figure 5-18b plots the  $NPSH_R$  curve for this "family" of impellers (different diameters, but exact same design dimensions and features), while Figure 5-18a shows the  $NPSH_R$  numbers printed at selected points on the curve.

Figure 5-18c illustrates the change in performance for the exact same pump, same impellers, but for different rotating speeds of 1750 and 3550 rpm. (Note that the respective motor-designed standard speeds are 1800 and 3600 rpm, but the pump manufacturer cannot count on these speeds under load in order to provide performance information the customer needs for the design of a system.)

#### 5.4 HYDRAULIC CHARACTERISTICS FOR CENTRIFUGAL PUMPS

**Capacity:** The rate of liquid or slurry flow through a pump. This is usually expressed as gallons per minute (gpm) or cubic meters per hour ( $m^3/h$ ) by pump manufacturers and design engineers in the chemical and petrochemical industries. A few convenient conversions are as follows:

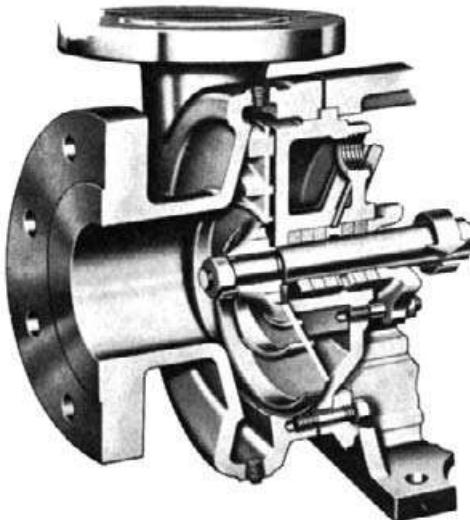
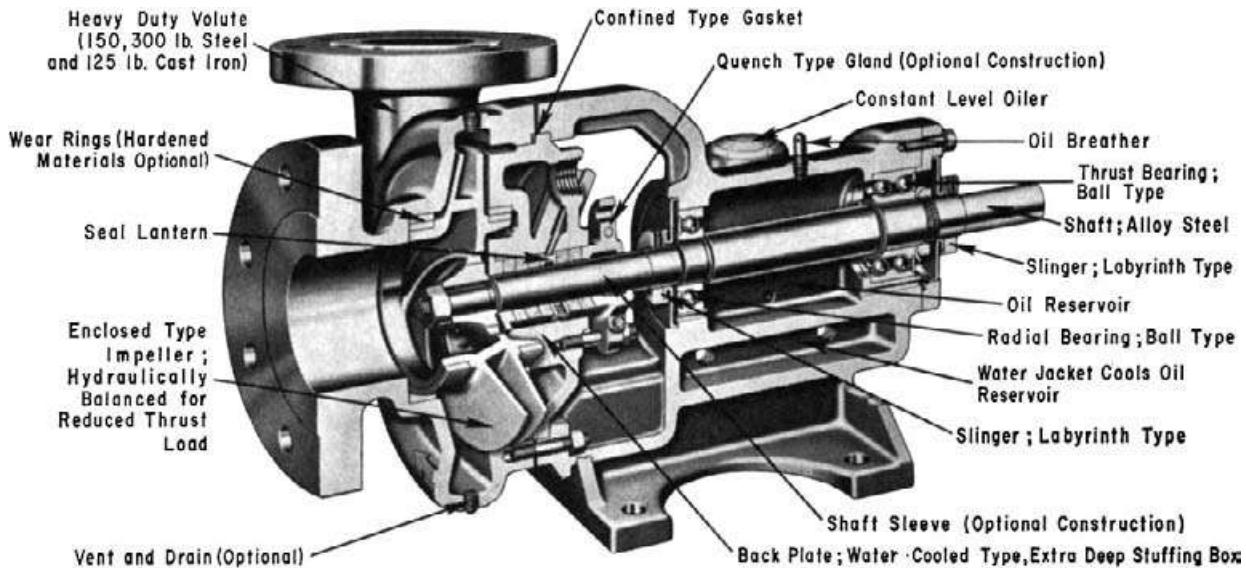


Figure 5-13 Cut-a-way section of single-stage pump. Part 1 (above) enclosed type impeller, Part 2 (lower left) open type impeller. (Courtesy of Peerless Pump Div. FMC Corp.)

1 Imperial gpm	= 1.2005 US gpm
1 barrel (42 gal)/day	= 0.0292 US gpm
1 m <sup>3</sup> /d	= 0.04167 m <sup>3</sup> /h
1 L/h	= 1 × 10 <sup>-3</sup> m <sup>3</sup> /h
1 L/s	= 3.6 m <sup>3</sup> /h

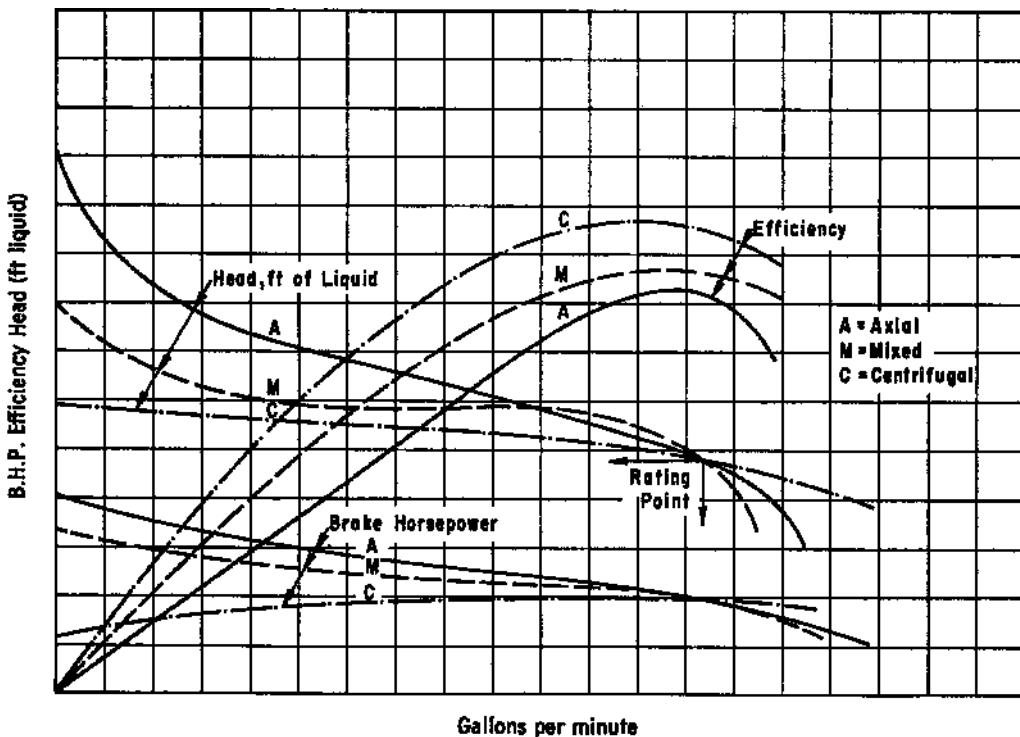
For proper selection and corresponding operation, a pump capacity must be identified with the actual pumping temperature of the liquid in order to determine the proper power requirements as well as the effects of viscosity.

Figure 5-18a illustrates typical manufacturers' performance curves for centrifugal pumps as a function of capacity.

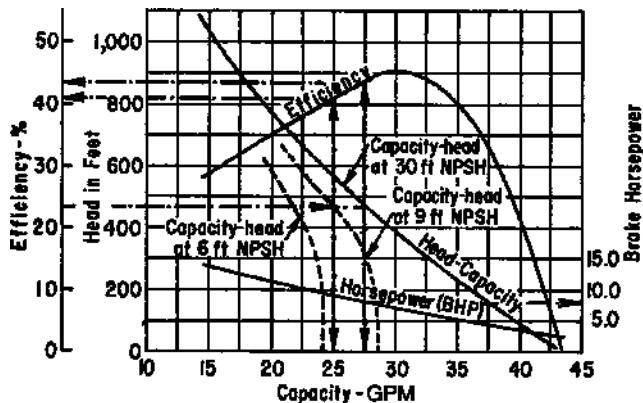
Pumps are normally selected to operate in the region of high efficiency, and particular attention should be given to avoiding the extreme right side of the characteristic curve where capacity and head may change abruptly.

**Total Head:** The pressure available at the discharge of a pump as a result of the change of mechanical input energy into kinetic and potential energy. This represents the total energy given to the liquid by the pump. Head, previously known as total dynamic head, is expressed as feet (meters) of fluid being pumped.

The *total head* read on the pump curve is the difference between the *discharge head* (the sum of the gauge reading on the discharge connection on the pump outlet, for most pumps corrected to the pump centerline, plus the velocity head at the point where the gauge is attached) and the *suction head* (the sum of the suction gauge reading corrected to the pump centerline and the velocity head at the point of attachment of the suction gauge) [10]. Note that the suction gauge reading may be positive or negative, and if negative, the discharge head minus a minus suction (termed "lift") creates an additive condition (see Section 5.5). This is shown on the curves of Figure 5-18a. This head produced is independent of



**Figure 5-14** Comparison of impeller types for centrifugal pump performance. (Adapted by permission, from Pic-a-Pump, Allis-Chalmers Mfg. Co.)



**Figure 5-15** Performance of turbine type centrifugal pump. (Courtesy of Roy E. Roth Co.)

the fluid being pumped and is, therefore, the same for any fluid through the pump at a given speed of rotation and capacity.

Through conversion, head may be expressed in units other than feet of fluid by taking the specific gravity (sp gr) of the fluid into account.

$$(Head \text{ in feet}), H = (\text{psi})(2.31 \text{ ft/psi})/\text{sp gr}, \text{ for any fluid} \quad (5-1)$$

In Metric units,

$$(Head \text{ in meters}), H = (\text{bar})(10.2 \text{ m/bar})/\text{sp gr}, \text{ for any fluid} \quad (5-1a)$$

Note that pounds per square inch (psi) is the pressure on the system and is not expressed as absolute unless the system is under absolute pressure. Feet (meters) are expressed as head, not head absolute or gauge (see Example 5-1). Note the conversion of psi pressure to feet of head pressure.

$$\text{or (Head in feet)}, H = (\text{psi})(144/\rho) \quad (5-2)$$

In Metric units,

$$(\text{Head in meters}), H = (\text{bar})(10200/\rho) \quad (5-2a)$$

where

$\rho$	= fluid density, $\text{lb}/\text{ft}^3$ ( $\text{kg}/\text{m}^3$ )
$1 \text{ lb/in.}^2$	= $2.31 \text{ ft}$ of water at sp gr = 1.0
$1 \text{ lb/in.}^2$	= $2.31 \text{ ft}$ of water/sp gr of liquid = foot liquid
1 in. mercury	= $1.34 \text{ ft}$ of water = $1.34/\text{sp gr}$ liquid
	= foot liquid
1 bar	= $10.2 \text{ m}$ of water at sp gr = 1.0

For water, sp gr = 1.0 at  $62^\circ \text{ F}$  ( $16.67^\circ \text{ C}$ ), although for general use it can be considered 1.0 over a much wider range. For explanation of vacuum and atmospheric pressure, see Chapter 4.

The three main components illustrated in the example are (adapted from [11])

1. static head
2. pressure head
3. friction in piping, entrance and exit head losses.

**EXAMPLE 5-1**  
**Liquid Heads**

If a pump were required to deliver 50 psig (3.45 barg) to a system, for water, the feet (m) of head on the pump curve must read,

$$2.31(50) = 115.5 \text{ ft}$$

In Metric units,

$$10.2(3.45) = 35.19 \text{ m}$$

For a liquid of sp gr 1.3, the feet (m) of head on the pump curve must read,  $115.5/1.3 = 88.8 \text{ ft}$  of liquid.

In Metric units, the head on the pump is  $35.19/1.3 = 27.07 \text{ m}$  of liquid.

For liquid of sp gr 0.86, the feet (m) of head on the pump curve must read,  $115.5/0.86 = 134.2 \text{ ft}$  of liquid.

In Metric units, the head on the pump is  $35.19/0.86 = 40.92 \text{ m}$  of liquid.

If a pump were initially selected to handle a liquid where sp gr = 1.3 at 88.8 ft (27.07 m), a substitution of light hydrocarbon where sp gr = 0.86 would mean that the head of liquid developed by the pump would still be 88.8 ft, but the pressure of this lighter liquid would only be  $88.8/[(2.31)/(0.86)]$  or 33.06 psi.

In Metric units, the pressure would be  $27.07/[(10.2)/(0.86)]$  or 2.3 bar.

Note that for such a change in service, the impeller seal rings, packing (or mechanical seal), and pressure rating of casing must be evaluated to ensure proper operation with a very volatile fluid. For other examples, see Figures 5-19a and b.

The total head developed by a pump is composed of the difference between the static pressure and the velocity heads plus the friction entrance and exit head losses for the suction and discharge sides of the pump (Figures 5-20 and 5-21).

$$H = h_d - h_s \quad (5-3)$$

The sign of  $h_s$  when a suction lift is concerned is negative, making  $H = h_d - (-h_s) = h_d + h_s$

A pump is acted on by the total forces, one on the suction (inlet) side, and the other on the discharge side. By subtracting (algebraically) all the suction side forces from the discharge side forces, the result is the net force that the pump must work against. However, it is extremely important to recognize the algebraic sign of the suction side components; that is, if the level of liquid to be lifted into the pump is below the pump centerline, its algebraic sign is negative (−). Likewise, if there is a negative pressure or vacuum on the liquid below the pump centerline, then this works against the pump and it becomes a negative (−) (see following discussion).

### STATIC HEAD

This is the overall height to which the liquid must be raised.

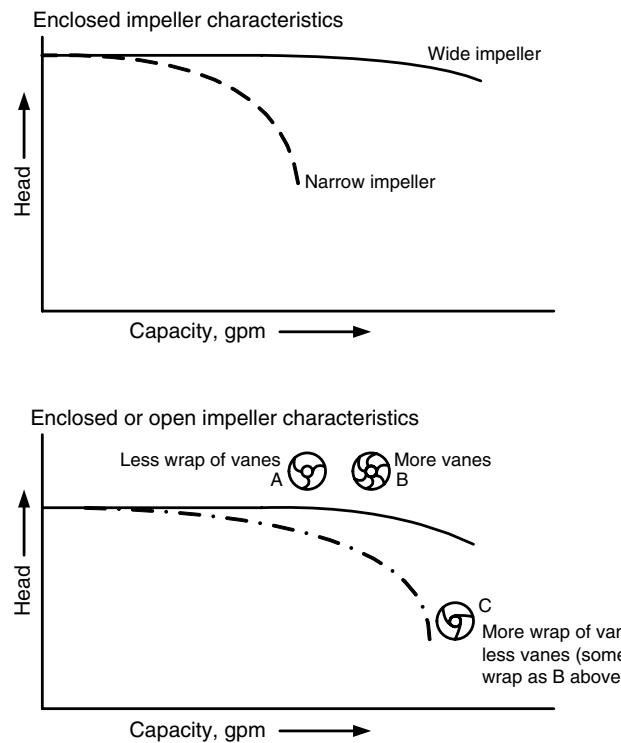
For Figure 5-22a

Discharge static head:  $H$

Suction static head:  $L$  (actually  $-L$ )

Total system static head:  $H + L$ ;

$$\text{actually } H - (-L) \quad (5-4)$$



**Figure 5-16** Impeller performance guide. Wrap refers to curvature of vanes on impeller. (Adapted by permission from Pic-a-Pump, Allis-Chalmers Mfg. Co.)

**TABLE 5-5 Pump Selection Guide**

Feature	Horizontal	Vertical
Space Requirements	Less head room	Less floor area, more head room
NPSH	Requires more*	Requires less
Priming	Require*	Usually not required
Flexibility (relative to future changes)	Less	More
Maintenance	More accessible	Major work project
Corrosion and Abrasion	No great problem	Can be considerable problem
Cost	Less	More (requires more alloy to handle corrosive fluid)

\* For some conditions

For Figure 5-22b

Discharge static head:  $H$  (from centerline of pump)

Suction static head:  $S$  (actually  $+S$ )

Total system static head:  $H - S$ ;

$$\text{or } H - (+S) \quad (5-5)$$

**TABLE 5-6 Type Selection Based on Liquid Handled**

Liquid	Basic Pump Type	Type Impellers
Water and other clear non-corrosive liquids at cold or moderate temperatures.	Single or double suction.	Closed except for very small capacities.
Water above 250° F	Single or double suction. This is usually boiler feed service at high pressures requiring multistage pumps.	Closed except for very small capacities
Hydrocarbons, hot	Single suction, often of the special type called refinery pumps, designed particularly for high temperature service.	Closed with large inlets.
Corrosives: Mild acid or alkaline Strongly acid or alkaline	Single or double suction Single or double suction with single suction probably less expensive if available for the rating.	Closed except for very small capacities or where liquid tends to form scale on surfaces of moving parts.
Hot corrosives	Single suction, with many refinery pump types also used here because of high temperatures and corresponding suction pressures.	
Water with solids in suspension:	Single suction with end clearance wearing fits. If all particles pass through 1/8" mesh screen, rubber-lined pumps are available which will give many times the life of metal pumps, providing no chemical action or excessive temperature will deteriorate the rubber. Special rubber compounds can be applied to improve resistance to certain chemicals.	Open, which allows better application of the rubber, except in larger sizes. Also made in closed type.
Coarse abrasives	Single suction. Not available for full range of ratings, that is, small capacities not too easily obtained. Often have very large impellers operated at slow speeds for use when solids larger than 1" diameter are the standard diet. This would be of the type called dredge pumps handling sizeable rocks.	Closed
Pulpy solids such as paper stock	Single suction. Double suction only used on very slight solids concentrations and then with special end clearance wearing fits.	Closed. Open type used to be standard but change to end clearance wearing fits made closed impellers better suited.

**PRESSURE HEAD***For Figure 5-22c*

$$\begin{aligned}
 \text{Discharge pressure head} &= 100 \text{ psig} \\
 \text{Suction pressure head} &= 0 \text{ psig} \\
 \text{Total pressure head} &= 100 - (+0) = 100 \text{ psig} \\
 &= 100 (2.31 \text{ ft/psi/sp gr}_{\text{H}_2\text{O}=1})^* \\
 &= 231 \text{ ft of water}
 \end{aligned}$$

*Note:* The totals are differentials and neither gauge nor absolute values.

\* Applies to water only. For the other fluids use appropriate specific gravity conversion.

*For Figure 5-22d*

$$\begin{aligned}
 \text{Discharge pressure head} &= 100 \text{ psig} \\
 \text{Suction pressure head} &= +50 \text{ psig (64.7 psia)} \\
 \text{Total pressure head} &= 100 - (+50) = 50 \text{ psi} \\
 &\quad \text{not gauge or absolute} \\
 &= 50 (2.31 \text{ ft/psi/sp gr}_{\text{H}_2\text{O}=1}) \\
 &= 115.5 \text{ ft water}
 \end{aligned}$$

Note that both the discharge and the suction pressures must be on the same base/units. These illustrations are for static head only,

while overall the pump has to work against the static and the pressure heads (discussed in Section 5.4).

*For Figure 5-22e*

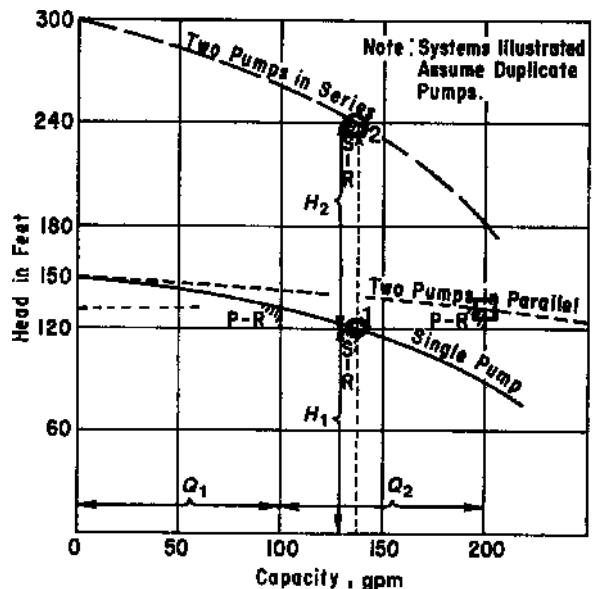
$$\begin{aligned}
 \text{Discharge pressure head} &= 100 \text{ psig} \\
 &= 100 (2.31 \text{ ft/psi/sp gr}_{\text{H}_2\text{O}=1}) \\
 &= 231 \text{ ft water (system fluid)} \\
 \text{Discharge static head} &= 50 \text{ ft} \\
 \text{Total discharge head} &= 231 + 50 = 281 \text{ ft}
 \end{aligned}$$

(\* Note that no flow friction losses or entrance/exit losses are included in this example.)

$$\begin{aligned}
 \text{Suction pressure head} &= +50 \text{ psig} \\
 &= 50 (2.31 \text{ ft/psi/sp gr}_{\text{H}_2\text{O}=1}) \\
 &= +115.5 \text{ ft water (system fluid)} \\
 \text{Suction static head} &= -10 \text{ ft} \\
 * \text{ Total suction head} &= +115.5 + (-10) = +105.5 \text{ ft} \\
 * \text{ Total head on pump} &= 281 - 105.5 = 175.5 \text{ ft fluid}
 \end{aligned}$$

**FRICITION LOSSES DUE TO FLOW**

**Friction, Entrance and Exit Heads, Valve Losses.** These losses and calculation methods were presented in Chapter 4.



**Pump in Series:**

- $Q = \text{Constant}$
- $H(\text{Total}) = H_1 + H_2 + \dots$
- $\textcircled{2} = \text{S-R denotes Series-Rating Point, Total}$

**Pumps in Parallel:**

- $= \text{Constant}$
- $Q(\text{Total}) = Q_1 + Q_2 + \dots$  (at  $H$  for each Single Pump Curve)
- $\square = \text{P-R denotes Parallel-Rating Point}$
- $\textcircled{1} = \text{Single pump rating}$

Figure 5-17 Operation curves of two duplicate centrifugal pumps in series and parallel.

Comments here will be limited. These losses are a function of the characteristics of the fluid flowing in the piping systems and the velocities of flow. Entrance and exit losses relate to the pipe and not the suction or discharge connections at the pump. Usually they

are very small, but cannot be ignored without checking. Velocity heads at the pump connections are considered internal losses. These are handled by the manufacturer's design of the pump and are not considered with the external losses in establishing the pump heads.

### 5.5 SUCTION HEAD OR SUCTION LIFT, $h_s$

The total suction head, Figure 5-23, is the difference in elevation between the liquid on the pump suction side and the centerline of the pump (plus the velocity head). Note that the suction head is positive when above the pump centerline and that it decreased with an increase in friction losses through the suction piping system. Thus,

$$\text{Total suction head (TSH)} = \text{static head} - h_{SL} \quad (5-6)$$

The total suction lift is defined as above except the level of the liquid is below the centerline of the pump or the head is below atmospheric pressure. Its sign is negative.

$$\text{Total suction lift (TSL)} = \text{static lift plus friction head losses}$$

In summary:

1. The pressure units (gauge or absolute) must be consistent for all components used in determining both suction side and discharge side conditions. Most designers use gauge as a reference, but this is not necessary.
2. Static head is positive pressure of fluid on pump suction above its centerline ( $S$ ) (+).
3. Positive external pressure  $P$  on the surface of fluid on pump suction is used as a positive integer, expressed as feet (m) of fluid (+).
4. Partial vacuum  $P$  on the surface of liquid is a negative pressure. As a partial vacuum expressed as a gauge reading as feet (mm) of liquid below atmospheric, the pressure is negative and would be designated by a minus (-) sign. A partial vacuum  $P$  expressed as absolute vacuum or absolute pressure would be

#### EXAMPLE 5-2

#### Illustrating Static, Pressure, and Friction Effects

Refer Figure 5-22f for the basis of this example.

To aid in speed of computation, the friction figures are taken from the Cameron Hydraulic tables in Chapter 4, and water, which is suited to these tables, is used as an example fluid.

$$\begin{aligned} \text{Discharge head} &= 60 \text{ ft} \\ \text{Discharge pressure head} &= 26 \text{ psig} \\ &= 26 (2.31 \text{ ft/psi/sp gr}) = 60 \text{ ft gauge} \end{aligned}$$

Discharge friction and exit head (at pipe/tank):

$$\begin{aligned} 140 \text{ ft of 8-in. pipe: } 6.32 \text{ ft/100(140)} &= 8.8 \text{ ft} \\ 3.8\text{-in. } 90^\circ \text{ ell: } (6.32/100)(3) (20.2) &= 3.8 \\ 1.8\text{-in. gate valve} &= 0.3 \\ 1.8\text{-in. check valve} &= 3.3 \\ * \text{Exit loss: Assume 8-in. pipe} &= 1.4 \\ \text{Velocity head} & \end{aligned}$$

$$\begin{aligned} \text{Subtotal, ft} &= 17.6 \\ \text{Total discharge head} &= 137.6 \text{ ft} \\ \text{Suction static head (lift)} &= -10.0 \text{ ft} \\ \text{Suction pressure head 0, psig (atm)} &= 0.0 \end{aligned}$$

Suction friction and entrance head:

$$\begin{aligned} 10 \text{ ft of 10-in. pipe } (2.1 \text{ ft/100})(10) &= 0.2 \\ 1.10\text{-in. suction } 90^\circ \text{ ell: } (2.1/100)(25.3) &= 0.5 \\ * \text{Entrance loss: Assume 10-in. pipe} & \\ \text{Velocity head} &= 0.6 \\ \text{Subtotal} &= -1.3 \\ \text{Total suction head} &= -10 + (-1.3) = -11.3 \text{ ft} \\ \text{Total pump head} &= 137.6 - (-11.3) = 148.9 \text{ ft.} \end{aligned}$$

\* These are not velocity heads at pump connections, but are related to the piping connections. See earlier note in this regard.

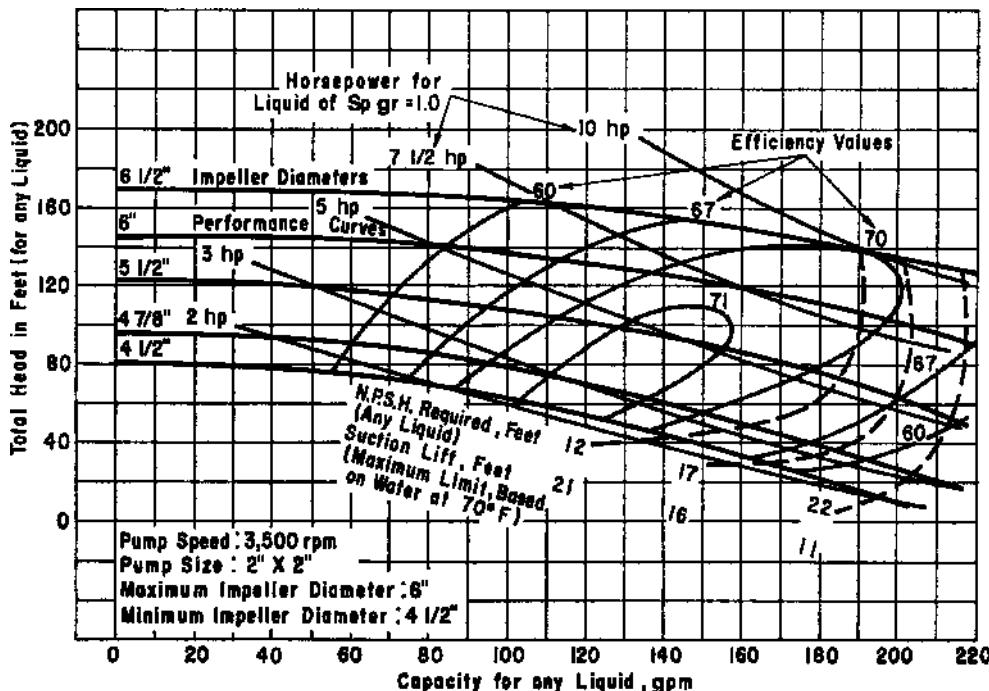


Figure 5-18a Typical centrifugal pump curves. (Adapted by permission from Allis-Chalmers, Mfg. Co.)

designated by a positive (+) sign. It is essential to be consistent for all pressure units. If absolute units are used, the total suction head would be in absolute units and the discharge head must be calculated in absolute units.

5. Suction lift is a negative suction head  $S$  used to designate a negative static condition on the suction of the pump (below atmospheric). The sign for suction head is positive (+), while its corresponding terminology of suction lift is negative (-), since the term "lift" denotes a negative condition. Note that the only difference in these terms is the difference in signs.

This applies because the total head for a pump is total discharge head  $a(+)$ , minus  $(-)$  the suction head  $[a(+)]$  or suction lift  $[a(-)]$ .

For general service the average centrifugal pump should lift about 15 ft (5 m) of water on its suction side. However, since each process situation is different, it is not sufficient to assume that a particular pump will perform the needed suction lift. Actually, certain styles or models of a manufacturer's pumps are often specially adapted to high lift conditions. On the other hand it is unnecessary to select a high lift pump when pressure head or flooded suction conditions prevail. Proper evaluation of suction lift conditions cannot be overemphasized.

The theoretical maximum suction lift at sea level for water ( $14.7 \text{ psi}$ ) ( $2.31 \text{ ft/psi/sp gr}$ ) =  $34 \text{ ft}$ . However, due to flow resistance, this value is never attainable. For safety, 15 ft (5 m) is considered the practical limit, although some pumps will lift somewhat higher columns of water. When sealing a vacuum condition above a pump, or the pump pumps from a vessel, a seal allowance to atmosphere is almost always taken as 34 ft of water. High suction lift causes a reduction in pump capacity, noisy operation due to release of air and vapor bubbles, vibration and erosion, or pitting (cavitation) of the impeller and some parts of the casing. (The extent of the damage depends on the materials of construction.)

## 5.6 DISCHARGE HEAD, $h_d$

The discharge head of a pump is the head measured at the discharge nozzle (gauge or absolute), and is composed of the same basic factors previously summarized:

1. static head
2. friction losses through pipe, fittings, contractions, expansions, entrances, and exits
3. terminal system pressure.

Some typical discharge systems are given in Figure 5-24. General practice is to express the terminal discharge pressure  $P$  in a vessel as in Figure 5-24 in terms of gauge pressure, and hence  $P = 0$  for atmospheric discharge. If  $P$  is less than atmospheric or otherwise expressed in absolute units, then it must be added as equivalent feet (m) of liquid to the value of  $h_d$  ordinarily expressed as a gauge reading.

Figures 5-20 and 5-21 illustrate the use of siphon action in pump systems. Theoretically, the head in the siphon should be recoverable, but actually it may not, at least not equivalent foot for foot. Usually not more than 20 ft (60 m) of siphon action can be included [12] even though 34 ft (10 m) are theoretical at sea level. The siphon length is  $D'$  in the figures [13]. For some systems, the discharge head on the pump should be used as  $(D + D')$ , neglecting the siphon action. In any case, if air can be trapped in the loop (and it usually can), it must be vented during start-up, otherwise the pump will be pumping against the head established using  $(D + D')$ . On start-up the flow can be gradually increased, making more head available from the pump to overcome the higher starting head of the system. This should not be overlooked nor underestimated in determining the specifications for the pump.

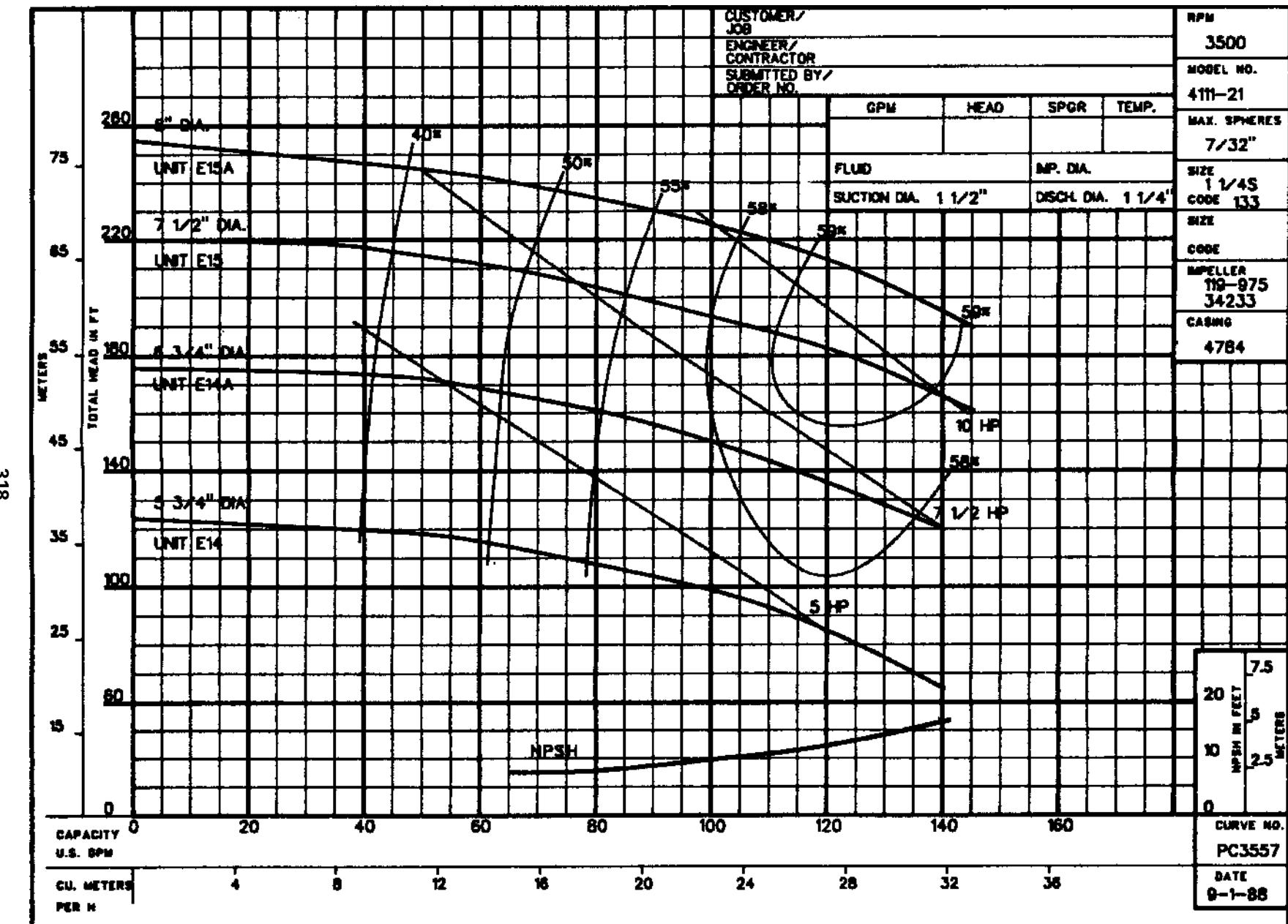


Figure 5-18b Typical performance curves showing  $NPSH_R$  in convenient form. (By permission from Crane Co., Deming Pump Div.)

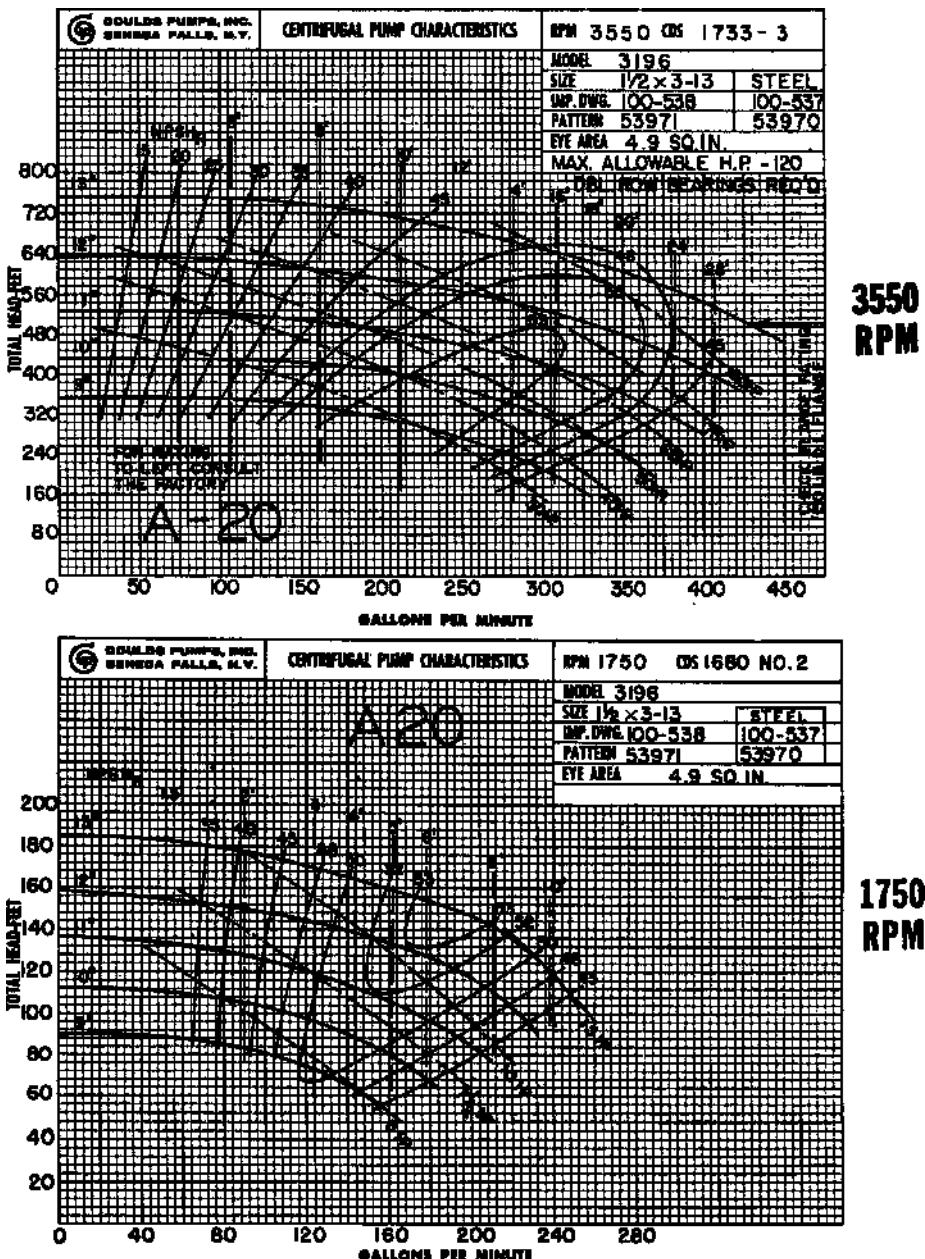


Figure 5-18c Exact same pump casing and impellers at different shaft speeds. (By permission from Goulds Pumps, Inc.)

## 5.7 VELOCITY HEAD

Velocity head is the kinetic energy of a liquid as a result of its motion at some velocity,  $V$ . It is the equivalent head in feet (m) through which water would have to fall to acquire the same velocity.

$$h_v = v^2/2g, \text{ ft (m) of fluid} \quad (5-7)$$

where

$h_v$  = velocity head, ft (m)

$v$  = liquid velocity, ft/s (m/s)

$g$  = acceleration of gravity, ft/s<sup>2</sup> (m/s<sup>2</sup>)

As a component of both suction and discharge heads, velocity head is determined at the pump suction or discharge flanges respectively, and added to the gauge reading. The actual pressure head at any point is the sum of the gauge reading plus the velocity head, the latter not being read on the gauge since it is a kinetic energy function as contrasted to the measured potential energy. The values are usually (but not always) negligible. Present practice is that these velocity head effects at the pump suction and discharge connections are to be included in the pump performance curve and pump design, and need not be actually added to the heads calculated external to the pump itself [11].

It is important to verify the effects of velocity head on the suction and discharge calculations for pump selection. In general, velocity head (kinetic energy) is smaller for high head pumps than

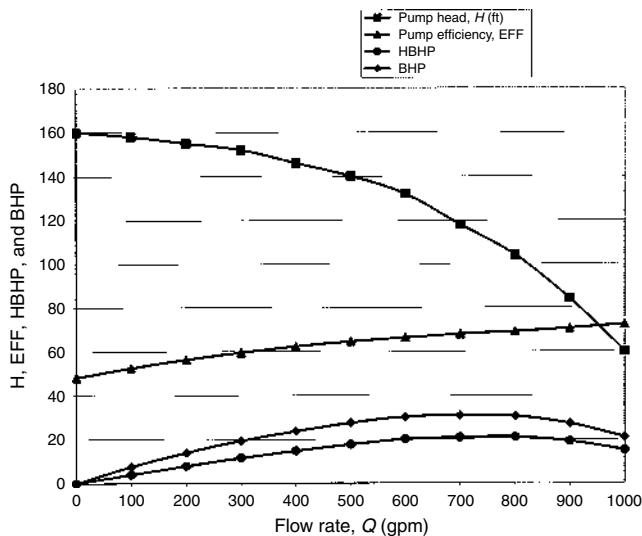
**320 PUMPING OF LIQUIDS**

 Job Item Name.		<b>PUMP CALCULATION SHEET</b>		Document No.			
				Sheet	of	Rev.	
				Item No. (s)			
		No. Working		Total No. off			
1		UNITS	CASE I	CASE II	SKETCH OF PUMP HOOK-UP		
2	Liquid Pumped			Gas-oil			
3	Corrosion/Erosion						
4	Due To						
5	Operating Temp. ( $T$ )	$^{\circ}\text{C}$	$^{\circ}\text{F}$	555			
6	Specific Gravity at $T$			1.04	1.04		
7	Viscosity	cP	cP	0.6	0.6		
8	Vapor Pressure at $T$	bara	psia	4	4		
9	Normal mass Flow rate	kg/h	lb/h	156000	171600		
10	Normal Vol. Flow rate	$\text{m}^3/\text{h}$	gpm	250			
11	Min. Vol. Flow rate						
12	Design Vol. Flow rate	$\text{m}^3/\text{h}$	gpm		275		
13	<b>SUCTION CONDITION</b>						
14	Pressure at Equipment (+)	barg	psig	13	13		
15	Static Head (+/-)	bar	psi	5.403	5.403		
16	Total – Lines 14 + 15	bar	psi	18.403	18.403		
17	Suction Line (-) $\Delta P$	bar	psi	0.22	0.22		
18	Filter/Strainer (-) $\Delta P$	bar	psi	0	0		
19							
20	Total Suction Pressure (+)	barg	psig	18.183	18.183		
21	<b>DISCHARGE CONDITION</b>						
22	Pressure at Equipment (+)	barg	psig	13.5	13.5		
23	Static Head (+/-)	bar	psi	20.94	20.94		
24							
25	Exchanger (+) $\Delta P$	bar	psi	5.2	5.2		
26							
27	Furnace (+) $\Delta P$	bar	psi	0	0		
28	Orifice (+) $\Delta P$	bar	psi	1.52	1.52		
29	Control Valve (+) $\Delta P$	bar	psi	10.71	10.71		
30							
31	Line loss (+) $\Delta P$	bar	psi	6.47	6.47		
32							
33	Total Discharge Press (+)	barg	psig	58.34	58.34		
34	Differential Pressure	bar	psi	40.157	40.157		
35	Differential Head	m	ft	89.19	89.19		
36	<b>NPSH</b>						
37	Total Suction Pressure	bara	psia	32.883	32.883		
38	Vapor Pressure	bara	psia	4	4		
39	NPSH – Lines 37–38	bara	psia	28.883	28.883		
40	=	m	ft	64.15	64.15		
41	Safety Margin	m	ft				
42	NPSH – Lines 40–41	m	ft				
43	Hydraulic Power	kW	Hp	7.04	7.74		
44	Estimated Efficiency	%	%	70	70		
45	Estimated Abs Power	kW	Hp	10.06	11.06		
46	Type of Pump						
47	Drive						
48							
49	Material – Casing						
50	– Impeller						
51	– Shaft						
52							
53	Sour Service	Yes/No					
54	HEAD	$m = 10.2 \times \text{bar}/\text{sp gr}$	$m = 10 \times \text{kg}/\text{cm}^2/\text{sp gr}$	$ft = 2.31 \times \text{psi}/\text{sp gr}$			
55	VOLUME	$\text{m}^3/\text{h} \times \text{sp gr} \times 1000 = \text{kg}/\text{h}$	$\text{igpm} \times \text{sp gr} \times 600 = \text{lb}/\text{h}$				
56	POWER	$\text{kW} = \text{m}^3/\text{h} \times \text{bar}/36.0$	$\text{kW} = \text{m}^3/\text{h} \times \text{kg}/\text{cm}^2 / 36.71$		$\text{Hp} = \text{igpm} \times \text{psi}/1427$		
57		1	Date	2	Date	3	Date
58	Description					4	Date
59	Made/Revised by					5	Date
60	Checked by						
61	Approved Process						
62	Approved						

**Figure 5-18d** Pump sizing calculation (Imperial units) for gas-oil centrifugal pump of Example 5-18.

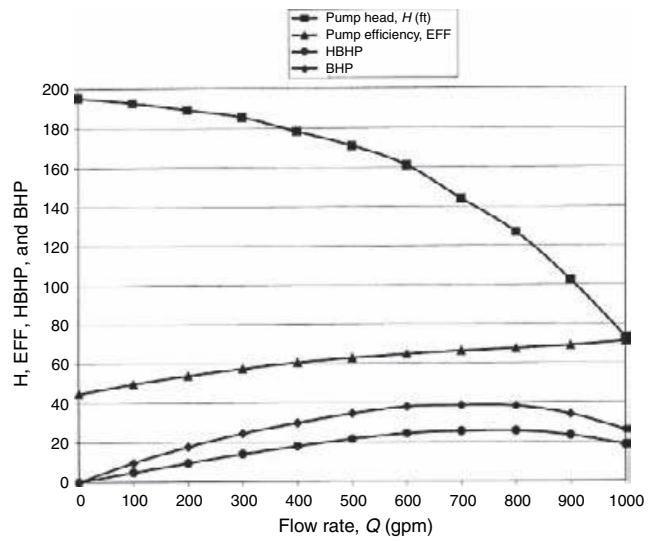
		PUMP CALCULATION SHEET		Document No.	
Job	Item Name.			Sheet	of Rev.
				Item No. (s)	
				No. Working	Total No. off
		UNITS	CASE I	CASE II	SKETCH OF PUMP HOOK-UP
1					
2	Liquid Pumped			liquid butane	
3	Corrosion/Erosion				
4	Due To				
5	Operating Temp. ( $T$ )	$^{\circ}\text{C}$	$^{\circ}\text{F}$	67	
6	Specific Gravity at $T$	cP	cP	0.488	0.488
7	Viscosity	bar	psi	0.112	0.112
8	Vapor Pressure at $T$	bara	psia	14.17	14.17
9	Normal mass Flow rate	kg/h	lb/h	52000	57201
10	Normal Vol. Flow rate	$\text{m}^3/\text{h}$	gpm	106.56	
11	Min. Vol. Flow rate				
12	Design Vol. Flow rate	$\text{m}^3/\text{h}$	gpm		117.216
13	SUCTION CONDITION				
14	Pressure at Equipment (+)	barg	psig	13.9	13.9
15	Static Head (+/-)	bar	psi	0.335	0.335
16	Total – Lines 14 + 15	bar	psi	14.235	14.235
17	Suction Line (-) $\Delta P$	bar	psi	0.0128	0.0128
18	Filter/Strainer (-) $\Delta P$	bar	psi	0	0
19					
20	Total Suction Pressure (+)	barg	psig	14.222	14.222
21	DISCHARGE CONDITION				
22	Pressure at Equipment (+)	barg	psig	13.7	13.7
23	Static Head (+/-)	bar	psi	2.057	2.057
24					
25	Exchanger (+) $\Delta P$	bar	psi	0	0
26					
27	Furnace (+) $\Delta P$	bar	psi	0	0
28	Orifice (+) $\Delta P$	bar	psi	0	0
29	Control Valve (+) $\Delta P$	bar	psi	0	0
30					
31	Line loss (+) $\Delta P$	bar	psi	0.447	0.447
32					
33	Total Discharge Pressure (+)	barg	psig	16.204	16.204
34	Differential Pressure	bar	psi	1.982	1.982
35	Differential Head	m	ft	41.43	41.43
36	NPSH				NOTES
37	Total Suction Pressure	bara	psia	15.235	15.235
38	Vapor Pressure	bara	psia	14.17	14.17
39	NPSH – Lines 37–38	bara	psia	1.065	1.065
40	=	m	ft	22.26	22.26
41	Safety Margin	m	ft		
42	NPSH – Lines 40–41	m	ft		
43	Hydraulic Power	kW	Hp	5.87	6.45
44	Estimated Efficiency	%	%	70	70
45	Estimated Abs Power	kW	Hp	8.39	9.21
46	Type of Pump	Centrifugal			
47	Drive				
48					
49	Material – Casing				
50	– Impeller				
51	– Shaft				
52					
53	Sour Service	Yes/No			
54	HEAD	$m = 10.2 \times \text{bar}/\text{sp gr}$	$m = 10 \times \text{kg}/\text{cm}^2/\text{sp gr}$	$ft = 2.31 \times \text{psi}/\text{sp gr}$	
55	VOLUME	$\text{m}^3/\text{h} \times \text{sp gr} \times 1000 = \text{kg}/\text{h}$	$\text{igpm} \times \text{sp gr} \times 600 = \text{lb}/\text{h}$		
56	POWER	$\text{kW} = \text{m}^3/\text{h} \times \text{bar}/36.0$	$\text{kW} = \text{m}^3/\text{h} \times \text{kg}/\text{cm}^2/36.71$	$\text{Hp} = \text{igpm} \times \text{psi}/1427$	
57		1 Date	2 Date	3 Date	4 Date
58	Description				
59	Made/Revised by				
60	Checked by				
61	Approved Process				
62	Approved				

Figure 5-18e Pump sizing calculation (Metric units) for reflux centrifugal pump of Example 5-19.



**Figure 5-18f** Pump efficiency calculation at varying flow rate for a 6-in. impeller centrifugal pump at 1750 rpm.)

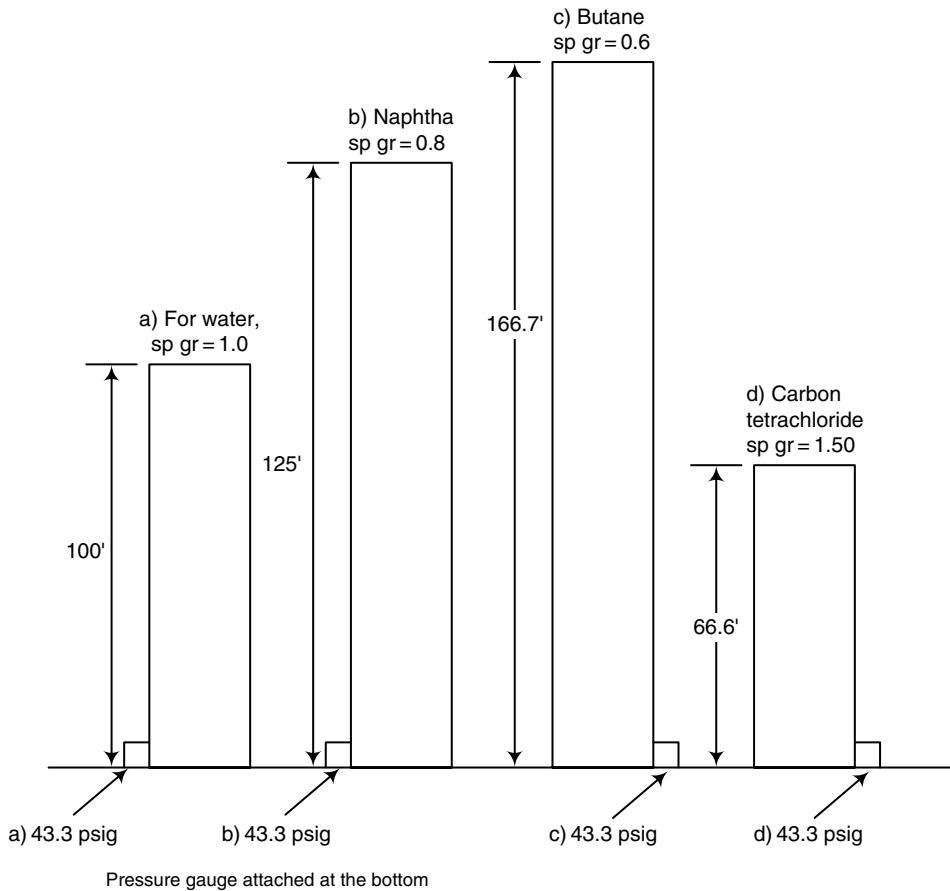
for low head units. Sometimes the accuracy of all the other system calculations does not warrant concern, but for detailed or close calculations velocity head should be recognized. The actual suction or discharge head of a pump is the sum of the gauge reading from



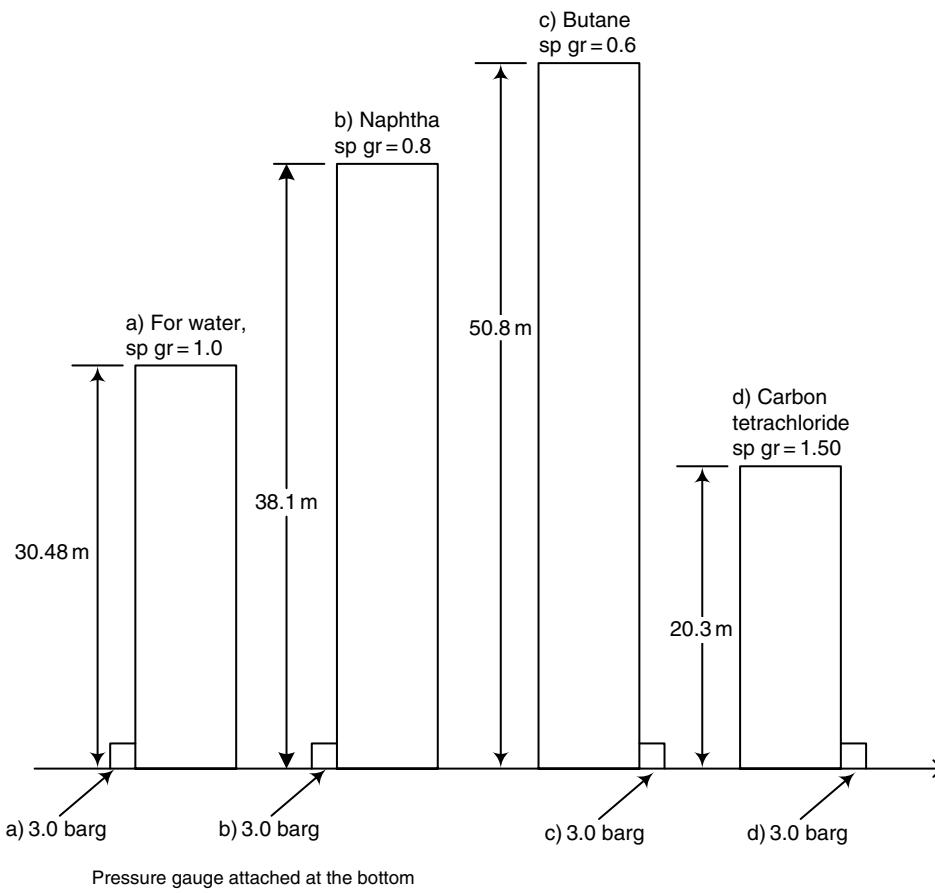
**Figure 5-18g** Pump efficiency calculation at varying flow rate for an 8-in. impeller centrifugal pump at 1450 rpm.)

a pressure gauge at the suction or discharge and the velocity heads calculated at the respective points of gauge measurement.

Regardless of their density, all liquid particles moving at the same velocity in a pipe have the same velocity head [14]. The



**Figure 5-19a** Comparison of columns of various liquids to register 43.3 psig on pressure gauge at bottom of column.



**Figure 5-19b** Comparison of columns of various liquids to register 3.0 barg on pressure gauge at bottom of column.

velocity head may vary across medium to large diameter pipes. However, the average velocity of flow (i.e., dividing the total flow as  $\text{ft}^3/\text{s}$  ( $\text{m}^3/\text{s}$ ) by the cross-sectional area of the pipe) is usually accurate enough for most design purposes.

Using the example of Karassik and Carter [10], for a pump handling 1500 gpm, having a 6-in. discharge connection and 8-in. suction connection, the discharge velocity head is 4.5 ft and the suction is 1.4 ft, calculated as shown above. If the suction gauge showed 8.6 ft, the true head would be  $8.6 + 1.4 = 10.0$  ft. If the discharge head showed 105.5 ft head, the true total head would be  $105.5 + 4.5 = 110.0$  ft, less  $(8.6 + 1.4)$  or 100 ft. The net true total head would be  $110 - 10 = 100.0$  ft. Looking only at the gauge readings, the difference would be  $105.5 - 8.6 = 96.9$  ft, giving an error of 3.1% of the total head. As an alternate example, if the discharge head were 45.5 ft, then the true total head =  $(45.5 + 4.5) - (8.6 + 1.4) = 40$  ft, and the difference in gauge readings would be  $45.5 - 8.6 = 36.9$  ft, or an error of 7.8%.

Most designers ignore the effects of velocity head, but the above brief examples emphasize that the effect varies depending on the situation and the degree of accuracy desired for the head determinations.

## 5.8 FRICTION

The friction losses for fluid flow in pipe valves and fittings are determined as presented in Chapter 4. Entrance and exit losses must be considered in these determinations, but are not to be determined for the pump entrance or discharge connections into the casing.

## 5.9 NET POSITIVE SUCTION HEAD (NPSH) AND PUMP SUCTION

Net positive suction head (in feet (m) of liquid absolute) above the vapor pressure of the liquid at the pumping temperature is the absolute pressure available at the pump suction flange, and is a very important consideration in selecting a pump which might handle liquids at or near their boiling points, or liquids of high vapor pressures.

Do not confuse NPSH with suction head, as suction head refers to pressure above atmospheric [8]. If this consideration of NPSH is ignored the pump may well be inoperative in the system, or it may be on the borderline and become troublesome or cavitating. The significance of NPSH is to ensure sufficient head of liquid at the entrance of the pump impeller to overcome the internal flow losses of the pump. This allows the pump impeller to operate with a full "bite" of liquid essentially free of flashing bubbles of vapor due to boiling action of the fluid.

The pressure at any point in the suction line must never be reduced to the vapor pressure of the liquid (see Eqs (5-10) and 5-10a). Both the suction head and the vapor pressure must be expressed in feet (m) of the liquid, and as gauge pressure or absolute pressure. Centrifugal pumps cannot pump any quantity of vapor, except possibly some vapor entrained or absorbed in the liquid. The liquid or its gases must not vaporize in the eye/entrance of the impeller. (This is the lowest pressure location in the impeller.)

For low  $NPSH_A$  (less than 10 ft or 3 m) the pump suction connection and impeller eye may be considerably oversized when

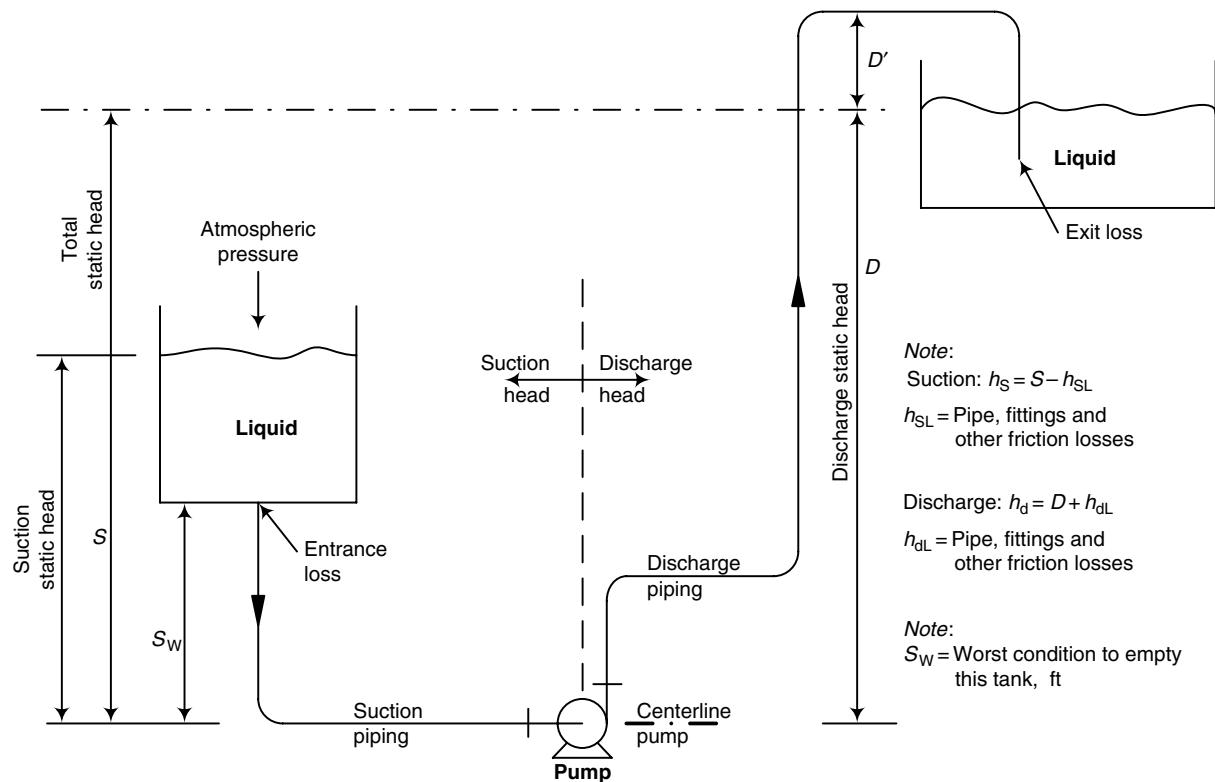


Figure 5-20 Suction head system.

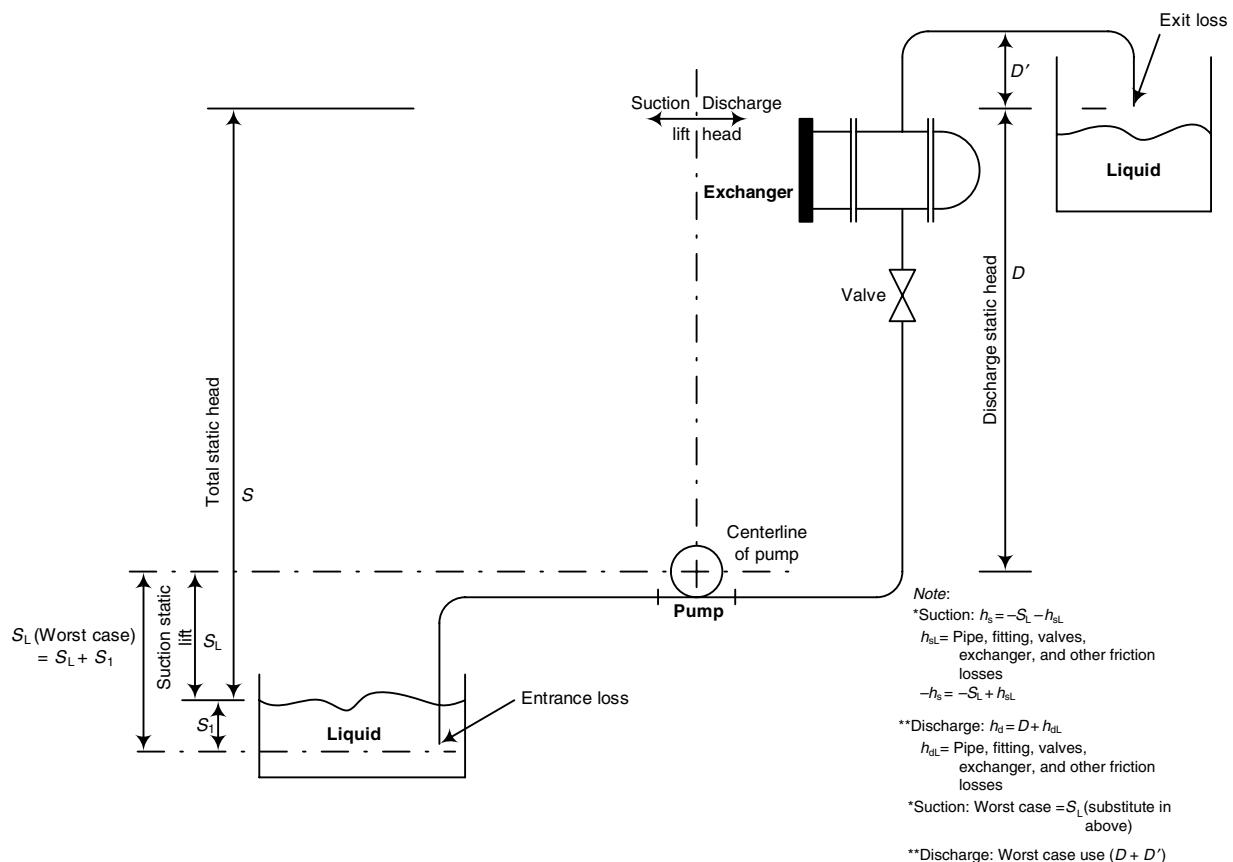
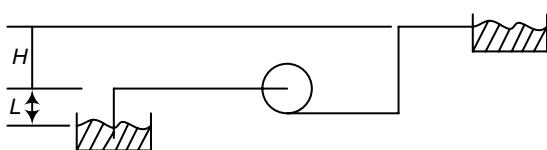
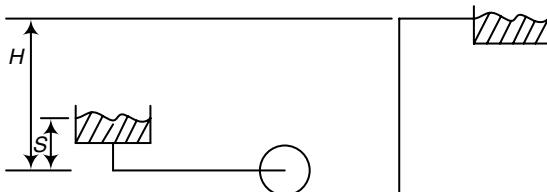


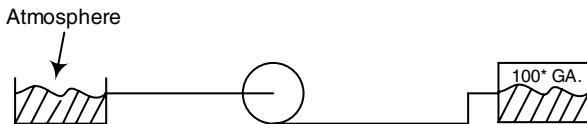
Figure 5-21 Suction lift system.



**Figure 5-22a** Static head, overall =  $H + L$ . (Adapted by permission from *Centrifugal Pumps Fundamentals*, Ingersoll-Rand Co., Washington, NJ 07882.)



**Figure 5-22b** Static head, overall =  $H - S$ . (Adapted by permission from *Centrifugal Pumps Fundamentals*, Ingersoll-Rand Co., Washington, NJ 07882.)

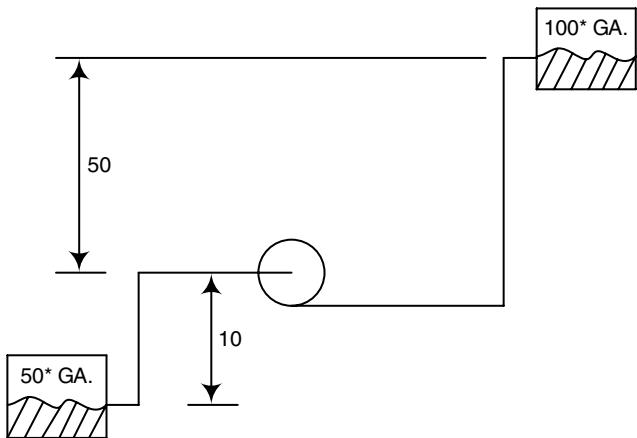


The above examples purposely disregarded pressure head, friction, entrance, and exit losses.

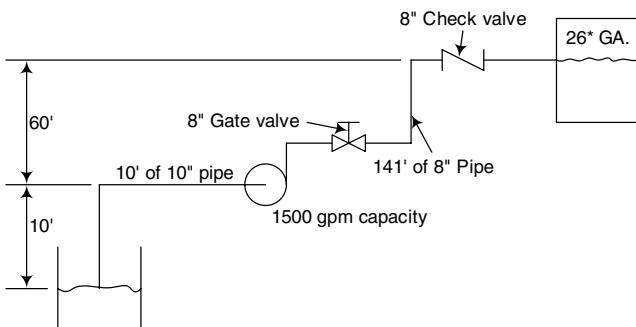
**Figure 5-22c** Pressure head. (Adapted by permission from *Centrifugal Pumps Fundamentals*, Ingersoll-Rand Co., Washington, NJ 07882.)



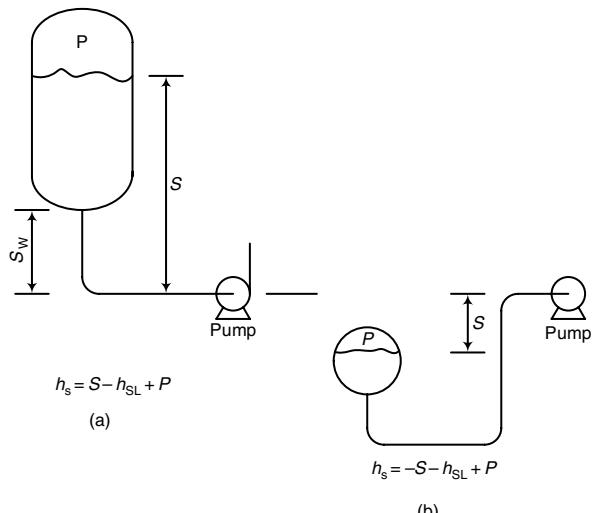
**Figure 5-22d** Pressure head, positive suction. (Adapted by permission from *Centrifugal Pumps Fundamentals*, Ingersoll-Rand Co., Washington, NJ 07882.)



**Figure 5-22e** Pressure head with negative suction. (Adapted by permission from *Centrifugal Pumps Fundamentals*, Ingersoll-Rand Co., Washington, NJ 07882.)



**Figure 5-22f** Pumping arrangement for Example 5-2. (Adapted by permission from *Centrifugal Pumps Fundamentals*, Ingersoll-Rand Co., Washington, NJ 07882.)



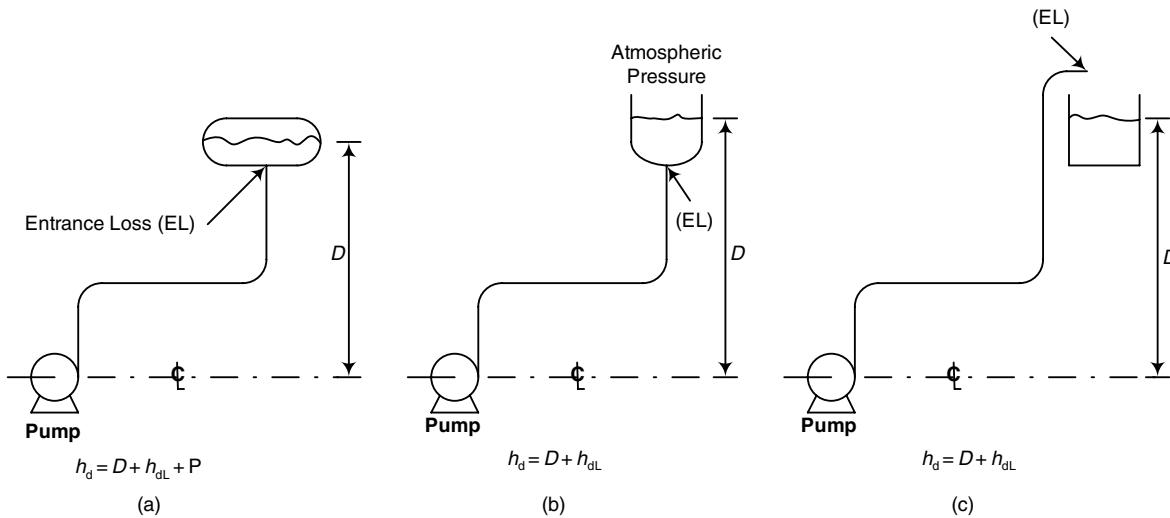
Note: When  $P$  is expressed in absolute pressure units,  $h_s$  will be in absolute units. If  $P$  is less than atmospheric pressure:  $P$  is (-) if expressed as a gauge reading and will be a negative feet of liquid.  $P$  is (+) if expressed in absolute units. The friction loss  $h_{SL}$  includes any entrance or exit losses and other such fittings in the system.

**Figure 5-23** Typical suction system. (Adapted by permission from carter, R. and Karassik, RP-477. Worthington Corp.)

compared to a pump not required to handle fluid under these conditions. Poor suction condition due to inadequate  $NPSH_A$  is one major contribution to cavitation in pump impellers, and this is a condition at which the pump cannot operate for very long without physical erosion damage to the impeller [14, 15].

Cavitation of a centrifugal pump, or any pump, develops when there is insufficient  $NPSH$  for the liquid to flow into the inlet of the pump, allowing flashing or bubble formation in the suction system and entrance to the pump. Each pump design or "family" of dimensional features related to the inlet and impeller eye area and entrance pattern requires a specific minimum value of  $NPSH$  to operate satisfactorily without flashing, cavitating, and loss of suction flow.

Under cavitating conditions, a pump will perform below its head-performance curve at any particular flow rate. Although the pump may operate under cavitation conditions, it will often be noisy because of collapsing vapor bubbles and severe pitting, and erosion of the impeller often results. This damage can become so severe as to completely destroy the impeller and create excessive clearances in the casing. To avoid these problems, the following are a few situations to watch.



**Note:**

For a system evaluation, including suction and discharge, the units of  $P$  must be the same either gauge or absolute, expressed as feet of fluid.

The friction losses from the pump to the vessel include any entrance or exit losses. Unless velocities are high, these losses are usually negligible.

**Figure 5-24** Typical discharge systems.



The NPSH<sub>A</sub> available from or in the liquid system on the suction side of a pump is expressed (corrected to pump centerline) as

$$\text{NPSH}_A = (p'_a - p'_{vp}) \pm S - h_{SL} \quad (5-9)$$

$$NPSH_A = (P_a - P_{vp}) (2.31 \text{ ft/psi/sp gr}) \pm S - h_{SL} \quad (5-10)$$

In Metric units,

$$\text{NPSH}_A = (P_a - P_{vp}) (10.2 \text{ m/bar/sp gr}) \pm S - h_{SL} \quad (5-10a)$$

where

$p'_a$  or  $P_a$  represents the absolute pressure in the vessel (or atmospheric) on the liquid surface on the suction side of the pump.  
 $p'_{vp}$  or  $P_{vp}$  represents the absolute vapor pressure of the liquid at the pumping temperature.

$h_{SL}$  is the suction line, valve, fitting, and other friction losses from the suction vessel to the pump suction flange.

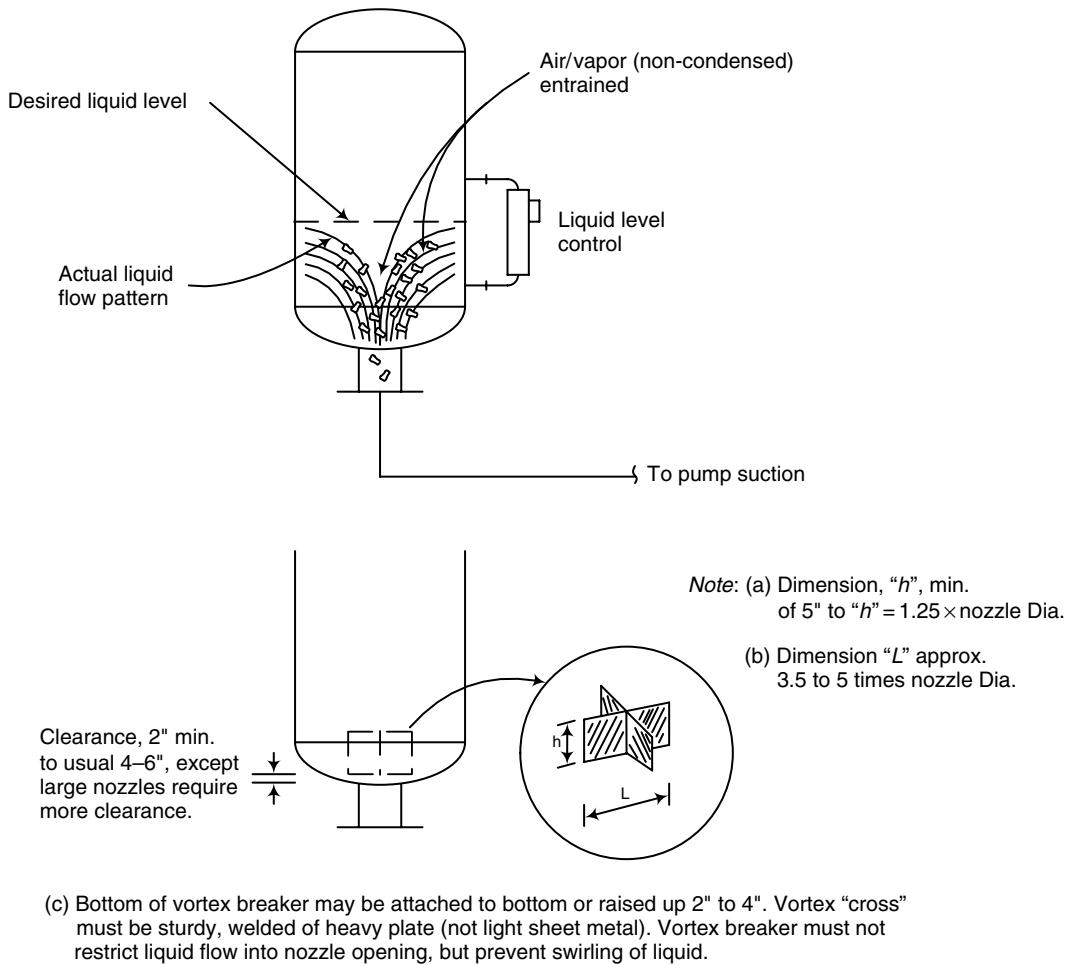
$S$  may be  $(+)$  or  $(-)$ , depending on whether static head or static lift is involved in the system.

This available value of  $NPSH_A$  (of the system) must always be greater by a minimum of 2 ft (610 mm) and preferably three or more feet than the  $NPSH_R$  stated by the pump manufacturer or shown on the pump curves in order to overcome the pump's internal hydraulic loss and the point of lowest pressure in the eye of the impeller. The  $NPSH$  required by the pump is a function of the physical dimensions of casing, speed, specific speed, and type of impeller, and must be satisfied for proper pump performance. The pump manufacturers must always be given complete suction conditions if they are expected to recommend a pump to give long and trouble-free service.

As the altitude of an installation increases above sea level, the barometric pressure, and hence  $p_a'$  or  $P_a$ , decreases for any open vessel condition. This decreases the NPSH<sub>A</sub>.

Figure 5-18a represents typical manufacturer's performance curves. The values of  $NPSH_R$  given are the minimum values required at the pump suction. As mentioned, good practice requires that the  $NPSH_A$  be at least 2 ft (610 mm) of liquid above these values. It is important to recognize that the  $NPSH_R$  and suction lift values are for handling water at about 70° F (21.1° C). To use with other liquids it is necessary to convert to the equivalent water suction lift at 70° F (21.1° C) and sea level.

Total Suction Lift (as water at 70° F) = NPSH<sub>A</sub> (calculated for fluid system) - 33 ft. The vapor pressure of water at 70° F is 0.36 psia.



**Figure 5-25** Liquid vortex in vessel and suggested design of vortex breaker.

### EXAMPLE 5-3 Suction Lift

What is the suction lift value to be used with the pump curves of Figure 5-18a if a gasoline system calculates an  $NPSH_A$  of 15 ft?

#### Solution

Total Suction Lift (as water) =  $15 - 33 = -18$  ft. Therefore, a pump must be selected which has a lift of at least 18 ft. The pump of Figure 5-18a is satisfactory using an interpolated suction lift line between the dotted curves for 16 ft and 21 ft of water. The performance of the pump will be satisfactory in the region to the left of the new interpolated 18-ft line. Proper performance should not be expected near the line.

If the previous system were at sea level, consider the same pump with the same system at an altitude of 6000 ft. Here the barometric pressure is 27.4 ft of water. This is  $34 - 27.4 = 6.6$  ft less than the sea level installation. The new  $NPSH_A$  will be  $15 - 6.6 = 8.4$  ft. Referring to the pump curve of Figure 5-18a, it is apparent that this pump cannot do greater than 21 ft suction lift of water or 12 ft  $NPSH_R$  of liquid (fluid). Total Suction Lift of water =  $8.4 - 33 = -24.6$  ft. The pump curves show that 21 ft suction lift of water is all the pump can do, hence the 24.6 ft is too great. A different pump must be used which can handle this high a suction lift. Such a pump may become expensive, and it may be preferable to use a positive displacement pump for this high lift. Normally lifts are not considered reasonable if over 20 ft.

**EXAMPLE 5-4****NPSH<sub>A</sub> in Open Vessel System at Sea Level**

Refer Figure 5-20 for this example.

Conditions: At sea level, atmospheric pressure,  $P_a = 14.7 \text{ psia}$  (1.013 bara)

Assume liquid is water at  $85^\circ \text{ F}$  ( $29.4^\circ \text{ C}$ ), vapor pressure,  $P_{vp} = 0.6 \text{ psia}$  (0.04 bara)

Assume tank liquid level is 10 ft above centerline of pump, then  $S = +10 \text{ feet}$  (3.05 m)

Assume that friction losses have been calculated to be 1.5 ft,  $h_{SL} = 1.5 \text{ ft}$  (0.46 m).

By using Eq. (5-10)

$$\begin{aligned}\text{NPSH}_A &= (14.7 - 0.6)(2.31/0.997) + 10 - 1.5 \\ &= 41.2 \text{ ft (good)}\end{aligned}$$

In Metric units, by using Eq. (5-10a)

$$\begin{aligned}\text{NPSH}_A &= (1.013 - 0.04)(10.2/0.997) + 3.05 - 0.46 \\ &= 12.54 \text{ m (41.2 ft)}\end{aligned}$$

*Note:* For the worst case, which is an empty tank, "S" becomes  $S_w$  on the diagram.

**EXAMPLE 5-5****NPSH<sub>A</sub> in Open Vessel Not at Sea Level**

Refer Figure 5-21 for this example.

Conditions: Vessel is at altitude 1500 ft (457.2 m), where atmospheric pressure is  $P_a = 13.92 \text{ psia}$  (0.96 bara)

Liquid: Water at  $150^\circ \text{ F}$  ( $65.56^\circ \text{ C}$ ), vapor pressure  $P_{vp} = 3.718 \text{ psia}$  (0.256 bara) and specific gravity, sp gr = 0.982

Assume vessel liquid level is 12 ft (3.657 m) below centerline of pump,  $S_L = -12$  (-3.657 m)

Friction losses: Assume as calculated to be 1.1 ft (0.335 m) of liquid.

By using Eq. (5-10)

$$\begin{aligned}\text{NPSH}_A &= (13.92 - 3.718)(2.31/0.982) - 12 - 1.1 \\ &= 10.90 \text{ ft}\end{aligned}$$

In Metric units, by using Eq. (5-10a)

$$\begin{aligned}\text{NPSH}_A &= (0.96 - 0.256)(10.2/0.982) - 3.657 - 0.335 \\ &= 3.32 \text{ m (10.90 ft)}\end{aligned}$$

The worst condition case should be calculated using  $S'_L$ , since this represents the maximum lift.

**EXAMPLE 5-6****NPSH Available in Vacuum System**

Refer Figure 5-23a for this example.

Conditions: Vessel is liquid collector at 28 in.Hg Vacuum (referred to a 30-in. barometer). This is  $30 - 28 = 2 \text{ in.Hg abs}$ , or  $P_a = [(14.7/30)](2) = 0.98 \text{ psia}$  (0.067 bara)

Liquid: Water at  $101.2^\circ \text{ F}$  ( $38.4^\circ \text{ C}$ ), vapor pressure = 0.98 psia (0.067 bara)

Assume vessel liquid level is 5 ft (1.524 m) above pump centerline,  $S = +5 \text{ ft}$ , worst case,  $S_w = 2 \text{ ft}$  (0.61 m)

Friction losses: Assume to be 0.3 ft (0.091 m) of liquid

By using Eq. (5-10)

$$\begin{aligned}\text{NPSH}_A &= (0.98 - 0.98)(2.31/0.994) + 5 - 0.3 \\ &= 4.7 \text{ ft}\end{aligned}$$

Worst case = 4.7 ft (not practical design).

The pump selected for this application (water boiling at 0.98 psia) must have a  $\text{NPSH}_R$  less than 4.7 ft preferably about 3–3.5 ft. This is a difficult condition. If possible the vessel should be elevated to make more head ( $S$ ) available which will raise the  $\text{NPSH}_A$ .

In Metric units, by using Eq. (5-10a)

$$\begin{aligned}\text{NPSH}_A &= (0.067 - 0.067)(10.2/0.994) - 1.524 - 0.091 \\ &= 1.433 \text{ m (4.7 ft)}\end{aligned}$$

**EXAMPLE 5-7****NPSH<sub>A</sub> in Pressure System**

Refer Figure 5-23b for this example.

Conditions: Vessel contains butane at  $90^\circ \text{ F}$  ( $32.22^\circ \text{ C}$ ) and 60 psia (4.138 bara) system pressure,  $P_a = 60 \text{ psia}$  (4.138 bara)

Butane vapor pressure,  $P_{vp}$  at  $90^\circ \text{ F}$  ( $32.22^\circ \text{ C}$ ) = 44 psia (3.034 bara), sp gr = 0.58

Assume liquid level is 8 ft (2.438 m) below pump centerline,  $S = -8 \text{ ft}$  (-2.438 m)

Friction losses: Assume to be 12 ft (3.658 m) of liquid.

Then from Eq. (5-10)

$$\begin{aligned}\text{NPSH}_A &= (60.0 - 44)(2.31/0.58) - 8 - 12.0 \\ &= 43.72 \text{ ft}\end{aligned}$$

In Metric units (from Eq. (5-10a)),

$$\begin{aligned}\text{NPSH}_A &= (4.138 - 3.034)(10.2/0.58) - 2.438 - 3.658 \\ &= 13.32 \text{ m (43.7 ft)}\end{aligned}$$

This presents no pumping problem.

**EXAMPLE 5-8**  
**Closed System Steam Surface Condenser NPSH Requirements**

Refer Figure 5-26 for this example.

This is a closed steam surface condenser system with condensate being pumped out to retreatment facilities. From the conditions noted on the diagram,

$$\text{Friction loss in suction line side} = 2.92 \text{ ft}$$

$$\begin{aligned}\text{Absolute pressure in condenser} &= p' = 1.5 \text{ in.Hg abs} \\ &= 1.5(1.137 \text{ ft/in.Hg}) \\ &= 1.71 \text{ ft H}_2\text{O}\end{aligned}$$

$$\text{Water from steam tables at saturation} = 1.5 \text{ in.Hg abs}$$

$$= 91.72^\circ \text{ F}$$

$$\begin{aligned}\text{Vapor pressure, } p'_{vp}, \text{ at } 1.5 \text{ in.Hg abs} &= 1.5(1.137) \\ &= 1.71 \text{ ft H}_2\text{O}\end{aligned}$$

$$\text{NPSH}_A = (1.71 - 1.71) + 10 - 2.92 = +7.08 \text{ ft}$$

The suction head or lift for the pump (separate calculation from  $\text{NPSH}_A$ ) is as follows.

The 28.42 in.Hg vacuum (gauge) is equivalent to 1.5 in.Hg abs

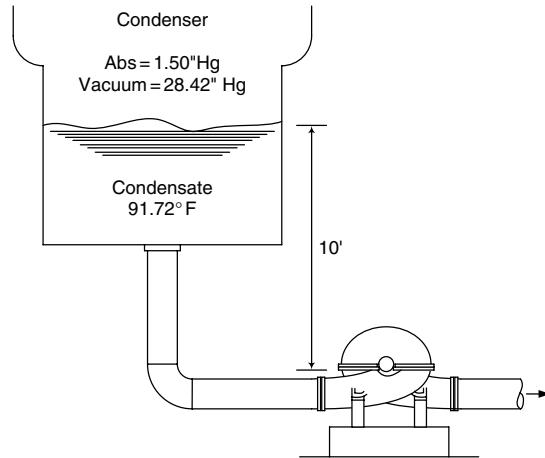
$$\begin{aligned}28.42 \text{ in. vacuum (1.137 ft/in.Hg)} &= 32.31 \text{ ft H}_2\text{O} \\ \text{Static submergence} &= 10.0 \text{ ft (see figure)} \\ \text{Friction/entrance losses} &= 2.92 \text{ ft} \\ \text{Net static submergence} &= 7.08 \text{ ft} \\ \text{Equivalent suction lift} &= 25.23 \text{ ft (Note: } 32.31 - 7.08 \\ &\quad (= \text{vacuum effect less net submergence})\end{aligned}$$

Note that the equivalent suction lift must be added to the total discharge head for the pump system to obtain the total system head. Keep in mind that the work the pump must accomplish is overcoming the suction losses (+ or -) plus the discharge losses, that is, + discharge loss (all) - (+ if head, or - if lift on suction losses (all)). Thus, the suction lift becomes a (-) (-) or a (+) to obtain the total system head. Keep in mind that a vacuum condition on the suction of a pump never helps the pump, but in effect is a condition that the pump must work to overcome.

### REDUCTIONS IN $\text{NPSH}_R$

Limitations for use of the Hydraulic Institute NPSH reduction chart are as follows (Figure 5-28) [8].

1. The NPSH reductions should be limited to 50% of the  $\text{NPSH}_R$  required by the pump for cold water, which is the fluid basis of the manufacturer's  $\text{NPSH}_R$  curves.
2. It is based on handling pure liquids, without entrained air or other non-condensable gases, which adversely affect the pump performance.
3. Absolute pressure at the pump inlet must not be low enough to release non-condensables of pure liquids. If such release can occur, then the  $\text{NPSH}_R$  would need to be increased above that of the cold water requirements to avoid cavitation and poor pump performance.



**Figure 5-26** Surface condenser condensate removal. Closed system steam surface condenser NPSH requirements. (By permission from *Cameron Hydraulic Data*, 16th ed., Ingersoll-Rand Co., 1979, pp. 1-12.)

4. For fluids, the worst actual pumping temperature should be used.
5. A factor of safety should be applied to ensure that NPSH does not become a problem.
6. The chart should not be extrapolated beyond NPSH reductions of 10 ft.

**EXAMPLE 5-9**  
**Process Vacuum System**

Refer Figure 5-27 for this example.

For this process example, again using water for convenience, a low pressure, low temperature water is emptied into a vented vessel, and then pumped to the process at a location at about 3000 ft altitude (see Appendix C-6) where atmospheric pressure is approximately 13.2 psia. Water sp gr is at 200°F = 0.963.

Determine the  $\text{NPSH}_A$  for pump:

Then

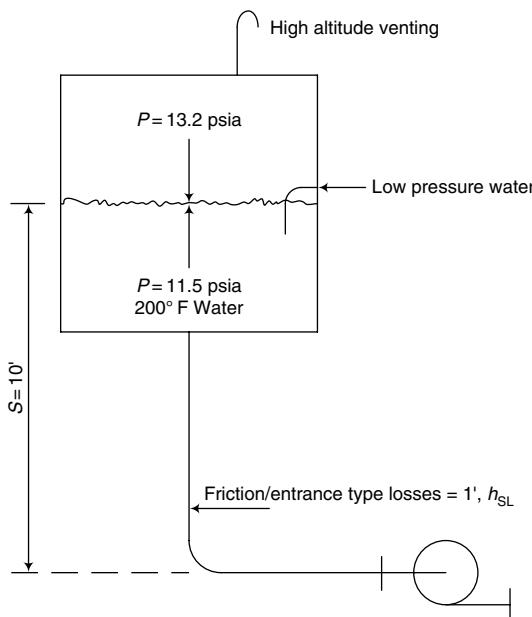
$$\text{NPSH}_A \text{ (according to Eq. (5-10))}$$

$$\begin{aligned}&= (13.2 - 11.5) \left( \frac{2.31 \text{ ft}}{0.963} \right) + 10 - 1.0 \\ &= 13.08 \text{ ft available}\end{aligned}$$

For hydrocarbons and water significantly above room temperatures, the Hydraulic Institute [8] recommends the use of a correction deduction as given in Figure 5-28. This indicates that the  $\text{NPSH}_R$  as given on the pump curves can be reduced for conditions within the range of the curve based on test data.

If the pump of the curve of Figure 5-18a were being used to pump butane at 90°F and 0.58 gravity, the correction multiplier from the NPSH curve is about 0.99 by interpolation. This means that the values of Figure 5-18a should be multiplied by 0.99 to obtain the actual NPSH the pump would require when handling a hydrocarbon of these conditions. The correction does not apply to other fluids.

If the system pressure were 46 psia, then  $\text{NPSH}_A = (46 - 44)(2.31/0.58) - 8 - 12 = -12 \text{ ft}$ , and this is an impossible and unacceptable condition. This means liquid will flash in the line and in the impeller, and cannot be pumped. And NPSH must always be positive in sign.



**Figure 5-27** High altitude process vacuum system, NPSH requirements.

#### EXAMPLE 5-10

#### Corrections to $NPSH_R$ for Hot Liquid Hydrocarbons and Water

In Figure 5-28, use the dashed example lines at a temperature of 55°F for propane [8], and follow the vertical line to the propane vapor pressure dashed line, which reads 100 psia vapor pressure. Then follow the slant lines (parallel) to read the scale for NPSH reductions, that is, feet at 9.5 ft.

Now the pump selected reads  $NPSH_R$  on its pump performance curve of 12 ft for cold water service.

$$\text{Now, half of } 12 \text{ ft} = 6 \text{ ft}$$

$$\text{But in Figure 5-28 the reduction} = 9.5 \text{ ft}$$

Corrected value of  $NPSH_R$  to use = 6 ft, since 9.5 ft is > half the cold water value

#### EXAMPLE 5-11

#### Alternate to Example 5-10

Assume that a boiler feed water is being pumped at 180°F. Read the chart in Figure 5-28 and the water vapor pressure curve, and follow over to read NPSH reduction = 0.45 ft. A pump selected for the service requires 6 ft cold water service  $NPSH_R$ :

$$\text{Half of } 6 = 3 \text{ ft}$$

$$\text{Value from chart for } 180^\circ \text{ F} = 0.45 \text{ ft reduction}$$

Then correct  $NPSH_R$  to use =  $6 - 0.45 = 5.55$  ft required by the pump for this service

## 5.10 SPECIFIC SPEED

The specific speed of a centrifugal pump correlates the basic impeller types as shown in Figure 5-29.

The formula for specific speed index number is

$$N_s = \frac{n\sqrt{Q}}{H^{0.75}} \quad (5-11)$$

where  $Q$  is the gpm capacity at speed  $n$  in rpm and head  $H$ .  $H$  is the total head per stage, in feet.

The principle of dynamical similarity expresses the fact that two pumps geometrically similar to each other will have similar (not necessarily identical) performance characteristics. The three main characteristics of capacity, head, and rotative speed are related into a single term designated "specific speed" [10]. The expression for specific speed is the same whether the pump has a single or double suction impeller.

The principal significance of specific speed for the process engineer is to evaluate the expected performance of a second pump in a particular manufacturer's series while basing it on the known performance (or curve) at the point of optimum efficiency of a first and different size pump. In effect the performance of any impeller of a manufacturer's homologous series can be estimated from the known performance of any other impeller in the series, at the point of optimum efficiency. Figures 5-30 and 5-31 represent the standardized conditions of essentially all pump manufacturers.

A typical "operating specific speed" curve is shown in Figure 5-32 and represents a technique for plotting the specific speed on the operating performance curve. Figure 5-32 represents a 6-in. pump operating at 1760 rpm, with maximum efficiency at 1480 gpm and 132 ft head [10]. The operating specific speed is zero at no flow and increases to infinity at the maximum flow of 2270 gpm and zero head. Stable operations beyond about 1600–1700 gpm cannot be planned from such a curve with a sharp cutoff drop for head capacity.

"Type specific speed" is defined as that operating specific speed that gives the maximum efficiency for a specific pump and is the number that identifies the pump type [10]. This index number is independent of the rotative speed at which the pump is operating, because any change in speed creates a change in capacity in direct proportion and a change in head that varies as the square of the speed [10]. The specific speed of the pump is reasonably close to the conditions of maximum efficiency, and Figure 5-29 illustrates the range of typical specific speed index numbers for particular types of impellers.

#### EXAMPLE 5-12

#### "Type Specific Speed"

In Figure 5-32, where the pump operates at 1760 rpm (a standard motor speed under load) and has maximum efficiency at 1480 gpm and 132 ft head, the "type" specific speed using Eq. (5-11) is

$$N_s = \left[ \frac{1760\sqrt{1480}}{(132)^{0.75}} \right] = 1740 \text{ rpm}$$

In Metric units,

$$N_s = \left[ \frac{n\sqrt{Q}}{(gH)^{0.75}} \right] \quad (5-11a)$$

where

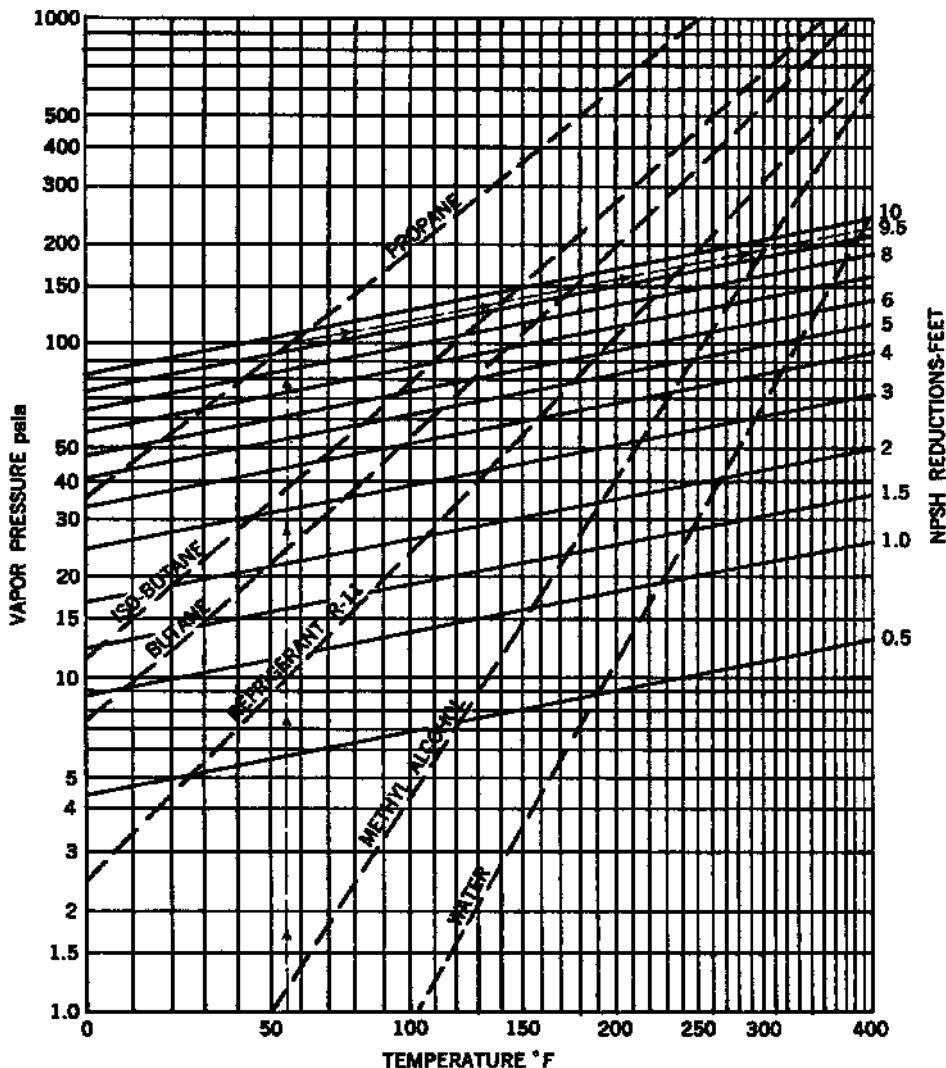
$g$  = acceleration due to gravity,  $9.81 \text{ m/s}^2$

$H$  = total head, m

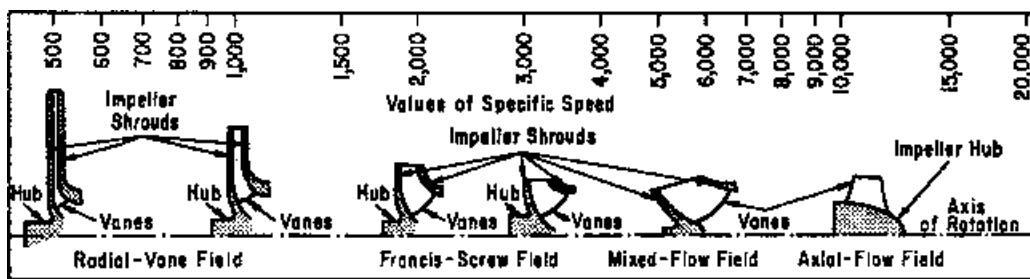
$n$  = speed, rps

$Q$  = flow rate,  $\text{m}^3/\text{s}$ .

Figure 5-29 indicates the general type of impeller installed.



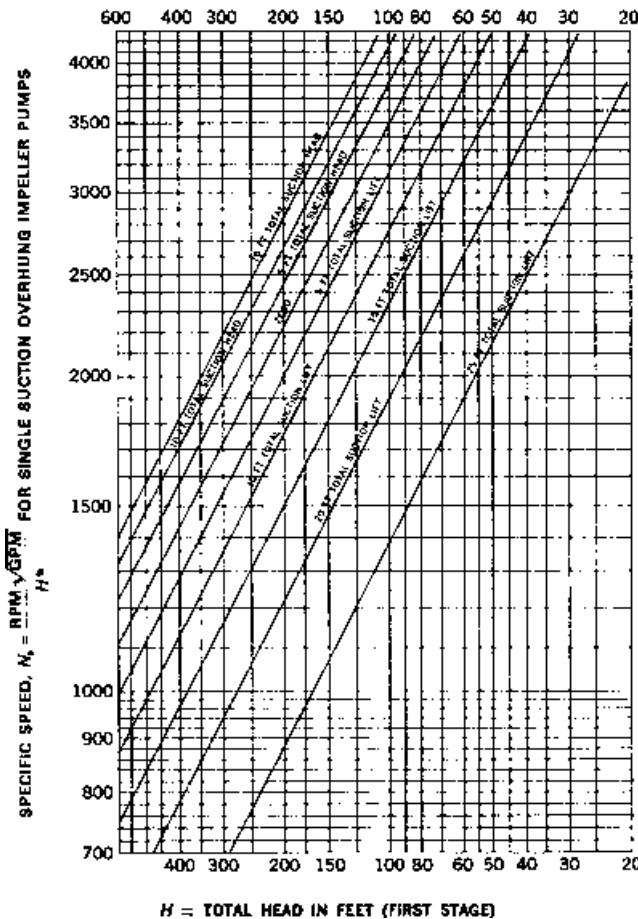
**Figure 5-28** NPSH reductions for pumps handling hydrocarbon liquids and high temperature water. (Note: Do not use for other fluids.) (By permission from *Hydraulic Institute Standards for Centrifugal, Rotary, and Reciprocating Pumps*, Hydraulic Institute, 13th ed., 1975.)



**Figure 5-29** Impeller designs and corresponding specific speed range. (By permission from *Standards of the Hydraulic Institute*, 10th ed.) (also see [18], Hydraulic Institute, 13th ed., 1975.)

The specific speed of a given pump type must not exceed the specific speed values presented by the Hydraulic Institute [8]. This is based on a known or fixed condition of suction lift, and relates speed, head, and capacity. This index is a valuable guide in establishing the maximum suction lifts and minimum suction heads

to avoid cavitation of the impeller with resultant unstable hydraulic performance and physical damage. For a given set of conditions on the suction and discharge of a pump, a slow rotative speed will operate safer at a higher suction lift than a pump of higher rotative speed.



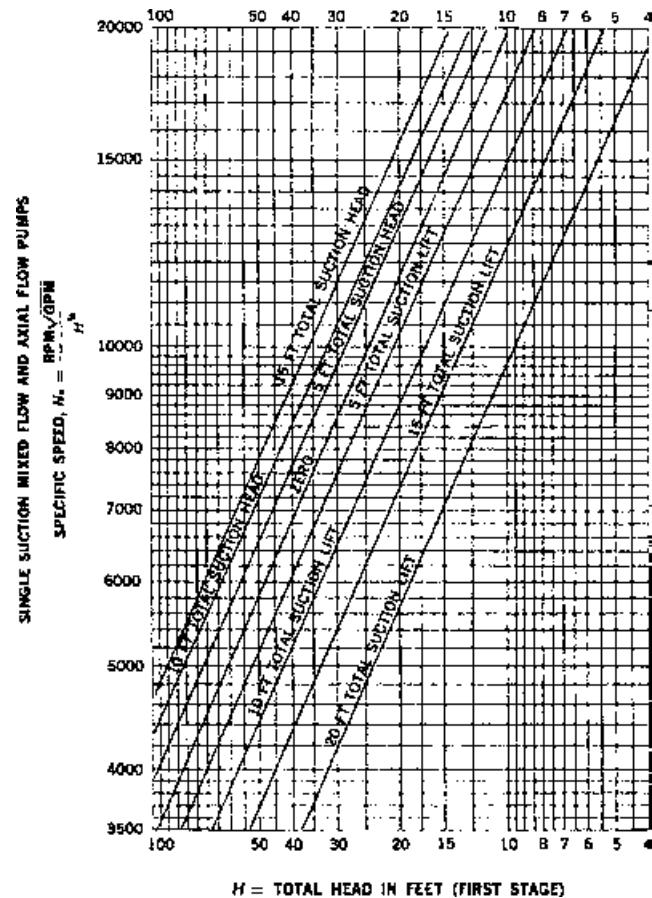
**Figure 5-30** Upper limits of specific speeds for single suction overhung impeller pumps handling clear water at 85°F at sea level. (By permission from *Hydraulic Institute Standards for Centrifugal, Rotary, and Reciprocating Pumps*, Hydraulic Institute, 13th ed., 1975.)

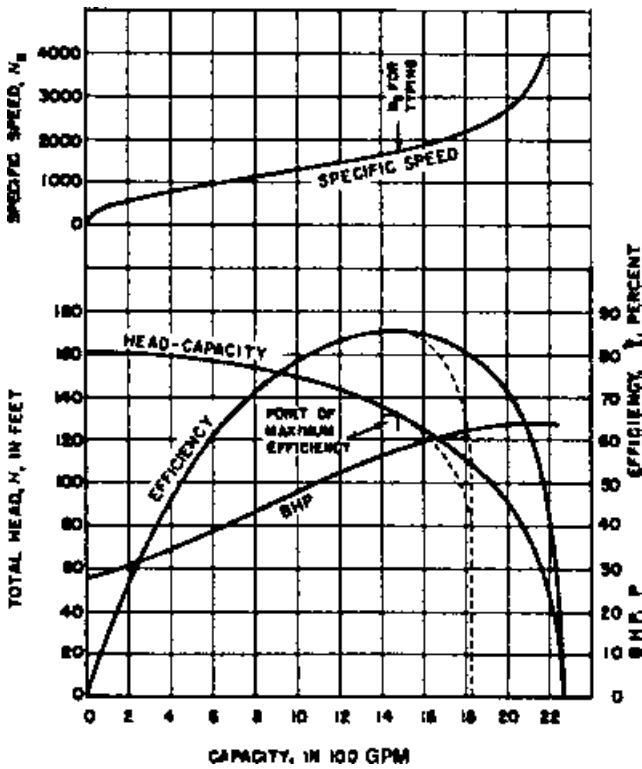
### 5.11 ROTATIVE SPEED

The rotative speed of a pump is dependent upon the impeller characteristics, fluid type,  $NPSH_A$ , and other factors for its final determination. The most direct method is by reference to manufacturer's performance curves. When a seemingly reasonable selection has been made, the effect of this selected speed on the factors such as  $NPSH_R$ , suction head or lift, fluid erosion and corrosion, and so on must be evaluated. For many systems these factors are of no concern or consequence.

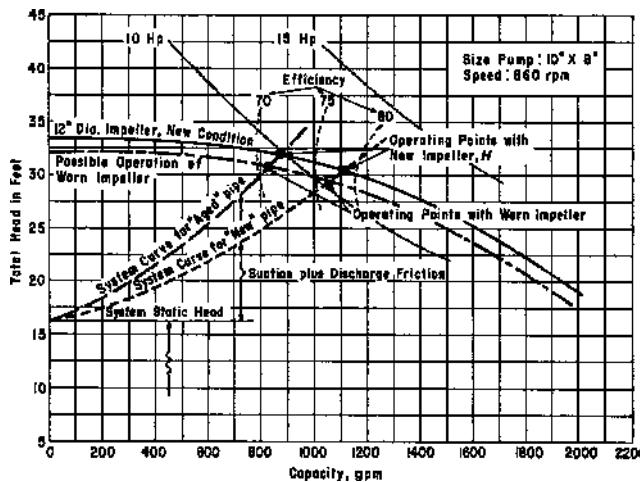
Normal electric motor speeds run from the standard induction speeds for direct connection of 3600, 1800, and 1200 rpm to the lower speed standards of the synchronous motors, and then to the somewhat arbitrary speeds established by V-belt or gear drives. For some cases, the pump speed is set by the type of drivers available, such as a gasoline engine.

Electric motors in pump application never run at the "standard" rotative design speeds noted above, but rotate at about (with some deviation) 3450, 1750, and 1150 rpm, which are the speeds that most pump manufacturers use for their performance curves. If the higher numbers were used (motor designated or name plate) for pump performance rating, the pumps would not meet the expected performance, because the motors would not be actually rotating fast enough to provide the characteristic performance curves for the specific size of impeller.





**Figure 5-32** Typical centrifugal pump characteristic curve with auxiliary specific speed curve. Double-suction, single-stage, 6-in. pump, operating at 1760 rpm constant speed. (By permission, Karassik, I. and R. Carter, *Centrifugal Pumps*, McGraw-Hill Book Co., Inc., 1960, p. 197.)



**Figure 5-33** System head curves for single pump installation.

$h_{DL}$ . The total static head is  $D - S_L$ , or  $[(D + D') - (-S_L)]$  if siphon action is ignored, and  $[(D + D') - S'_L]$  for worst case, good design practice.

#### Procedure:

- Calculate the friction losses  $h_{SL}$  and  $h_{DL}$  for three or more arbitrarily chosen flow rates, but rates which span the area of interest of the system.

- Add  $[h_{SL} + h_{DL} + (D \pm S)]$  for each value of flow calculated. These are the points for the system head curve.
- Plot the GPM values vs the points of step 2 above.
- The intersection of the system curve with the pump impeller characteristic curve is the operating point corresponding to the total head  $H$ . This point will change only if the external system changes. This may be accomplished by adding resistance by partially closing valves, adding control valves, or decreasing resistance by opening valves or making pipe larger, and so on.

For the system of Figure 5-21, the total pumping head requirement is

$$\begin{aligned} H &= (D + h_{DL}) - [-S_L + (-h_{SL})] \\ &= (D + h_{DL}) + (S_L + h_{SL}) \end{aligned} \quad (5-13)$$

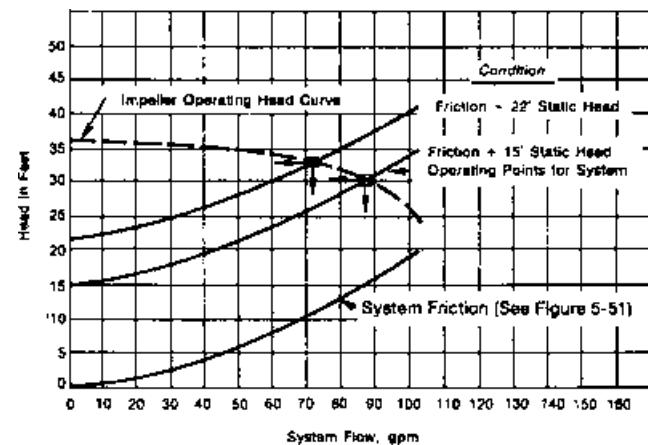
The total static head of the system is  $[D - (-S)]$  or  $(D + S)$ , and the friction loss is still  $h_{DL} + h_{SL}$ , which includes the heat exchanger in the system.

For a system made up of the suction side as shown in Figure 5-23a and the discharge as shown in Figure 5-24a, the total head is

$$H = D + h_{DL} + P_1 - [S - h_{SL} + P_2] \quad (5-14)$$

where  $P_2$  is used to designate a pressure different than  $P_1$ . The static head is  $[(D + P_1) - (S + P_2)]$ , and the friction head is  $h_{DL} + h_{SL}$ .

Figure 5-34 illustrates the importance of examining the system as it is intended to operate, noting that there is a wide variation in static head, and therefore there must be a variation in the friction of the system as the GPM delivered to the tank changes. It is poor and perhaps erroneous design to select a pump which will handle only the average conditions, for example, about 32 ft total head. Many pumps might operate at a higher 70-ft head when selected for a lower GPM value; however, the flow rate might be unacceptable to the process.



**Figure 5-34** System head curves for variable static head.

**EXAMPLE 5-13****System Head Using Two Different Pipe Sizes in Same Line**

The system of Figure 5-35 consists of the pump taking suction from an atmospheric tank and 15 ft of 6-in. pipe plus valves and fittings; on the discharge there is 20 ft of 4-in. pipe in series with 75 ft of 3-in. pipe plus a control valve, block valves, fittings, and so on. The pressure of the discharge vessel (bubble cap distillation tower) is 15 psig, with water as the liquid at 40°F in a 6-in. suction pipe (using Cameron Tables – Table 4-46). To simplify calculations for greater accuracy, use detailed procedure of Chapter 4.

Pipe or Fitting Loss	200 gpm	300 gpm
Loss, ft/100 ft		
For 15 ft	15.0 ft	
Two 90° ells, eq.	22.8	
Gate valve, open	3.2	
Total	41.0 ft	0.24 ft
		0.51 ft

The total suction head =  $h_s = +7 - 0.24 = +6.76$  at 200 gpm

$$h_s = +7 - 0.51 = +6.49 \text{ at } 300 \text{ gpm}$$

Discharge:

$$\begin{aligned} \text{Total static head} &= 45 - 7 + 15 \text{ (2.31 ft/psi/sp gr)} \\ &= 72.65 \text{ ft, sp gr} = 1.0 \end{aligned}$$

Composite head curve

$$\text{at } 200 \text{ gpm, head} = 72.65 + 0.24 + 2.65 + 34.0 = 109.54 \text{ ft}$$

$$\text{at } 300 \text{ gpm, head} = 72.65 + 0.51 + 5.57 + 72.0 = 150.73 \text{ ft}$$

Total head on pump at 300 gpm:

$$H = 45 + 15 \text{ (2.31)} + 5.57 + 72.0 - 7 + 0.51 = 150.73 \text{ ft}$$

The head at 200 gpm (or any other) is developed in the same manner.

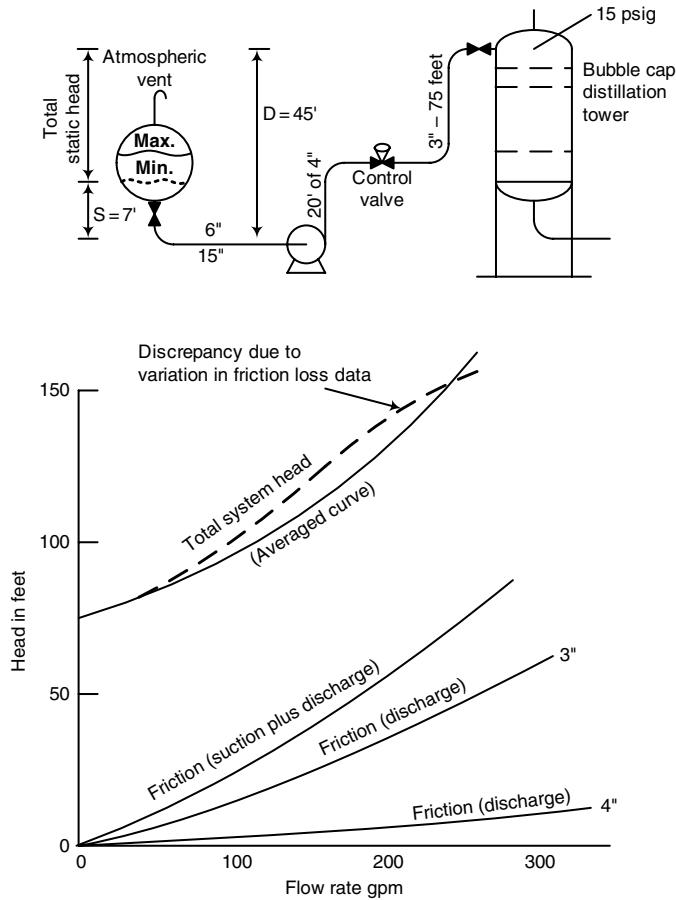
	4-in. Pipe		3-in. Pipe	
	200 gpm	300 gpm	200 gpm	300 gpm
Loss, ft/100 ft	4.29	9.09	16.1	34.1
For 20 ft	20.0	20.0		
For 75 ft			75.0	75.0
Two, 3", 90° ells, eq.			8	8
One, 4", 90° ells, eq.	4.6	4.6		
One Gate valve open			1.7 ft	1.7 ft
Total, equivalent ft	24.6	24.6	84.7	84.7
Friction loss, ft fluid	1.06	2.23	13.6	28.8
Control valve at 60% of total, ft	1.59	3.34	20.4	43.2
Total discharge friction loss, ft	2.65	5.57	34.0	72.0

**EXAMPLE 5-14****System Head for Branch Piping with Different Static Lifts**

The system of Figure 5-36 has branch piping discharging into tanks at different levels [17]. Following the diagram, the friction in the piping from point B to point C is represented by the line B–P–C. At point C, the flow will all go to tank E unless the friction in line C–E exceeds the static lift *b* required to send the first liquid into D. The friction for the flow in line C–E is shown on the friction curve, as is the corresponding friction for flow through C–D. When liquid flows through both C–E and C–D, the combined capacity is the sum of the values of the individual curves read at constant head values, and given on curve (C–E) + (C–D). Note that for correctness the extra static head *b* required to reach tank D is shown with the friction head curves to give the total head above the “reference base.” This base is an arbitrarily but conveniently selected point.

The system curves are the summation of the appropriate friction curves plus the static head *a* required to reach the base point. Note that the suction side friction is represented as a part of B–P–C in this example. It could be handled separately, but must be added in for any total curves. The final total system curve is the friction of (B–P–C) + (C–E) + (C–D) plus the head *a*. Note that liquid will rise in pipe (C–D) only to the reference base point unless the available head is greater than that required to flow through (C–E), as shown by following curve (B–P–C) + (C–E) + *a*. At point Y, flow starts in both pipes, at a rate corresponding to the *Y* value in gpm. The amounts flowing in each pipe under any head conditions can be read from the individual System curves.

The principles involved here are typical and may be applied to many other system types.



**Figure 5-35** System head using two different pipe sizes in same line.

### 5.13 POWER REQUIREMENTS FOR PUMPING THROUGH PROCESS LINES

A fluid flows of its own accord as long as its energy per unit mass decreases in the direction of flow. Alternatively, it will flow in the opposite direction only if a pump is used to supply energy, and to increase the pressure at the upstream end of the system. From the energy balance equation in Chapter 4 (i.e., neglecting the internal energy and the heat input to the system).

$$\frac{\Delta P}{\rho} + \alpha \frac{\Delta v^2}{2} + g\Delta z + W_s + f = 0 \quad (5-15)$$

The work done on unit mass of fluid ( $-W_s$ ) is

$$(-W_s) = \frac{\Delta P}{\rho} + \alpha \frac{\Delta v^2}{2} + g\Delta z + f \quad (5-16)$$

where

$\Delta P$  = difference in the pressure between points 1 and 2

$\Delta v$  = difference in the velocity between points 1 and 2

$\Delta z$  = difference in the distance/elevation between points 1 and 2

$f$  = irreversible energy dissipated between points 1 and 2.

The total rate at which energy must be transferred to the fluid is  $G (-W_s)$ , where  $G$  is the mass flow rate, and the power supplied  $P$  is

$$P = G (-W_s) = G \left( \frac{\Delta P}{\rho} + \alpha \frac{\Delta v^2}{2} + g\Delta z + f \right) = G h g \quad (5-17)$$

where

$G$  = mass flow rate, kg/s

$h$  = total head, m

$g$  = acceleration due to gravity,  $9.81 \text{ m/s}^2$ .

The overall power requirement taking into account the pump efficiency  $e$  is

$$P = \frac{1}{e} G h g \quad (5-18)$$

where

$e$  = pump efficiency, fraction.

### HYDRAULIC POWER

Once the flow and corresponding system resistance have been established, the pumping hydraulic power can be calculated from the following:

$$\text{hydraulic, HP} = \frac{(Q)(H)(\text{sp gr})}{3960} \quad (5-19)$$

where

$Q$  = flow, gpm

$H$  = total head, ft

sp gr = Specific gravity of liquid.

or

$$\text{hydraulic, hp} = \frac{(Q)(\Delta P)}{1714} \quad (5-20)$$

where

$\Delta P$  = differential pressure,  $\text{lb/in}^2$ .

In Metric units, the pump hydraulic power output  $P_Q$  is

$$P_Q = \frac{(\rho)(Q)(H)}{367 \times 10^3} \text{ kW} \quad (5-21)$$

where

$\rho$  = fluid density,  $\text{kg/m}^3$

$Q$  = flow,  $\text{m}^3/\text{h}$

$H$  = total head, m.

or

$$P_Q = \frac{(Q)(\Delta P)}{36} \text{ kW} \quad (5-22)$$

where

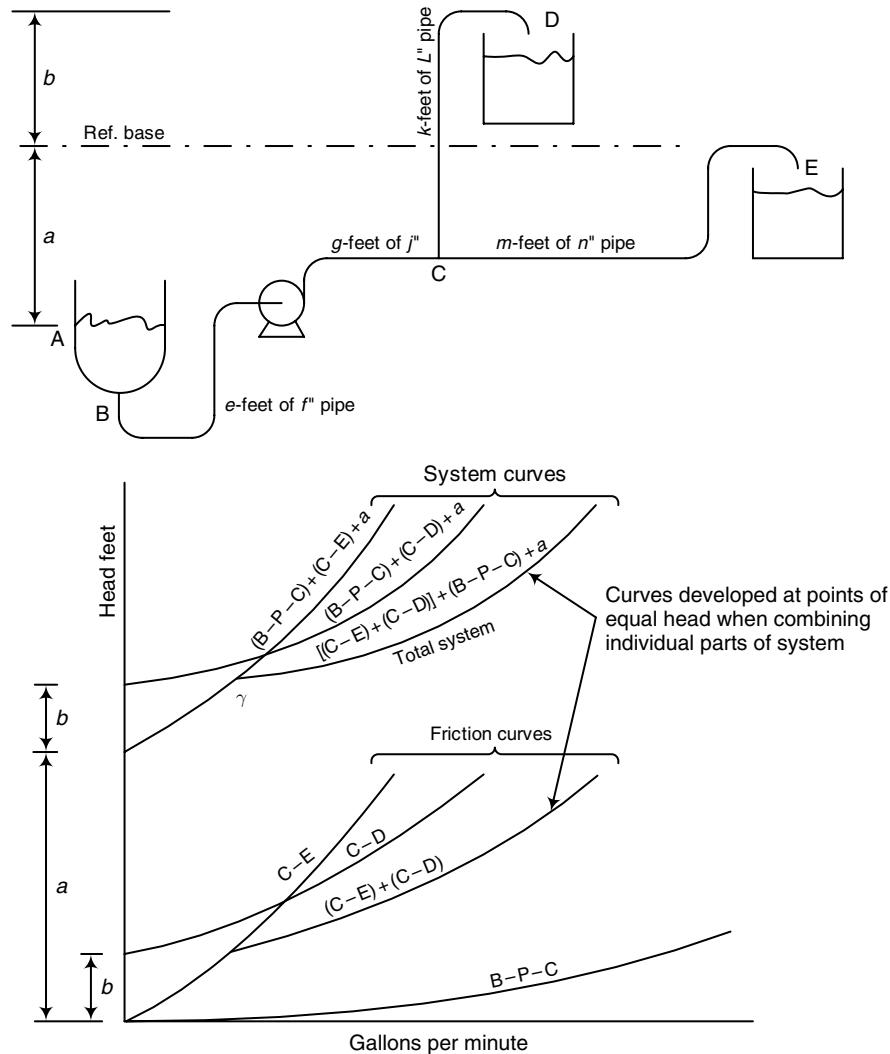
$\Delta P$  = differential pressure, bar.

The pump power input

$$P = \frac{P_Q}{e} = \frac{(\rho)(Q)(H)}{367e \times 10^3} \text{ kW} \quad (5-23)$$

where

$e$  = pump efficiency, fraction.



**Figure 5-36** System head for branch piping with different static lifts.

#### RELATIONS BETWEEN HEAD, HORSEPOWER, CAPACITY, SPEED

##### *Brake Horsepower Input at Pump*

$$\text{BHP} = (Q)(H) \frac{(\text{sp gr})}{(3960e)} \quad (5-24)$$

where

$H$  = total head, ft

$Q$  = flow, gpm

sp gr = specific gravity of liquid

$e$  = hydraulic power, HP/BHP. The efficiency  $e$  (fraction) is the ratio of power out to power absorbed.

Water or liquid horsepower [10]

$$\text{WHP} = (Q)(H) \frac{(\text{sp gr})}{3960} \quad (5-25)$$

The difference between the brake horsepower and the water or liquid horsepower is the pump efficiency. The requirement in

either case is the horsepower input to the shaft of the pump. For that reason, the brake horsepower represents the power required by the pump, which must be transmitted from the driver through the drive shaft through any coupling, gearbox, and/or belt drive mechanism to ultimately reach the driven shaft of the pump. Therefore, the losses in transmission from the driver to the pump itself must be added to the input requirement of the driven pump and are not included in the pump's BHP requirement.

Pump efficiency [8]

$$= \frac{\text{LHP} \text{ (energy delivered by pump to fluid)}}{\text{BHP} \text{ (energy to pump shaft)}} \quad (5-26)$$

Overall efficiency [8]

$$= \frac{\text{WHP} \text{ (energy delivered by pump to fluid)}}{\text{EHP} \text{ (energy supplied to input side of pump's driver)}} \quad (5-27)$$

where

EHP = electrical horsepower

WHP = liquid horsepower.

For the rising type characteristic curve, the maximum BHP required to drive the pump over the entire pumping range is expressed as a function of the BHP at the point of maximum efficiency for any particular impeller diameter [18].

$$\text{BHP (max)} = 1.18 \text{ (BHP at max efficiency point)} \quad (5-28)$$

Unless specifically identified otherwise, the BHP values read from a manufacturer's performance curve represent the power only for handling a fluid of viscosity about the same as water and a specific gravity the same as water; that is, sp gr = 1.0. To obtain actual horsepower for liquids of specific gravity other than 1.0, the curve values must be multiplied by the gravity referenced to water. Viscosity corrections are discussed in another section. Good design must allow for variations in these physical properties.

## DRIVER HORSEPOWER

The driver horsepower must be greater than the calculated (or value read from curves) input BHP to the shaft of the pump. The mechanical losses in the coupling, V-belt, gearbox, or other drive plus the losses in the driver must be accounted for in order that the driver-rated power output will be sufficient to handle the pump.

Best practice suggests the application of a non-overloading driver to the pump. Thus a motor rated equal to or greater than the minimum required BHP of the pump, assuming no other power losses, would be non-overloading over the entire pumping range of the impeller. It is important to examine the pump characteristic curve and follow the changes in power requirements before selecting a driver.

### EXAMPLE 5-15

About  $3.5 \text{ m}^3/\text{h}$  water at 328 K is pumped through a 2-in. Sch. 40 (ID = 52.5 mm) stainless steel pipe, through a length of 200 m in a horizontal direction and up through a vertical height of 20 m. In the pipe configuration, there is a control valve equivalent to 200 pipe diameters and other pipe fittings equivalent to 80 pipe diameters. Also in the line is a heat exchanger across which there is a loss in head of 2.5 m of water. What power must be required from the pump if it is 70% efficient?

*Solution*

Viscosity of water at 328 K:  $\mu = 0.511 \times 10^{-3} \text{ N s/m}^2$

$$\text{Flowrate} = 3.5 \text{ m}^3/\text{h} = 9.72 \times 10^{-4} \text{ m}^3/\text{s}$$

$$\begin{aligned} \text{Area of flow} &= \frac{\pi D^2}{4} = \frac{\pi (52.5 \times 10^{-3})^2}{4} \\ &= 2.16 \times 10^{-3} \text{ m}^2 \end{aligned}$$

Thus

$$\text{velocity, } v = \frac{Q}{A} = \frac{9.72 \times 10^{-4}}{2.16 \times 10^{-3}} = 0.45 \text{ m/s}$$

$$\begin{aligned} \text{Reynolds number, } Re &= \frac{\rho v d}{\mu} \\ &= \frac{(1000)(0.45)(0.0525)}{(0.511 \times 10^{-3})} \\ &= 46,233 \text{ (Turbulent)} \end{aligned}$$

Friction factor:

Using Chen's explicit Eq. (4-35) to calculate the friction factor  $f_C$ ,

Pipe roughness of stainless steel,  $\varepsilon = 0.045 \text{ mm}$

Relative roughness is

$$\begin{aligned} \frac{\varepsilon}{D} &= \frac{0.045}{52.5} = 0.00857 \\ A &= \frac{\varepsilon/D}{3.7} + \left( \frac{6.7}{Re} \right)^{0.9} \\ &= \left( \frac{0.00087}{3.7} \right) + \left( \frac{6.7}{46,233} \right)^{0.9} = 5.85895 \times 10^{-4} \end{aligned}$$

$$\begin{aligned} \frac{1}{\sqrt{f_C}} &= -4 \log \left( \frac{\varepsilon}{3.7D} - \frac{5.02}{Re} \log A \right) \\ &= -4 \log_{10} \left[ \frac{0.00087}{3.7} - \frac{5.02}{46,233} \log_{10} (5.85895 \times 10^{-4}) \right] \\ &= 12.92815 \end{aligned}$$

$$\begin{aligned} f_D &= 4f_C \\ &= 4(5.9831 \times 10^{-3}) \\ &= 0.02393 \end{aligned}$$

$$\text{Equivalent length of pipe} = 200 + (280 \times 0.0525) = 214.7 \text{ m}$$

$$\begin{aligned} \text{Head loss, } h_f &= 4f_D \left( \frac{L}{D} \right) \frac{v^2}{2g} = 4f_C \left( \frac{L}{D} \right) \frac{v^2}{2g} \\ &= f_D \left( \frac{L}{D} \right) \frac{v^2}{2g} = \frac{l_f}{g} \\ h_f &= (0.02393) \left( \frac{214.7}{0.0525} \right) \left( \frac{0.45^2}{2 \times 9.81} \right) \\ &= 1.01 \text{ m} \end{aligned}$$

$$\text{Total head to be developed} = (1.01 + 10 + 2.5) = 13.51 \text{ m}$$

$$\begin{aligned} \text{Mass flowrate of water} &= 9.72 \times 10^{-4} \times 1000 \left\{ \frac{\text{m}^3}{\text{s}} \cdot \frac{\text{kg}}{\text{m}^3} \right\} \\ &= 0.972 \text{ kg/s} \end{aligned}$$

$$\begin{aligned} \text{Power supplied} &= Ghg \\ &= (0.972)(13.5)(9.81) \\ &= 129 \text{ W} \end{aligned}$$

$$\text{Power to be required} = 129 \times \left( \frac{100}{70} \right) = 184 \text{ W}$$

For example, referring to Figure 5-18a, if your pump were selected with a 6-in. diameter impeller for a rated normal pumping of 100 gpm, the pump would put out about 138 ft of head of any fluid (neglecting viscosity effects for the moment). The intersection of the 100-gpm vertical line with the 6-in. performance curve would indicate that 5.75 BHP would be required for water (between 5 and 7.5 hp). Therefore, to be non-overloading (i.e., the motor driver will not overheat or lose power) at this condition would require a 7.5 hp motor (if no other losses occur between driver and pump), because there is no standard motor for direct connected service between the standard 5 and 7.5 hp. Now (1) if you know or project that you may need at some time to pump 160 gpm of any fluid with this pump at 160 ft head, then this pump could not be used because it will not physically take an impeller larger than 6.5-in. diameter. However, recognizing this, (2) if you change the external physical piping, valves, and so on, and reduce the head to fit the 6.5-in. impeller curve, at 160 gpm, you could handle 152 ft head (estimated from the curve for a 6.5-in. impeller).

This condition would require a BHP from the pump curve between 7.5 and 10, that is, about 9.25 BHP for the pump's input shaft (for water calculates at 9.03 BHP), estimating the spread between 7.5 and 10. Thus 10 hp (next standard size motor) would be required, and this would satisfy the original condition and the second condition for water. It would still be satisfactory for any fluid with a specific gravity <1.0, but if pumping a liquid of 1.28 sp gr (e.g., ethyl chloride), then (1) the original BHP would need to be  $1.28(5.75) = 7.36$  BHP, and (2) the second condition would require  $1.28(9.25) = 11.84$  BHP (calculates 11.56). Whereas, a 10-hp motor would be non-overloading for the water pumping case, it would require a 15-hp (next standard above a 10 hp) motor direct drive to satisfy the ethyl chloride case under the 160-gpm condition. If you do not select a non-overloading motor, and variations in head and/or flow occur, the motor could overheat and stop operating. Study the pump-capacity curve shape to recognize the possible variations.

*Important note:* Any specific pump impeller operating in a physical (mechanical) system will only perform along its operating characteristic curve. If there is a change in the system flow characteristics (rate or friction resistance or pressure head), the performance will be defined by the new conditions and the pump performance will "slide" along its fixed curve. Thus, the designer cannot arbitrarily pick a point and expect the pump to "jump" to that point. Refer to Figure 5-18a. Using a 6-in. impeller curve, for example, the designer cannot make this pump operate at a point of 100 gpm and 150 ft head. This would require about a 6 $\frac{1}{4}$ -in. diameter impeller. The 6-in. curve will only put out 138 ft (approx.) at the intersection of 100 gpm and the 6-in. curve.

A driver selected to just handle the power requirements of the design point (other than maximum) is usually a poor approach to economy. Of course, there are applications where the control system takes care of the possibilities of power overload.

## 5.14 AFFINITY LAWS

The affinity laws relate the performance of a known pump along its characteristic curve to a new performance curve when the speed is changed. This would represent the same "family" of pump curves. As an example, see Figures 5-18a–c.

1. For change in speed with a geometrically similar family of fixed impeller design, diameter, and efficiency, the

following conditions and characteristics change *simultaneously* [10]:

$$Q_2 = Q_1 \left( \frac{n_2}{n_1} \right) \quad (5-29)$$

$$H_2 = H_1 \left( \frac{n_2}{n_1} \right)^2 \quad (5-30)$$

$$\text{BHP}_2 = \text{BHP}_1 \left( \frac{n_2}{n_1} \right)^3 \quad (5-31)$$

For a fixed speed [10],

$$Q_2 = Q_1 \left( \frac{d_2}{d_1} \right) \quad (5-32)$$

$$H_2 = H_1 \left( \frac{d_2}{d_1} \right)^2 \quad (5-33)$$

$$\text{BHP}_2 = \text{BHP}_1 \left( \frac{d_2}{d_1} \right)^3 \quad (5-34)$$

For geometrically similar impellers operating at the same specific speed, the affinity laws are as follows [10, 14].

$$\frac{Q_2}{Q_1} = \left( \frac{n_2}{n_1} \right) \left( \frac{d_2}{d_1} \right)^3 \quad (5-35)$$

$$\frac{H_2}{H_1} = \left( \frac{n_2}{n_1} \right)^2 \left( \frac{d_2}{d_1} \right)^2 \quad (5-36)$$

$$\frac{\text{BHP}_2}{\text{BHP}_1} = \left( \frac{n_2}{n_1} \right)^3 \left( \frac{d_2}{d_1} \right)^5 \quad (5-37)$$

where subscript (1) represents the condition for which a set of conditions are known and subscript (2) represents the new non-cavitating or desired condition.

These relations do not hold exactly if the ratio of speed change is greater than 1.5–2.0, nor do they hold if suction conditions become limiting, such as NPSH.

Figure 5-37 illustrates the application of these performance laws to the 1750 rpm curves (capacity, BHP, and efficiency) of a particular pump to arrive at the 1450 and 1150 rpm curves. Note that the key value is the constant efficiency of points (1) and (2). When the speed drops to 1450 rpm, capacity drops:

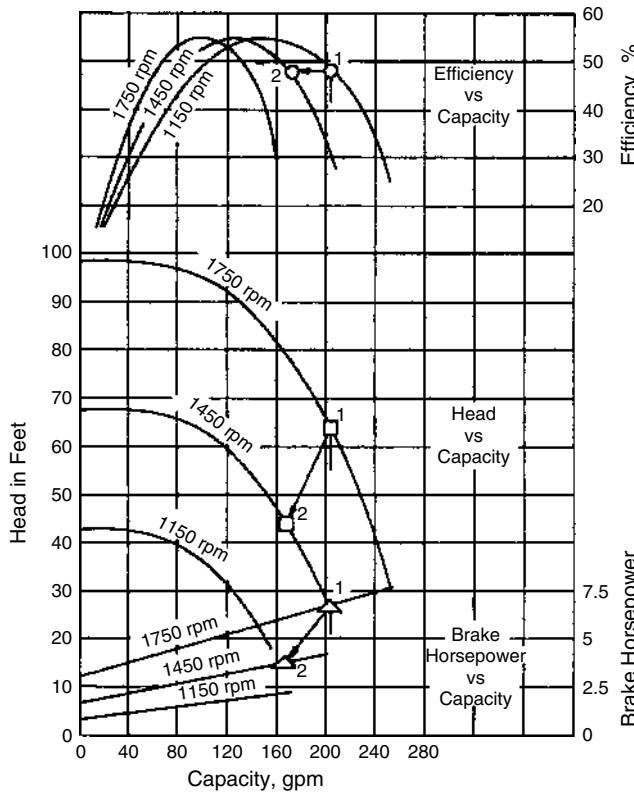
$$Q_2 = 204 \left( \frac{1450}{1750} \right) = 169 \text{ gpm}$$

The head also drops:

$$H_2 = 64 \left( \frac{1450}{1750} \right)^2 = 44 \text{ ft}$$

and

$$\text{BHP}_2 = 6.75 \left( \frac{1450}{1750} \right)^3 = 3.84 \text{ BHP}$$



**Figure 5-37** Relation of speed change to pump characteristics.

where

- d = diameter of impeller (ft, m)
- n = rotational speed (rpm, 1/s)
- H = output head (ft, m)
- Q = volumetric discharge (gpm, m<sup>3</sup>/s)
- BHP = brake horsepower (bhp, kw)

2. For changes (*cut-down*) in impeller diameter (*not design*) at fixed efficiency [14]

$$Q_2 = Q_1 \left( \frac{n_2}{n_1} \right) \left( \frac{d_2}{d_1} \right)^3 \quad (5-38)$$

$$H_2 = H_1 \left( \frac{n_2}{n_1} \right)^2 \left( \frac{d_2}{d_1} \right)^2 \quad (5-39)$$

$$\text{BHP}_2 = \text{BHP}_1 \left( \frac{n_2}{n_1} \right)^3 \left( \frac{d_2}{d_1} \right)^3 \quad (5-40)$$

where  $d_1$  is the original impeller diameter in inches, subscript (1) represents the condition for which a set of conditions are known and subscript (2) designates the new or desired conditions corresponding to the new impeller diameter  $d_2$ . All performance changes occur simultaneously when converting from condition (1) to condition (2), and no single condition can be true unless related to its corresponding other conditions.

An impeller can be cut from one size down to another on a lathe, and provided the change in diameter is not greater than 20%, the conditions of new operation can be described by the type of calculations above. A cut to reach 75–80% of the original diameter may adversely affect performance by greatly lowering the efficiency [12].

Most standard pump curves illustrate the effect of changing impeller diameters on characteristic performance (Figure 5-18a). Note change as reflected in the different impeller diameters. However, the slight change in efficiency is not recorded over the allowable range of impeller change.

Recognizing the flexibility of the affinity laws, it is better to select an original pump impeller diameter that is somewhat larger than required for the range of anticipated performance, and then cut this diameter down after in-service tests to a slightly smaller diameter. This new performance can be predicted in advance. Once the impeller diameter is too small, it cannot be enlarged. The only solution is to order the required large impeller from the manufacturer.

### EXAMPLE 5-16 Pump Parameters

For a centrifugal pump handling water, its manufacturer supplied the following data at 1150 rpm:

$$\text{Pump output} = 60 \text{ m}^3/\text{h}$$

$$\text{Head developed} = 50 \text{ m}$$

$$\text{Power requirement} = 12 \text{ kW}$$

Assuming that density of water is 1000 kg/m<sup>3</sup>, determine the pump efficiency, head, output, and power consumed at  $N = 1400$  rpm taking into account that efficiency remains constant.

*Solution*

Using Eq. (5-21), the pump hydraulic power output is

$$P_Q = \frac{(1000)(60)(50)}{367 \times 10^3} = 8.17 \text{ kW}$$

Pump efficiency  $e$  (fraction) is:

$$e = \frac{\text{Power supplied}}{\text{Power required}} = \frac{P_Q}{P} = \frac{(\rho)(Q)(H)}{\rho \times 367 \times 10^3} = \frac{8.17}{12} = 0.68 \quad (5-41)$$

Head, output, and power:

Head,  $h_2$ :

$$\frac{h_2}{h_1} = \left( \frac{n_2}{n_1} \right)^2$$

$$h_2 = h_1 \left( \frac{n_2}{n_1} \right)^2 = 50 \left( \frac{1400}{1150} \right)^2 = 74.1 \text{ m}$$

Output,  $Q_2$ :

$$\frac{Q_2}{Q_1} = \left( \frac{n_2}{n_1} \right)$$

$$Q_2 = Q_1 \left( \frac{n_2}{n_1} \right) = 60 \left( \frac{1400}{1150} \right) = 73.04 \text{ m}^3/\text{h}$$

Power consumed,  $\text{BHP}_2$ :

$$\frac{\text{BHP}_2}{\text{BHP}_1} = \left( \frac{n_2}{n_1} \right)^3$$

$$\text{BHP}_2 = \text{BHP}_1 \left( \frac{1400}{1150} \right)^3 = 12 \left( \frac{1400}{1150} \right)^3 = 21.65 \text{ kW}$$

**EXAMPLE 5-17****Specific Speed, Flowrate, and Power Required by a Pump**

A centrifugal pump is required to deliver 1600 tonne/h of water against a head of 30 m while operating at its maximum operating efficiency. Determine its specific speed if its driven at 20 Hz. If the same pump operating at maximum efficiency under these conditions delivers at a head of 50 m, what should be its speed and its rate of discharge? Determine the power requirement of the pump if its overall efficiency is 75% while operating under the same operating conditions.

*Solution*

$$\text{Density of water} = 1000 \text{ kg/m}^3$$

$$1 \text{ tonne} = 1000 \text{ kg}$$

Specific speed by using Eq. (5-11a):

$$\text{Mass rate, } G = 1.6 \times 10^6 \text{ kg/h}$$

$$\text{Volume flowrate, } Q = \frac{\text{Mass}}{\text{Density}}$$

$$Q = \frac{1.6 \times 10^6}{(10^3 \times 3600)} = 0.44 \text{ m}^3/\text{s}$$

$$N_s = \left[ \frac{20\sqrt{0.44}}{(9.81 \times 30)^{0.75}} \right] = 0.187 \text{ Hz}$$

Speed:

$$\frac{h_2}{h_1} = \left( \frac{n_2}{n_1} \right)^2$$

or

$$\frac{n_2}{n_1} = \left( \frac{h_2}{h_1} \right)^{0.5}$$

$$n_2 = n_1 \left( \frac{h_2}{h_1} \right)^{0.5} = 20 \left( \frac{50}{30} \right)^{0.5} = 25.82 \text{ Hz}$$

Rate of discharge,  $Q$ :

Also  $Q \propto n$

$$\frac{Q_2}{Q_1} = \frac{n_2}{n_1}$$

$$Q_2 = Q_1 \left( \frac{n_2}{n_1} \right) = 0.44 \left( \frac{25.82}{20.0} \right)$$

$$= 0.568 \text{ m}^3/\text{s} (2044.8 \text{ m}^3/\text{h})$$

Power required  $P$  is calculated using Eq. (5-23)

$$P_Q = \frac{(1000)(2044.8)(50.0)}{(0.75 \times 367 \times 10^3)} = 371.4 \text{ kW}$$

**EXAMPLE 5-18****Pump Sizing of Gas-Oil**

A centrifugal pump having a 4-in. nozzle and a 3-in. discharge nozzle will handle gas-oil at a normal flow rate of 250 gpm through a piping and component system as described below. What is pump power requirement?

Specific gravity and density are as follows.

$$\text{Specific gravity, } S_{60} = 1.18$$

$$\text{Density of gas-oil, } \rho_{60} = 73.6 \text{ lb/ft}^3$$

$$\text{Vapor pressure at operating temperature} = 4.0 \text{ psia}$$

$$\text{At flowing conditions, temperature } T = 555^\circ \text{ F}$$

$$\text{Specific gravity, sp gr} = 1.04$$

$$\text{Density of gas-oil, } \rho = 64.87 \text{ lb/ft}^3$$

$$\text{Viscosity of gas-oil} = 0.6 \text{ cP}$$

The data for the suction and discharge sides are given below.

Suction Condition		Discharge Condition	
Pressure at Equipment, psig	13	Pressure at equipment, psig	13.5
Static head, psi	5.403	Static head, psi	20.94
Line loss $\Delta P$ , psi	0.22	Line loss $\Delta P$ , psi	6.47
Filter/Strainer $\Delta P$ psi	0	Exchanger $\Delta P$ , psi	5.2
Valves, $\Delta P$ psi		Furnace $\Delta P$ , psi	0
		Orifice $\Delta P$ , psi	1.52

*Solution*

The Excel spreadsheet program from the companion website (Example 5-18.xls) is developed to determine the estimated absolute power requirement for the sizing the gas-oil centrifugal pump. Figure 5-18d shows the spreadsheet results of the pump sizing calculation for Example 5-18, with an estimated pump efficiency of 70%. The power requirement is 10.1 hp.

**EXAMPLE 5-19****Debutanizer Unit of Example 4-2**

What is the required power of pump P1017A for the controlled reflux rate of 1235.52 tonnes/day ( $106.56 \text{ m}^3/\text{h}$ ) (see Figure 4-23)?

Density of condensate,  $\rho = 488 \text{ kg/m}^3$

Vapor pressure at operating temperature = 14.17 bara

At flowing conditions, temperature  $T = 57^\circ \text{C}$

Specific gravity, sp gr = 0.488

Viscosity of condensate = 0.112 cP

Suction Condition	Discharge Condition
Pressure at Equipment, barg	13.9
Static head, bar	0.335
Line loss $\Delta P$ , bar	0.0128
Filter/Strainer $\Delta P$ , bar	0
Pressure at equipment, barg	13.7
Static head, bar	2.057
Line loss $\Delta P$ , bar	0.447
Exchanger $\Delta P$ , bar	
Furnace $\Delta P$ , bar	
Orifice $\Delta P$ , bar	

The data for the suction and discharge sides are given below.

*Solution*

The Excel spreadsheet program from the companion website (Example 5-18.xls) is developed to determine the estimated absolute power requirement for sizing the reflux LPG centrifugal pump P1017A/B. Figure 5-18e shows the spreadsheet results of the pump sizing calculation for Example 5-19, with an estimated pump efficiency of 70%. The absolute pump power is 8.4 kW.

## 5.15 CENTRIFUGAL PUMP EFFICIENCY

The design engineer must use the expected pump efficiency provided in the pump performance curve to evaluate the required BHP for a centrifugal pump. In the early stages of the design, it is customary to estimate a value for the efficiency. Final values depend on the specified pump, and on the operating conditions that are encountered. Branan [19] has developed an equation to calculate the centrifugal pump efficiency and pump horsepower. The equation was based on pump efficiency curves of the Natural gas transmitting pipeline safety act (NGPSA) Engineering Data Book. These efficiency curves give good results with the vendor data (the range of developed heads, 50–300 ft, and the flow rates, 100–1000 gpm). The equation for calculating the pump efficiency is expressed as follows:

$$\begin{aligned} \text{EFF} &= (80 - 0.2855H + 3.78 \times 10^{-4}HQ - 2.38 \times 10^{-7}HQ^2 \\ &\quad + 5.39 \times 10^{-4}H^2 - 6.39 \times 10^{-7}H^2Q \\ &\quad + 4 \times 10^{-10}H^2Q^2) \left( \frac{1}{100} \right) \end{aligned} \quad (5-42)$$

The actual BHP required for pump operation is

$$\text{BHP} = \frac{\text{HBHP}}{\text{EFF}} \quad (5-43)$$

where

EFF = Pump efficiency (decimal fraction)

H = Developed pump head, ft

Q = Flow rate through pump, gpm

HBHP = Hydraulic brake horsepower, hp

BHP = Actual brake horsepower required for pump operation, hp.

Using Eq. (5-42), Figure 5-18f shows typical characteristic curves for a 6-in. centrifugal pump operating at 1750 rpm. Equation (5-42) gives results within 7% of the actual pump curves for the range of applicability ( $H = 50\text{--}300 \text{ ft}$ ,  $Q = 100\text{--}1000 \text{ gpm}$ ). For flows in the range of 25–99 gpm, an approximate efficiency can be obtained by solving Eq. (5-43) for 100 gpm and then subtracting 0.35% gpm

times the difference between 100 gpm and the low flow rate in gpm. For low flow rates, near 25–30 gpm, this will give results within about 15% for the middle of the head range (pump  $\Delta P$ ) and 25% at the extremes. The horsepower at the 25–30 gpm level is normally below 10 [19]. The Excel spreadsheet program from the companion website (pump-efficiency.xls) has been developed to determine the pump efficiency, hydraulic BHP and actual BHP for pumps at varying flow rate and developed head. Figures 5-18f and g respectively show the plots for a 6-in. impeller at 1750 rpm and an 8-in. impeller at 1450 rpm.

**EXAMPLE 5-20****Reducing Impeller Diameter at Fixed RPM**

If you have a non-cavitating (sufficient NPSH) operating 9-in. impeller producing 125 gpm at 85 ft total head pumping kerosene of sp gr = 0.8 at 1750 rpm using 6.2 BHP (not motor nameplate), what diameter impeller should be used to make a permanent change to 85 gpm at 60 ft head, at the same speed?

By using Eq. (5-32)

$$85 = 125 \left( \frac{d_2}{9} \right)^2 \quad (5-32)$$

$d_2$  = 6.1-in. diameter (new)

The expected head calculated using Eq. (5-33) is

$$H_2 = 85 \left( \frac{6.1}{9} \right)^2 \quad (5-33)$$

= 39.0 ft (must check system new total head to determine if it will satisfy this condition)

The expected brake horsepower BHP calculated using Eq. (5-34) is

$$\text{BHP}_2 = 6.2 \left( \frac{6.1}{9} \right)^3 \quad (5-34)$$

= 1.93 BHP (use a 2- or 3-hp motor)

### 5.16 EFFECTS OF VISCOSITY

When viscous liquids are handled in centrifugal pumps, the BHP is increased, the head is reduced, and the capacity is reduced as compared to the performance with water. The corrections may be negligible for viscosities in the same order of magnitude as water, but become significant above 10 cSt (10 cP for sp gr = 1.0) for heavy materials. While the calculation methods are acceptably good, for exact performance charts test must be run using the pump in the service.

When the performance of a pump handling water is known, the following relations are used to determine the performance with viscous liquids [8]:

$$Q_{\text{vis}} = C_Q (Q_w) \quad (5-44)$$

$$H_{\text{vis}} = C_H (H_w) \quad (5-45)$$

$$E_{\text{vis}} = C_E (E_w) \quad (5-46)$$

$$\text{BHP}_{\text{vis}} = \frac{(Q_{\text{vis}})(H_{\text{vis}})(\text{sp gr})}{3960(E_{\text{vis}})} \quad (5-47)$$

Determine the correction factors from Figures 5-38 and 5-39, which are based on water performance because this is the basis of most manufacturer's performance curves (except that the "standard" manufacturer's performance curves of head vs gpm reflect the head of any fluid, water, or other non-viscous liquids). Do not extrapolate these curves!

Referring to Figure 5-38 [8]:

1. The values are averaged from tests of conventional single-stage pumps, 2–8 in., with capacity at best efficiency point of less than 100 gpm on water performance.
2. Tests use petroleum oils.
3. The values are not exact for any specific pump.

Referring to Figure 5-39 [8]:

1. Tests were on smaller pumps, 1 in. and below.
2. The values are not exact for any specific pump.

The charts are to be used on Newtonian liquids, but not for gels, slurries, paperstock, or any other non-uniform liquids [8].

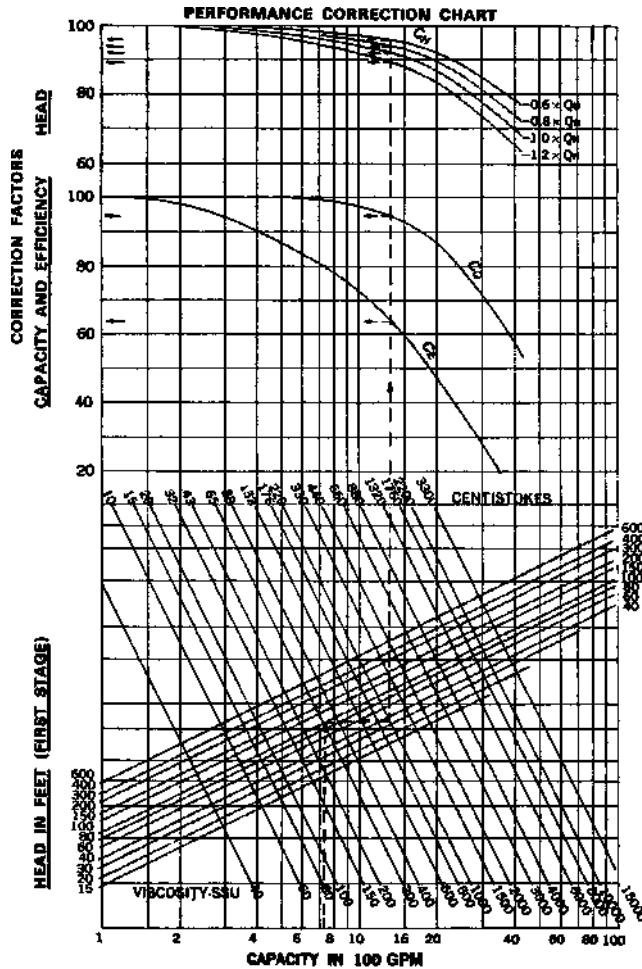
Figures 5-38 and 5-39 are used to correct the performance to a basis consistent with the conditions of the usual pump curves. In order to use the curves, the following conversions are handy:

$$\text{Centistokes (cSt)} = \text{centipoise (cP)/sp gr}$$

$$\begin{aligned} \text{SSU} &= \text{Saybolt Seconds Universal} \\ &= (\text{cSt}) (4.620) \text{ at } 100^\circ \text{F} \\ &= (\text{cSt}) (4.629) \text{ at } 130^\circ \text{F} \\ &= (\text{cSt}) (4.652) \text{ at } 210^\circ \text{F} \end{aligned}$$

### TEMPERATURE RISE AND MINIMUM FLOW

When a pump operates near shutoff (low flow) capacity and head, or is handling a hot material at suction, it may become overheated and create serious suction as well as mechanical problems. To avoid overheating due to low flow, a minimum rate (gpm) should be recognized as necessary for proper heat dissipation. However, it is not necessarily impossible to operate at near shutoff conditions,



**Figure 5-38** Viscosity performance correction chart for centrifugal pumps. Note: Do not extrapolate. For centrifugal pumps only, not for axial or mixed flow. NPSH must be adequate. For Newtonian fluids only. For multistage pumps, use head per stage. (By permission from *Hydraulic Institute Standards for Centrifugal, Rotary, and Reciprocating Pumps*, 13th ed., Hydraulic Institute, 1975.)

provided (1) it does not operate long under these conditions, as temperature rises per minute vary from less than 1° F to 30–40° F, or (2) a bypass is routed or recycled from the discharge through a cooling arrangement and back to suction to artificially keep a minimum safe flow through the pump while actually withdrawing a quantity below the minimum, yet keeping the flowing temperature down [20].

1. Temperature rise in average pump during operation [21]:

$$\Delta T_r = \frac{42.4 P_{\text{so}}}{W_1 c_p}, ^\circ \text{F/min} \quad (5-48)$$

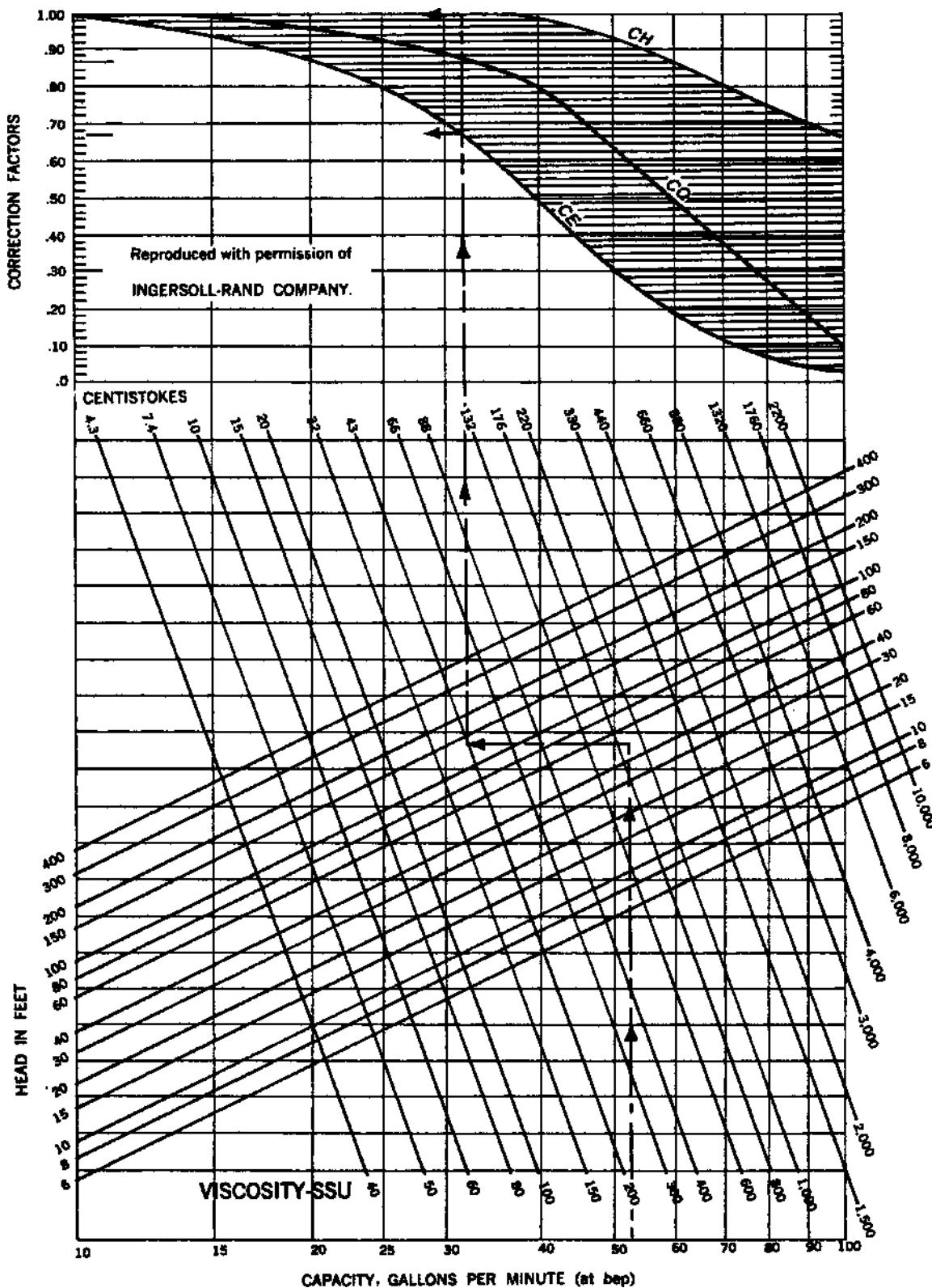
where [10]

$$\Delta T_r = \text{temperature rise, } ^\circ \text{F/min}$$

$$P_{\text{so}} = \text{brake horsepower at shutoff or no flow}$$

$$W_1 = \text{weight of liquid in pump, lb}$$

$$c_p = \text{specific heat of liquid in pump, or alternate procedure [21, 22].}$$



**Figure 5-39** Viscosity performance correction chart for small centrifugal pumps with capacity at best efficiency point of less than 100 gpm (water performance). *Note:* Do not extrapolate. For small centrifugal pumps only, not for axial or mixed flow. NPSH must be adequate. For Newtonian fluids only. For multistage pumps, used head per stage. (By permission from *Hydraulic Institute Standards for Centrifugal, Rotary, and Reciprocating Pumps*, 13th ed., Hydraulic Institute, 1975.)

**EXAMPLE 5-21****Pump Performance Correction for Viscous Liquid**

When the required capacity and head are specified for a viscous liquid, the equivalent capacity when pumping water needs to be determined using Figure 5-38 or 5-39 in order to rate pump selection from manufacturer's curves.

Determine proper pump selection and specifications when pumping oil with sp gr of 0.9 and viscosity of 25 cP at the pumping temperature, if the pump must deliver 125 gpm at 86 ft total head (calculated using the viscous liquid).

Viscosity conversion:

$$\text{cSt} = 25/0.9 = 27.8$$

Referring to Figure 5-30:

- Enter capacity at 125 gpm, follow vertically to 86 ft of head, then to right to viscosity of 27.8 cSt, and up to correction factors:

Efficiency,  $C_E = 0.80$

Capacity,  $C_Q = 0.99$

Head,  $C_H = 0.96$  (for 1.0  $Q_N$ )

$Q_N$  = head at best efficiency point

Note: This represents a flow rate using water under maximum efficiency conditions [8].

- Calculate approximate water capacity:

$$Q_w = \frac{Q_{\text{vis}}}{C_Q} = \frac{125}{0.99} = 126.3 \quad (5-44)$$

- Calculate approximate water head:

$$H_w = \frac{H_{\text{vis}}}{C_H} = \frac{86}{0.96} = 89.6 \text{ ft} \quad (5-45)$$

- A pump may now be selected using water as the equivalent fluid with capacity of 126.3 gpm and head of 89.6 ft. The selection should be made at or very near to the point (or region) of peak performance as shown on the manufacturer's curves.
- The pump described by the curves of Figures 5-18a and b fit these requirements. The peak efficiency is 71% using water.
- Calculate the viscous fluid pumping efficiency:

$$e_{\text{vis}} = C_e(e_w) = (0.80)(71) = 56.8\% \quad (5-46)$$

- Calculate BHP for viscous liquid

$$\begin{aligned} \text{BHP}_{\text{vis}} &= \frac{Q_{\text{vis}} H_{\text{vis}} (\text{sp gr})}{3960 (e_{\text{vis}})} \\ &= \frac{(125)(86)(0.9)}{3960(0.568)} = 4.3 \text{ BHP} \end{aligned} \quad (5-47)$$

**EXAMPLE 5-22****Corrected Performance Curves for Viscosity Effect**

When a pump performance is defined for water, the corrected performance for a viscous fluid can be developed using Figure 5-38 or 5-39. In order to develop the curves for viscosity conditions of 100 SSU or 1000 SSU as shown in Figure 5-40, the following general procedure is used [8].

- Starting with performance curve based on pumping water:
  - Read the water capacity and head at peak efficiency. This capacity is the value of 1.0  $Q_{\text{nw}}$ .
  - Using this value of gpm, calculate 0.6, 0.8, and 1.2 times this value, giving 0.6, 0.8, and 1.2  $Q_{\text{nw}}$  respectively, and read the corresponding heads and water efficiencies.
- Using Figure 5-38 or 5-39 enter gpm at value corresponding to peak efficiency, 1.0  $Q_{\text{nw}}$ , and follow up to the corresponding head value  $H_w$ , then move to the viscosity value of the liquid, and up to the correction factors  $C_E$ ,  $C_Q$ ,  $C_H$ .

- Repeat step (2) using gpm and head values of step (lb).

- Correct head values:

$$H_{\text{vis}} = H_w C_H \quad (5-45)$$

- Correct efficiency values:

$$e_{\text{vis}} = e_w C_E \quad (5-46)$$

- Correct capacity values:

$$Q_{\text{vis}} = Q_w C_Q \quad (5-44)$$

- Calculate the viscous BHP as indicated in the previous example.

- Plot values as generally indicated on Figure 5-40 and obtain the performance curves corresponding to the viscous liquid conditions.

For low capacity,

$$\Delta T_r = \frac{H_{\text{so}} (1 - e)}{778 (c_p) (e)} \quad (5-49)$$

where

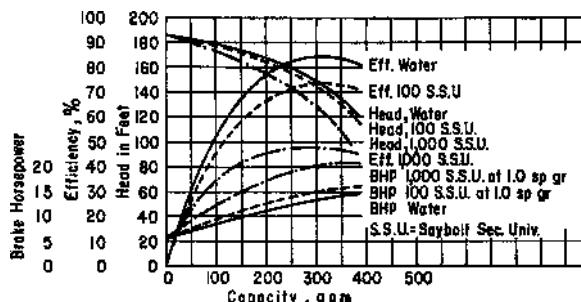
$H_{\text{so}}$  = total head of pump at no flow or shutoff or at any flow rate with corresponding efficiency from pump curve, ft

$e$  = pump efficiency at the flow capacity involved (low flow), decimal.

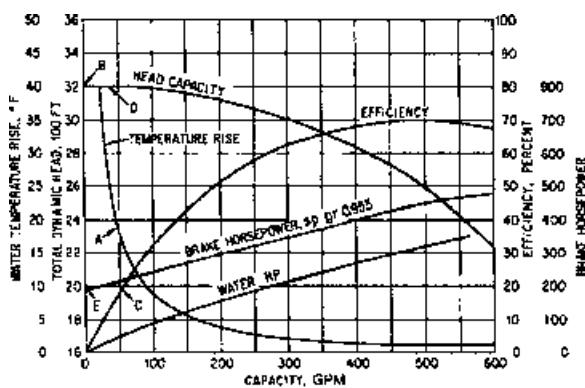
Another alternate procedure [16] is as follows.

$$\Delta T_r = \frac{(\text{GPM})(H_{\text{so}})(\text{sp gr})}{3960} \quad (5-50)$$

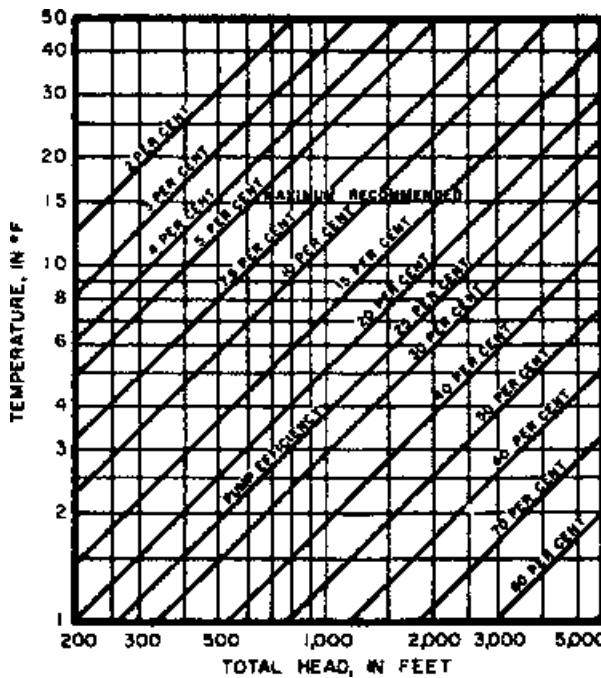
See Figures 5-41 and 5-42 for a graphical solution to the equation above for temperature rise. Figure 5-41 illustrates



**Figure 5-40** Typical curves showing the effect on a pump designed for water when pumping viscous fluids. (By permission from Pic-a-Pump, 1959, Allis-Chalmers Mfg Co.)



**Figure 5-41** Typical temperature rise for boiler feed water pump. (By permission from *Transamerica Delaval engineering Handbook*, 4th ed., H.J. Welch, ed., 1983. Transamerica Deleval, Inc., IMO Industries, Inc. Div.)



**Figure 5-42** Temperature rise in centrifugal pumps in terms of total head and pump efficiency. (By permission from Karrasik, I. and R. Carter, *Centrifugal Pumps*, McGraw-Hill Book Co., Inc., 1960, p. 438.)

the characteristics of a boiler feed water pump set to handle 500 gpm water at 220°F for a total of 2600 ft head. The temperature rise curve has been superimposed on the performance chart for the pump, and values of  $\Delta T_r$  are calculated for each flow-head relationship. Note how rapidly the temperature rises at the lower flows. This heating of the fluid at low flow or no flow (discharge valve shut, no liquid flowing through the pump) can be quite rapid and can cause major mechanical problems in the pump's mechanical components. The maximum temperature rise recommended for any fluid is 15°F (can be a bit higher at times for the average process condition) except when handling cold fluids or using a special pump designed to handle hot fluid, such as a boiler feed water pump of several manufacturers.

$$\Delta T_r = \frac{(BHP \text{ at shutoff}) (42.4)}{(\text{weight of liquid in pump}) (c_p)}, ^\circ\text{F}/\text{min} \quad (5-51)$$

or

$$\Delta T_r = \frac{(BHP - WHP) (2545)}{(\text{pump capacity})}, ^\circ\text{F}/\text{min} \quad (5-52)$$

where

BHP = brake horsepower

WHP = Water or liquid horsepower.

## 2. Minimum Flow (Estimate) [21]:

The validity of the method has not been completely established, although it has been used rather widely in setting approximate values for proper operation [16]. For multistage pumps use only the head per stage in temperature limit by this method.

- Determine  $NPSH_A$  at pump suction.
- Add the NPSH value to the vapor pressure of the liquid at suction conditions. This represents the vapor pressure corresponding to the temperature of the liquid at the flash point. Read temperature  $t_2$  from vapor pressure chart of liquid.
- Allowable temperature rise =  $t_2 - (\text{actual pumping temperature})$ . Boiler feed water practice uses 15°F rise for average conditions [16].
- Approximate minimum safe continuous flow efficiency:

$$e_M = \frac{H_{so}, \text{ at shutoff from curve}}{778 (\Delta T_r) c_p + H_{so}, \text{ at shutoff}} \quad (5-53)$$

where

$e_M$  = minimum safe flowing efficiency, overall pump, fraction

$H_{so}$  = head at no flow or shutoff, ft

$c_p$  = specific heat of liquid, Btu/lb/°F

$\Delta T_r$  = temperature rise in liquid, °F

- Read minimum safe flow in gpm from pump performance curve at value of minimum efficiency calculated in (d).

**EXAMPLE 5-23****Maximum Temperature Rise Using Boiler Feed Water**

Using the example of [21], assume a pump with characteristic curve and added temperature rise data as shown on Figure 5-41 is to handle boiler feed water at 220°F, with a system  $NPSH_A = 18.8$  ft. The vapor pressure of water at 220°F is 17.19 psia from steam tables and the sp gr = 0.957. Correcting the 18.8 ft  $NPSH_A = 18.8 [1/(2.31/0.957)] = 7.79$  psia at 220°F.

The vapor pressure to which the water may rise before it flashes is  $17.19 + 7.79 = 24.98$  psia.

From steam tables (or fluid vapor pressure tables), read at 24.98 psia (for water in this example), temperature = 240°F.

Therefore, allowable temperature rise of the water (in this example) =  $240 - 220 = 20^\circ$ F.

A plotted curve as shown on Figure 5-41 [22] shows that at point A, a rise of 20°F on the temperature rise curve corresponds to a minimum of 47 gpm safe for the pump at 220°F, with  $NPSH_A$  of 18.8 ft.

An alternate estimate for minimum flow [14]: Minimum flow (for water) through pump,

$$Q_M = 0.3P_{so}, \text{ gpm} \quad (5-54)$$

where

$$P_{so} = \text{shutoff horsepower.}$$

For cold liquids, general service can often handle  $\Delta T_r$  of up to 100°F, a rule with approximately 20% factor of safety:

$$Q_M = \frac{6P_{so}}{\Delta T_r} \text{ gpm} \quad (5-55)$$

where

$$\Delta T_r = \text{permissible temperature rise, } ^\circ\text{F.}$$

The  $NPSH_R$  at the higher temperature may become the controlling factor in order to avoid cavitation.

The minimum flow simply means that this flow must circulate through the pump casing (not recirculation with no cooling) back to at least the initial temperature of the feed, if excessive temperatures are not to develop. The best practice is to request the manufacturer to state this value for the fluid handled and the calculated  $NPSH_A$  condition. For  $NPSH_R$  refer to corrections discussed earlier.

## 5.17 CENTRIFUGAL PUMP SPECIFICATIONS

Figures 5-43a and b present specifications and calculation for a centrifugal pump respectively. Although the process engineer cannot or should not specify each item indicated, he must give the pertinent data to allow the pump manufacturer to select a pump and then identify its features. Pumps are selected for performance from the specific characteristic curves covering the casing size and impeller style and diameter. Often the process fluids are not well known to the pump manufacturer; therefore the materials of construction, or at least any limitations as to composition, must be specified by the engineer.

**EXAMPLE 5-24****Pump Specifications (Figures 5-43a and b)**

The pump specified identifies the design data, key portions of the construction materials, and driver data as required information for the pump manufacturer. If the pump is to be inquired to several manufacturers this is all that is necessary. The individual manufacturers will identify their particular pump selection, details of construction materials, and driver data. From this information a pump can be selected with performance, materials of construction, and driver requirements specified.

In the example (Figure 5-43a), the manufacturer has been specified from available performance curves, and the details of construction must be obtained. The pump is selected to operate at 22 gpm and 196–200 ft head of fluid, and must also perform at good efficiency at 18 gpm and a head which has not been calculated, but which will be close to 196–200 ft, say about 185 ft. Ordinarily, the pump is rated as shown on the specification sheet. This insures adequate capacity and head at conditions somewhat in excess of normal. In this case, the design gpm was determined by adding

## NUMBER OF PUMPING UNITS

A single pump is the cheapest first-cost installation. However, if downtime has any value such as in lost production, in hazards created in the rest of the process, and so on, then a standby duplicate unit should be considered. A spare or standby can be installed adjacent to the operating unit, and switched into service on very short notice, provided it is properly maintained. Spare pumps which do not operate often should be placed in service on a regular schedule just to be certain they are in working order.

If solids are carried in the fluid, this can present a difficult problem if they are not properly flushed from the pump on

10% to the capacity and allowing for operation at 90% of the rated efficiency. Often this latter condition is not considered, although factors of safety of 20% are not unusual. However, the efficiency must be noted and the increase in horsepower recognized as factors which are mounted onto normal operating conditions.

Sometimes the speed of the pump is specified by the purchaser. However, this should not be done unless there is experience to indicate the value of this, such as packing life, corrosion/erosion at high speeds, and suspended particles, as the limitation on speed may prevent the manufacturer from selecting a smaller pump. In some cases it must be recognized that high heads cannot be reached at low speeds in single-stage pumps. Table 5-7 presents suggestions for materials of construction for pump parts in the services indicated. The effect of impurities, temperature, analysis variations, and many other properties make it important to obtain specific corrosion service data in the specific fluid being pumped. Sometimes this is not possible, and generalized corrosion tables and experience of other users must be relied on as the best information for the selection of materials.

Job No.					SPEC. DWG. NO.
B/M No.					A- Page 1 of 1 Pages Unit Price 2 (one as spare) No. Units P-68 A & B Item No.
<b>CENTRIFUGAL PUMP SPECIFICATIONS</b>					
<b>DESIGN DATA</b>					
Service % Solids	Forwarding to T-98 None	Calc. GPM 18	Liquid Des. GPM 73% Caustic 22		
Vapor Press.	- PSIA	Temp. 265 °F	Vis. 5.9 Cp sp gr 1.68		
HEAD	SUCTION	DISCHARGE		DIFFERENTIAL	
Static	-8.0 ft	+4.1 ft		+12.1 ft	
Pressure	+20.2 ft abs	+3.4 ft abs		-16.8 ft	
Friction	-0.1 ft	+199.8 ft		+199.9 ft	
Total	+12.1 ft abs	+207.3 ft abs		+195.2 ft	
NPSH Available	9.5 ft	Rating Head	196 to 200 ft		
<b>PUMP SELECTION</b>					
Mfgr. "A"	Size & Type Centrifugal, 2×3	Model AAA	Case Type Vert. Split		
Stages one	Imp. Type Enclosed	Design Imp. Dia. 13 1/8"	Max. Imp. Dia. 14		
Drive Motor	RPM 1750	Design Eff. 73%	Rotation* Counter Clockwise		
Rating BHP	14.5 Non-Overhead BHP	15 Shut-off Press. 143 psig	NPSH Required 6 ft		
Suc. Location	End	Disch. Location Vertical Top	Curve No. C - 1234		
<b>CONSTRUCTION MATERIALS</b>					
Case Cast. 29 Cr-9 N1, Misc.	Impeller 29 Cr-9 N1, Misc.	Shaft Nickel			
Packing	Shaft Seal: Internal <input type="checkbox"/>	External <input type="checkbox"/>	Plugged <input type="checkbox"/>		
Stuffing Box	Seal Cage	Bushings	Gland Bolts		
Mechanical Seal: Make "x"	Type "Y" ; coolant Water	Inside <input type="checkbox"/> Outside <input checked="" type="checkbox"/>	Single <input checked="" type="checkbox"/> Double <input type="checkbox"/>		
Bearings Cast Iron	Coupling Mfg. "A"	Type Flexible	Guard Yes		
Bearings: Type Bell, double	Make	Coolant			
Impeller Nut Nickel	Thrust Bearings: Type Force Lubricated	Make "Z"			
Case Studs Nickel	Case End Covers Nickel				
Diffusers Nickel	Diaphragm				
Piping: Nickel					
Suction Size 3"	Rating 150# Std.	Discharge Size 2"	Rating 150# Std.		
<b>DRIVER</b>					
Type TEFC	Mfg. B	H.P. 15			
RPM 1750	Frame	Volts 440	Phase 3	Cycle 60	
Elec. Class.	Connections: Direct	Coupling	Belts	Gear	
Inlet Steam psig at	lb/h	°F	Exhaust Steam psig at	°F	
<b>REMARKS</b>					
* Facing Pump from Driver End.					
By	Chlc'd.	App.	Rev.	Rev.	Rev.
Date					
P.O. Tes _____					

Figure 5-43a Centrifugal pump specifications.

shutdown. Some spare or second pumps are selected for 100% spare; others are selected so that each of two pumps operate in parallel on 50% of the flow, with each being capable of handling 67–75% of total load if one pump should fall off the line. This then only reduces production by about 25% for a short period, and is acceptable in many situations. These pumps are usually somewhat smaller than the full size spares.

When it is necessary to plan several pumps in parallel, the pump manufacturer must be advised, and care must be taken in arranging suction piping for the pumps, otherwise each may not carry its share of the flow.

There are many flow conditions, and pumps should be selected to operate as efficiently as possible over the widest range of capacity. If the flow is expected to vary during the system

**348** PUMPING OF LIQUIDS

Job Item Name.		PUMP CALCULATION SHEET		Document No.			
				Sheet of Rev.			
				Item No. (s)			
				No. Working	Total No. off		
		UNITS	CASE I	CASE II	SKETCH OF PUMP HOOK-UP		
1							
2	Liquid Pumped						
3	Corrosion/Erosion						
4	Due To						
5	Operating Temp. ( $T$ )	$^{\circ}\text{C}$	$^{\circ}\text{F}$				
6	Specific Gravity at $T$	cP	cP				
7	Viscosity						
8	Vapor Pressure at $T$	bara	psia				
9	Normal mass Flowrate	kg/h	lb/h				
10	Normal Vol. Flowrate	$\text{m}^3/\text{h}$	gpm				
11	Min. Vol. Flowrate						
12	Design Vol. Flowrate	$\text{m}^3/\text{h}$	gpm				
13	SUCTION CONDITION						
14	Pressure at Equipment	barg	psig	+	+		
15	Static Head	bar	psi	+/-	+/-		
16	Total—Lines 14 + 15	bar	psi	++			
17	Suction Line $\Delta P$	bar	psi	-	-		
18	Filter/Strainer $\Delta P$	bar	psi	-	-		
19							
20	Total Suction Pressure	barg	psig	+	+		
21	DISCHARGE CONDITION						
22	Pressure at Equipment	barg	psig	+	+		
23	Static Head	bar	psi	+/-	+/-		
24							
25	Exchanger $\Delta P$	bar	psi	+	+		
26							
27	Furnace $\Delta P$	bar	psi	+	+		
28	Orifice $\Delta P$	bar	psi	+	+		
29	Control Valve $\Delta P$	bar	psi	+	+		
30							
31	Line $\Delta P$	bar	psi	+	+		
32							
33	Total Discharge Press.	barg	psig	+	+		
34	Differential Pressure	bar	psi				
35	Differential Head	bar	psi				
36	NPSH						
37	Total Suction Pressure	bara	psia				
38	Vapor Pressure	bara	psia				
39	NPSH—Lines 37–38	bara	psia				
40	=	m	ft				
41	Safety Margin	m	ft				
42	NPSH—Lines 40–41	m	ft				
43	Hydraulic Power	kW	Hp				
44	Estimated Efficiency	%	%				
45	Estimated Abs Power	kW	Hp				
46	Type of Pump						
47	Drive						
48							
49	Material – Casing						
50	– Impeller						
51	– Shaft						
52							
53	Sour Service	Yes/No					
54	HEAD	$m = 10.2 \times \text{bar}/\text{sp gr}$	$m = 10 \times \text{kg}/\text{cm}^2/\text{sp gr}$	$ft = 2.31 \times \text{psi}/\text{sp gr}$			
55	VOLUME	$\text{m}^3/\text{h} \times \text{sp gr} \times 1000 = \text{kg}/\text{h}$	$\text{igpm} \times \text{sp gr} \times 600 = \text{lb}/\text{h}$				
56	POWER	$\text{kW} = \text{m}^3/\text{h} \times \text{bar}/36.0$	$\text{kW} = \text{m}^3/\text{h} \times \text{kg}/\text{cm}^2/36.71$				$\text{Hp} = \text{igpm} \times \text{psi}/1427$
57		1	Date	2	Date	3	Date
58	Description						
59	Made/Revised by						
60	Checked by						
61	Approved Process						
62	Approved						

**Figure 5-43b** Centrifugal pump calculation.

**TABLE 5-7 Pump Materials of Construction**

Liquid	Casing and Wearing Rings	Impeller and Wearing Rings	Shaft	Shaft Sleeves	Type of Seal	Seal Cage	Gland	Remarks
Ammonia, Anhydrous and Aqua	Cast iron	Cast iron	Carbon steel	Carbon steel	Mechanical	—	Mall. Iron	Note: Materials of Construction shown will be revised for some jobs.
Benzene	Cast iron	Cast iron	Carbon steel	Nickel Moly. Steel	Ring packing	Cast iron	Mall. Iron	
Brine (Sodium Chloride)	Ni-Resist*	Ni-Resist*	K Monel	K Monel	Ring Packing	Ni-Resist**	Ni-Resist**	* Cast iron acceptable. ** Mall. Iron acceptable.
Butadiene	Casing; C Steel-Rings: C.I.	Impeller: C.I.-Rings C. Steel	Carbon Steel	13% Chrome Steel	Mechanical	—	Carbon Steel	
Carbon Tetrachloride	Cast Iron	Cast Iron	Carbon Steel	Carbon Steel	Mechanical	—	Mall. Iron	
Caustic, 50% (Max Temp. 200° F)	Misco C	Misco C	18-8 Stainless steel	Misco C	Ring Packing	Misco C	Carbon Steel	Misco C manufactured by Michigan Steel Casting Company. 29 Cr-9 Ni Stainless Steel or equal.
Caustic, 50% (over 200° F)	Nickel	Nickel	Nickel or 18-8 Stainless Steel	Nickel	Ring Packing	Nickel	Nickel	
Caustic, 10% (with some sodium chloride)	Cast iron	23% Cr 52% Ni Stainless steel	23% Cr 52% Ni Stainless steel	23% Cr 52% Ni Stainless steel	Ring Packing	Cast Iron	—	Specifications for 50% Caustic (Max. Temp. 200° F) also used.
Ethylene	Cast Steel	Carbon Steel	Carbon Steel	Carbon Steel	Mechanical	Cast iron	Mall. Iron	
Ethylene Dichloride	Cast Iron	Cast Iron	Steel	K Monel	Mechanical	—	K Monel	

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(continued)

**TABLE 5-7—(continued)**

Liquid	Casing and Wearing Rings	Impeller and Wearing Rings	Shaft	Shaft Sleeves	Type of Seal	Seal Cage	Gland	Remarks
Ethylene Glycol	Bronze	Bronze	18-8 stainless steel	18-8 stainless steel	Ring Packing	—	Bronze	
Hydrochloric Acid, 32%	Impregnated Carbon	Impregnated Carbon	18-8 stainless steel	Impregnated Carbon	Mechanical	—	Impregnated Carbon	
Hydrochloric Acid, 32% (Alternate) Methyl Chloride	Rubber Lined C. Iron Cast Iron	Hard Rubber	Carbon Steel	Rubber or Plastic	Ring Packing	Rubber	Rubber	
Propylene	Casing: C. Steel Rings: C.I.	Imp.: C.I. Rings C. Stl	Carbon Steel	Carbon Steel	Mechanical	—	Mall. Iron	
Sulfuric Acid, Below 55%	Hard Rubber Lined C.I.	Special Rubber	Carbon Steel	Hastelloy C	Ring Packing	Special Rubber	Special Rubber	
Sulfuric Acid, 55–95%	Cast Si-iron	Si-iron	Type 316 Stn. Stl.	Si-Iron	Ring Packing	Teflon	Si-iron	
Sulfuric Acid, Above 95%	Cast Iron	Cat Iron	Carbon Steel	13% Chrome Steel Bronze	Ring Packing	Cast iron	Mall. Iron	
Water, River	Cat Iron	Bronze	18-8 Stainless Steel	Bronze	Mechanical	Cast Iron	Mall. Iron	
Water, Sea	Casing, 1–2% Ni, Cr 3–0.5% Cast Iron Rings: Ni-Resist, 2B	Impeller: Monel Rings: S-Monel	K Monel (Aged)	K Monel or Alloy 20 SS	Ring Packing	Monel or Alloy 20 SS	Monel or Alloy 20 SS	

Note: Table Materials are for general use, specific service experience is preferred when available.

operation, the high and the low GPM (and corresponding heads) should be given to allow proper evaluation.

## FLUID CONDITIONS

The manufacturer must be told the conditions of the liquid, percent suspended solids, physical properties, corrosive nature, and maximum and minimum temperature ranges. For extremely hot liquids, special hot pumps must be used, and temperature effects taken into account.

## SYSTEM CONDITIONS

The manufacturer must know if the suction side of the pump is associated with vacuum equipment, or is to lift the liquid. This can make a difference as to the type of impeller suction opening he provides. If the system operates intermittently it should be noted. A piping diagram is often helpful in obtaining full benefit of the manufacturer's special knowledge.

## TYPE OF PUMP

If there is a preference as to horizontal or vertical split casing, it should be stated. Also the suction and discharge connections should be stated as to top or end, or special, together with the preference as to flanged (rating) or screwed. Small pumps are commonly furnished with screwed connections unless otherwise specified.

## TYPE OF DRIVER

Pumps are usually driven by electric motors, steam or gas turbine or gas (or gasoline) engines, either direct or through V-belts or gears. The pump manufacturer should know the preferred type of drive. If the manufacturer is to furnish the driver, the data on the specification sheet under Driver (see Figure 1-35b) should be completed as far as applicable. If a gas or gasoline engine is to be used, the type of fuel and its condition must be stated. Engine cooling water (if air is not used) must be specified.

## SUMP DESIGN FOR VERTICAL LIFT

The proper design of sumps for the use of vertical lift pumps or horizontal pumps taking suction from a sump is important to good suction conditions at the pump [23–25].

The arrangement and dimensions indicated in Figures 5-44 and 5-45 are satisfactory for single or multiple pump installations (for more details, refer to [8]). A few key points in sump-pump relationships for good non-vortexing operation are as follows.

1. Avoid sudden changes in direction or elevation of flow closer than five bell diameters to pump.
2. Avoid sump openings or projections in water path close to pump.
3. Have water flowing parallel to sump walls as it enters pump. Water should enter pump suction with as low turbulence as possible.
4. Water velocity in sump must be low, 1.5 ft/s is good practice.
5. Inlet channel width to each pump is considered optimum at  $2B_d$  to prevent secondary turbulence effects [24].
6. Avoid placing several pumps in one open channel removing water in series fashion. If this must be done, velocity at each pump must be kept at same value as for a single pump. The channel width at each pump would be taken from [8].

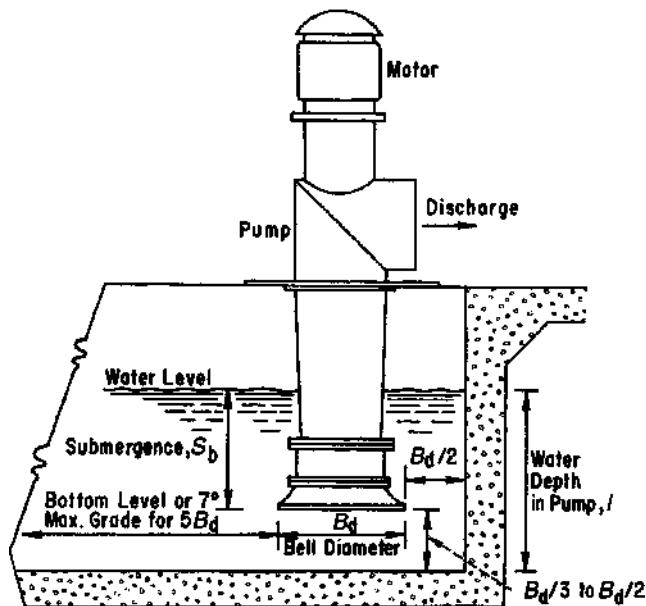
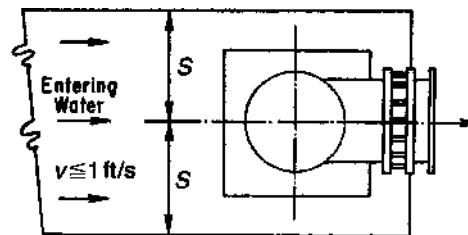


Figure 5-44 Sump design. Note:  $S = (1.5-2) B_d$ .

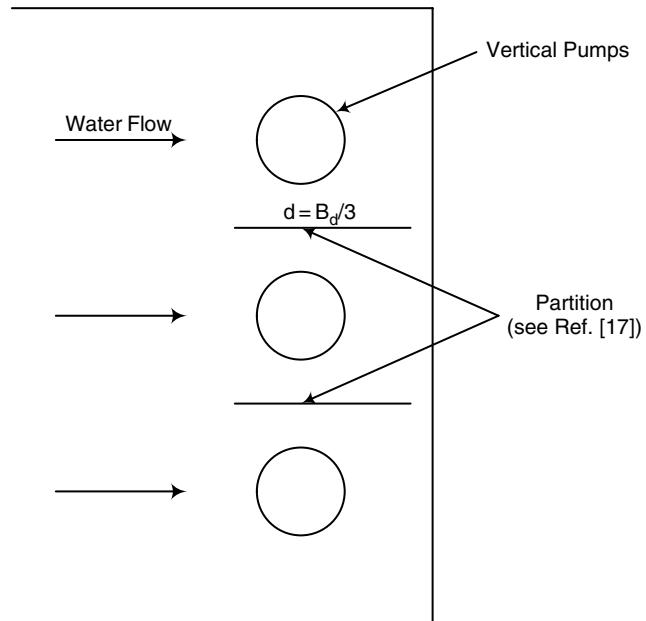


Figure 5-45 Acceptable sump arrangement for multiple pumps.

A suction bell on the inlet of a vertical pump (or the inlet pipe of the suction side of a horizontal pump) is not necessary as far as pump or sump operation is concerned. If a bell is omitted, the entrance losses due to flow will be higher with only a straight pipe, and this must be considered in pump operation. An economic comparison will help decide the value of the bell. Strainers should not be placed on suction bells unless this is the only arrangement. Inlet water should be screened with trash racks, bars, and screens to keep the sump free of debris.

Submergence of the inlet pipe column or bell inlet below the water level is necessary for good operation and to prevent vortexes and entrained air. The minimum submergence as recommended by the manufacturer must be maintained at all times. Generally, for 70° F water, each 1000 ft of elevation above sea level adds 14 in. to the required submergence. If the water is at 100° F at sea level, approximately 17 in. must be added to the 70° F submergence value [25].

### 5.18 ROTARY PUMPS

There are many different types of positive displacement rotary pumps [26] as illustrated in Figures 5-46 and 5-47a-c. The majority of these types are capable of handling only a clean solution essentially free of solids. The designs using rubber or plastic parts for the pressure device can handle some suspended particles. In general, these pumps handle materials of a wide range of viscosity (up to 500,000 SSU), and can develop quite high pressures (over 1000 psi). Additionally, the units can handle some vapor or dissolved gases mixed with the liquid being pumped. The capacity is generally low per unit, and at times, they are used for metering. For specific

performance characteristics of any type consult the appropriate manufacturer. These pumps are low in cost, require small space, and are self priming. Some can be rotated in either direction, have close clearances, require overpressure-relief protection on discharge due to positive displacement action, and have low volumetric efficiency [27].

### PERFORMANCE CHARACTERISTICS OF ROTARY PUMPS

1. Flow proportional to speed and almost independent of pressure differential.
  - a) Internal slip reduces efficiency, and increases with pressure and decreasing viscosity.
  - b) Entrained gases reduce liquid capacity and cause pulsations.
  - c) Liquid displacement [21]

$$d' = \frac{d'' (1 - E_n)}{(1 - E_n) + E_n (P/P_1)}, \text{ ft}^3/\text{min} \quad (5-56)$$

where

$P$  = the atmospheric pressure, and  $P_1$  is the inlet absolute pressure to the pump

$d''$  = theoretical displacement,  $\text{ft}^3/\text{min}$

$d'$  = liquid displacement,  $\text{ft}^3/\text{min}$

$E_n$  = percent entrained gas by volume at atmospheric pressure.

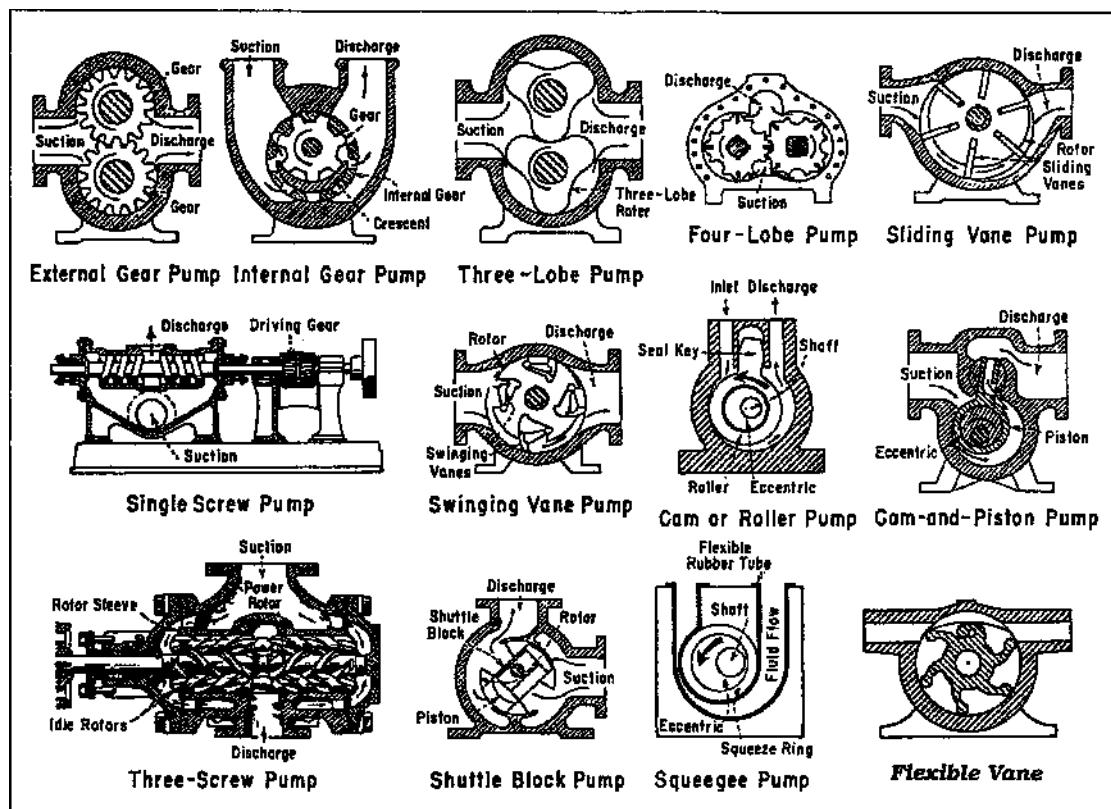
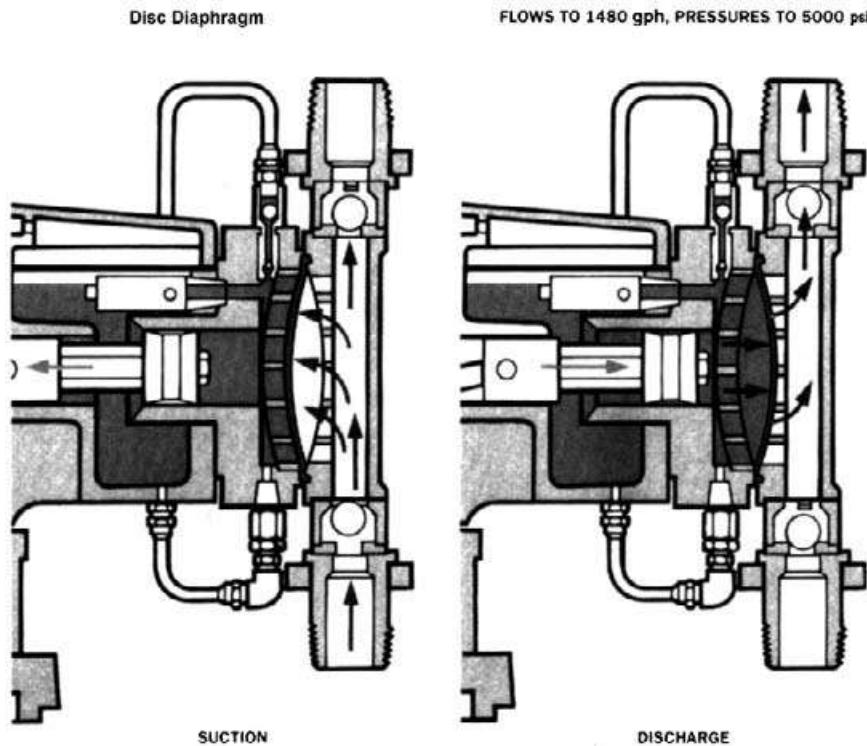
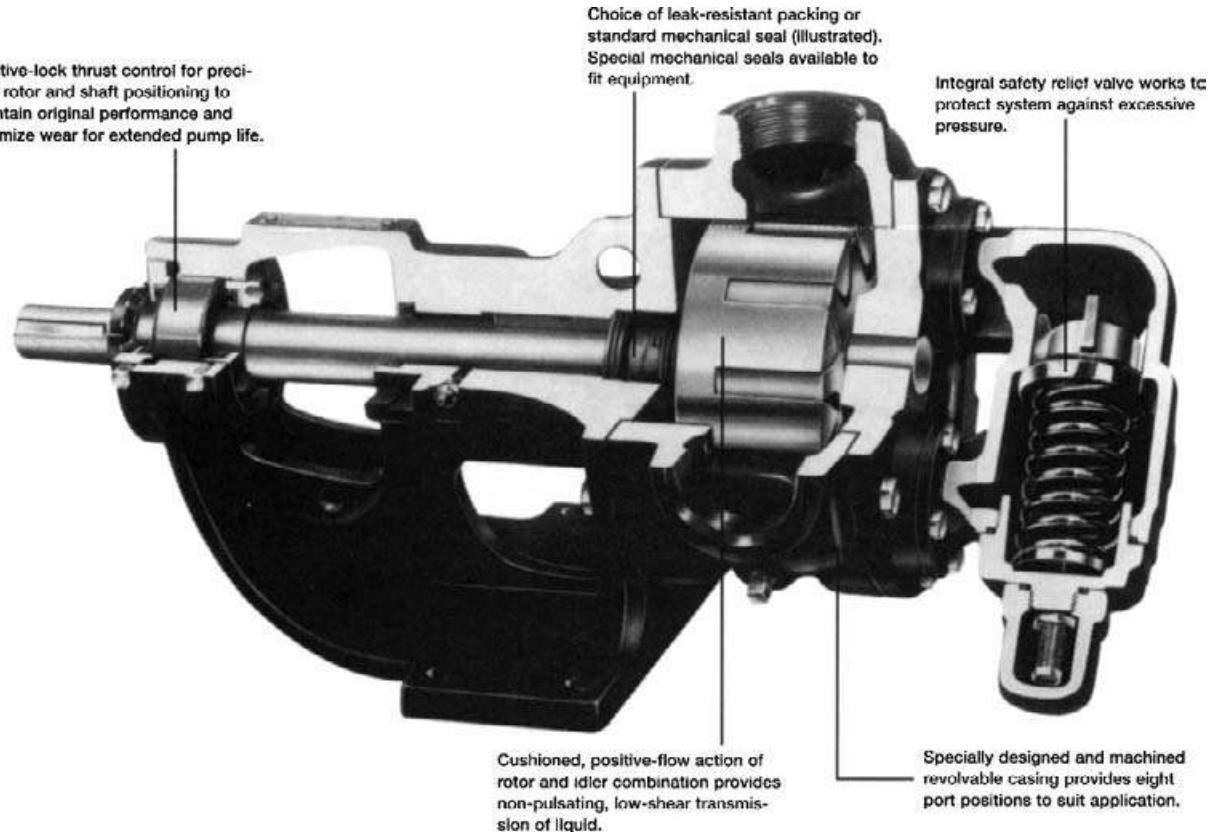


Figure 5-46 Rotary pumps. (By permission from Dolman, R.E., *Chem. Eng.*, Mar 1952, p. 159.)



**Figure 5-47a** Diaphragm metering pump, "Pulsa" series. One of several styles/types. (By permission from Pulsafeeder Inc.)



**Figure 5-47b** Typical rotary gear pump. (By permission from Viking Pump, Inc., Unit of IDEX Corp.)

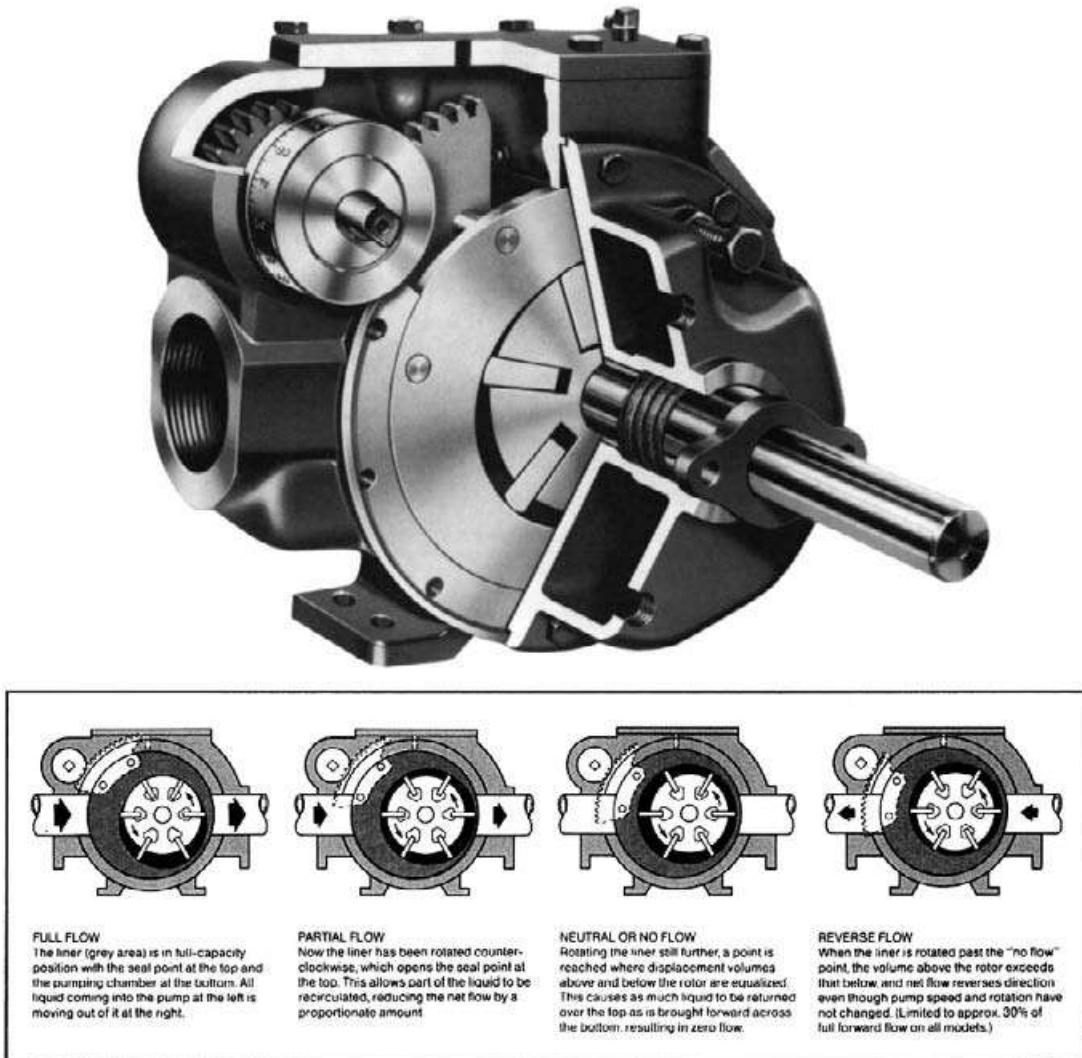


Figure 5-47c Sliding vane rotary pump. (By permission from Blackmer Pump, Dover Resources Co.)

## 2. Volume displaced [8]:

$$Q' = \frac{D''n}{231} - S'', \text{ gpm} \quad (5-57)$$

(if no vapor or gas present)

where

- $Q'$  = capacity of rotary pump, fluid plus dissolved gases/entrained gases, at operating conditions, gpm  
 $D''$  = displacement (theoretical) volume displaced per revolution(s) of driving rotor, in.<sup>3</sup>/rev  
 $n$  = speed, revolutions per minute of rotor(s), rpm  
 $S''$  = slip, quantity of fluid that leaks through internal clearances of pump per unit time, gpm.

## 3. Pump power output (WHP) [8]:

$$WHP_1 = \frac{(Q'P_{td})}{1714} \quad (5-58)$$

where

- $P_{td}$  = differential pressure between absolute pressures at the outlet and inlet to pump, psi  
 $WHP_1$  = power imparted by the pump to the fluid discharged (also liquid HP)  
 $E_v$  = volumetric efficiency, ratio of actual pump capacity to the volume displaced per unit time.

$$E_v = \frac{231Q'(100)}{(D''n)} \quad (5-59)$$

4. Brake horsepower varies directly with pressure and speed.  
5. For speed and pressure constant, BHP varies directly with viscosity.

## SELECTION

Suction and discharge heads are determined the same way as for centrifugal pumps. Total head and capacity are used in selecting

the proper rotary pump from a manufacturer's data or curves. Since viscosity is quite important in the selection of these pumps, it is sometimes better to select a larger pump running at low speeds than a smaller pump at high speeds when dealing with viscous materials.

As a general guide, speed is reduced to 25–35% below rating for each tenfold increase in viscosity above 1000 SSU. Generally, the mechanical efficiency of the pump is decreased 10% for each tenfold increase in viscosity above 1000 SSU, and referenced to a maximum efficiency of 55% at this point. [28]

## 5.19 RECIPROCATING PUMPS

Reciprocating pumps are positive displacement piston units driven by a direct connected steam cylinder or by an external power source connected to the crankshaft of the pump piston. Being positive displacement piston units, these pumps can develop very high pressures (10,000 psi and higher) for very low or high capacities (up to 1000 gpm).

### SIGNIFICANT FEATURES IN RECIPROCATING PUMP ARRANGEMENTS

#### I. Liquid pump end

##### A. Pump pressure component

1. piston
2. plunger.

##### B. Types

1. simplex, one piston
2. duplex, two piston (Figure 5-48)
3. triplex, three piston (not used as steam driven).

##### C. Piston or plunger action

1. Single-acting, one stroke per rpm
2. Double-acting, two strokes per rpm, cylinder fills and discharges each stroke (Figure 5-49).

#### D. Packing for piston or plunger

1. Piston packed: packing mounted on piston and moves with piston; applied to comparatively low pressures
2. Cylinder packed: packing stationary; plunger moves; applied to high pressures; more expensive than piston packed.

#### II. Drive end: Steam

##### A. Steam cylinders

1. Simple: single cylinder per cylinder of liquid pump; uses more steam than compound
2. Tandem compound: high and low pressure cylinders on same centerline; usually requires 80 psi or greater steam to be economical
3. Cross compound: high and low pressure cylinders arranged side-by-side with cranks 90° apart. Need for crank and flywheel arrangement only; usually requires 80 psi or greater steam to be economical. Percentage gain in compounding steam cylinders varies from 25 to 35% for non-condensing, and 25 to 40% for condensing [29].

##### B. Cylinder action

1. Direct: steam piston connected direct to liquid piston or plunger through piston rod
2. Crank and flywheel: flywheel mounted on crank shaft driven by steam cylinder.

#### III. Drive end: Power

General features same as steam, except drive always through crankshaft; speed gear increasers or reducers; V-belts, or direct coupling connection to drive shaft.

#### IV. Designation

Units are identified as steam cylinder diameter, inches; liquid cylinder diameter, inches; and length of stroke, inches.

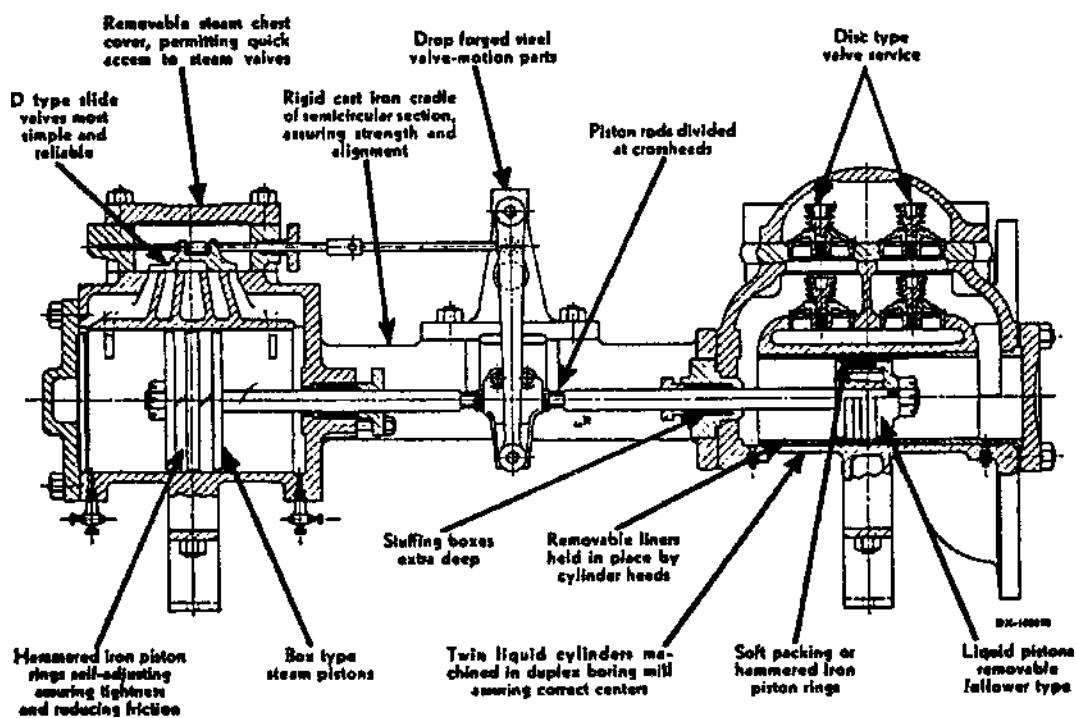


Figure 5-48 General service duplex steam-driven piston pump. (Courtesy of Worthington Corp.)

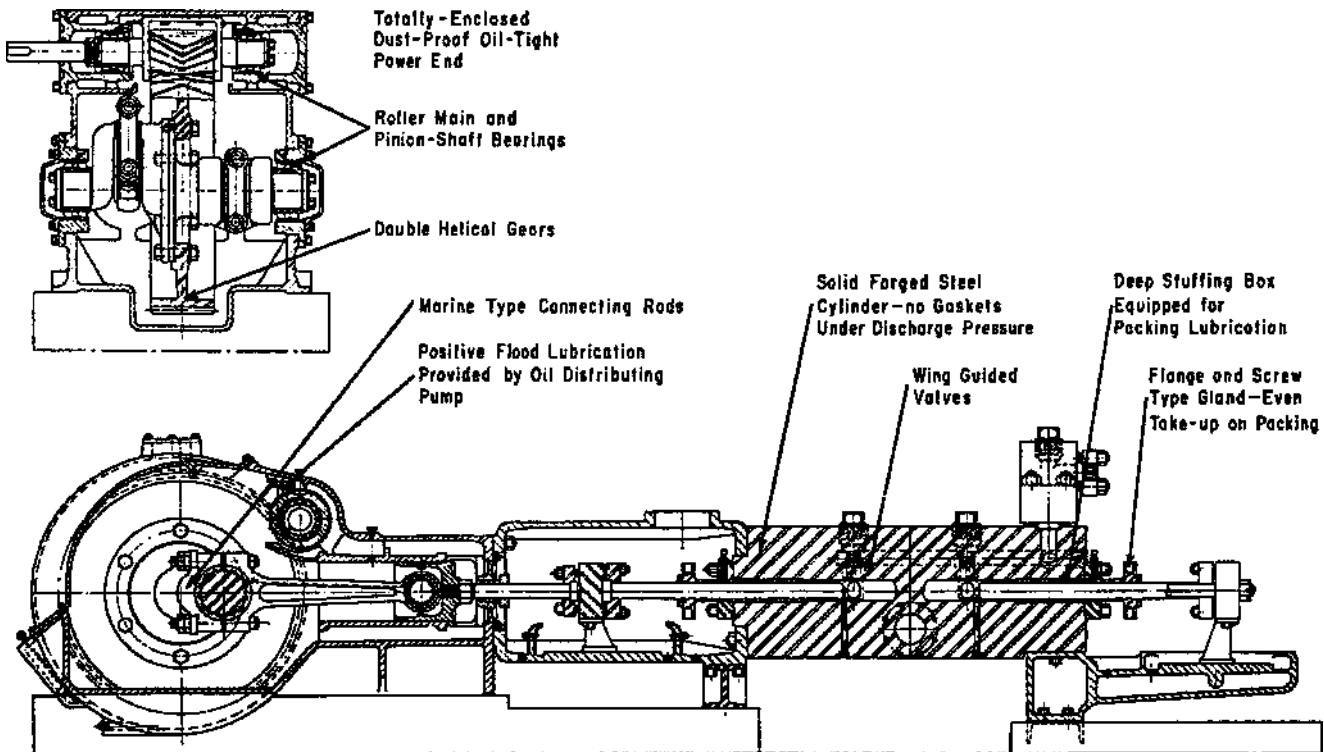


Figure 5-49 Duplex double-acting plunger pump, power driven. (Courtesy of Worthington Corp.)

## APPLICATION

*Piston type:* used for low pressure light-duty or intermittent service. Less expensive than the plunger design, but cannot handle gritty liquids.

*Plunger type:* used for high pressure heavy-duty or continuous service. Suitable for gritty and foreign material service, and more expensive than the piston design.

## PERFORMANCE

The performance of reciprocating pumps provides for ease of operation and control. Depending upon the type of piston action, the fluid may be subject to pulsations unless accumulator or surge drums are provided.

The slip of a pump is fraction or percent loss of capacity relative to theoretical. Slip is  $(1 - e_{vol})$ , where  $e_{vol}$  is the volumetric efficiency. Volumetric efficiency is the actual liquid pumped (usually considered water) relative to that which should theoretically be pumped based on piston displacement.

The NPSH<sub>R</sub> is approximately 3–5 psi of liquid above the vapor pressure of the liquid. The capacity of a pump is given in manufacturers' tables as actual, after deducting for volume occupied by piston rod and slippage. Slip varies from 2 to 10% of displacement, with 3% being a fair average.

Capacity (actual), for single-acting pumps, single cylinder:

$$Q = \frac{(12at)(e_{vol})}{(231)(2)} = 0.0204 d_p^2 t e_{vol}, \text{ gpm} \quad (5-60)$$

For double-acting pumps, single cylinder:

$$Q = (\text{two times the value for single-acting pumps}) - 0.0204 d_r^2 t, \text{ gpm} \quad (5-61)$$

where

- $a$  = area of piston or plunger, in.<sup>2</sup>
- $t$  = piston speed or travel, ft/min
- $e_{vol}$  = volumetric efficiency, fraction
- $d_p$  = diameter of piston or plunger, in.
- $d_r$  = diameter of piston rod, in.

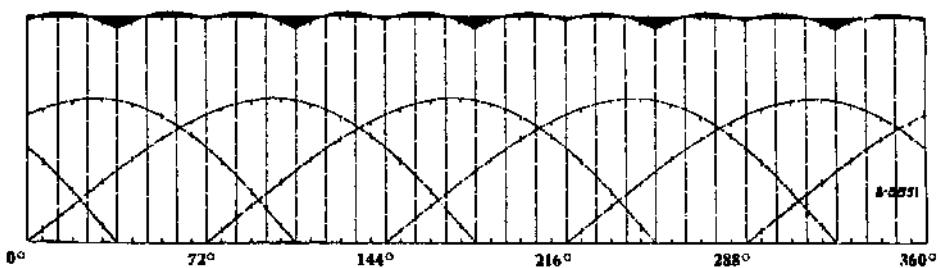
For multiple cylinders, multiply the capacities just obtained by the number of cylinders. If the piston rod does not replace pumping volume as in some arrangements, the last term of Eq. (5-58) is omitted.

## DISCHARGE FLOW PATTERNS

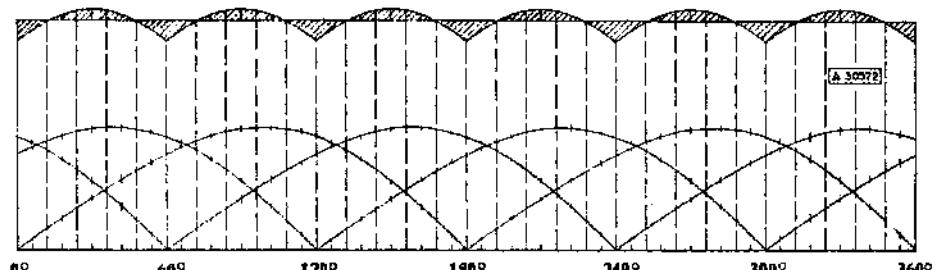
Figure 5-50 shows the discharge flow patterns for several reciprocating power pump actions which are essentially the same for steam pumps. The variations above and below theoretical mean discharge indicate the magnitude of the pulsations to be expected. Although not shown, the simplex double-acting discharge would follow the action of one piston on the duplex double-acting curve from 0 to 360°. Its variation or pulsing is obvious by inspection, and accumulator bottles would be required to smooth the flow. The simplex single-acting discharge would be one pumping stroke from 0 to 180°, then no pumping from 180° to 360°; and here again the pulse action is obvious.

**QUINTUPLEX SINGLE-ACTING PUMP**

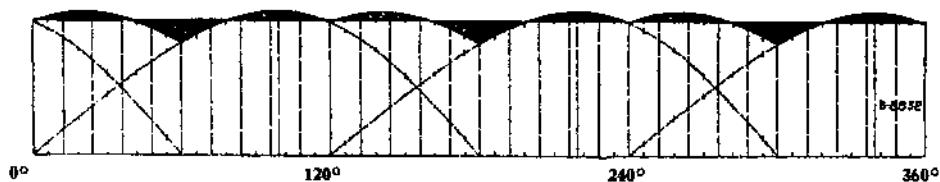
Variation Above Mean, 1.8%  
 Variation Below Mean, 5.2%  
 Total Variation, 7.0%

**SEXTUPLEX SINGLE-ACTING PUMP**

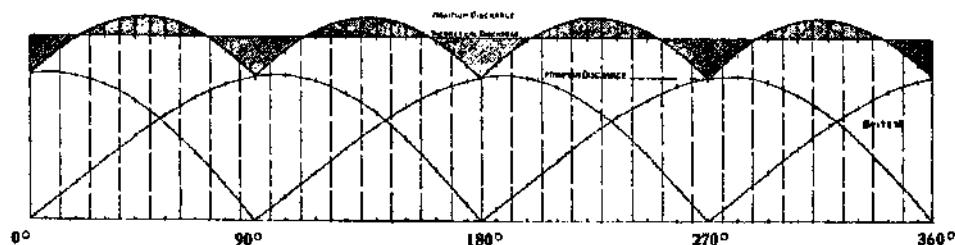
Variation Above Mean, 4.82%  
 Variation Below Mean, 9.22%  
 Total Variation, 14.04%

**TRIPLEX SINGLE-ACTING PUMP**

Variation Above Mean, 6.1%  
 Variation Below Mean, 16.9%  
 Total Variation, 23.0%

**QUADRUPLEX SINGLE-ACTING PUMP**

Variation Above Mean, 11.0%  
 Variation Below Mean, 21.5%  
 Total Variation, 32.5%

**DUPLEX DOUBLE-ACTING PUMP**

Variation Above Mean, 24.1%  
 Variation Below Mean, 21.5%  
 Total Variation, 45.6%

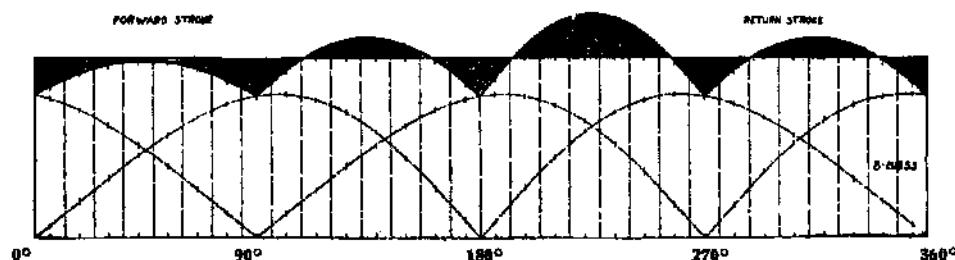


Figure 5-50 Reciprocating pump discharge flow patterns. (Courtesy of the Aldrich Pump Co.)

**HORSEPOWER**

Hydraulic:

$$HHP = \frac{(Q_{\text{actual}})(H)}{3960} \quad (5-62)$$

Brake:

$$BHP = \frac{HHP}{e} \quad (5-63)$$

where

 $e$  represents the total overall efficiency, and is

$$e = e_m(e_{\text{vol}})$$

and  $e_m$  is the mechanical efficiency and  $e_{\text{vol}}$  is the volumetric efficiency, fraction.

Mechanical efficiencies of steam pumps vary with the types of pump, stroke, and the pressure differential. Some representative values are 55–80% for piston pumps with strokes of 3 and

24 in., and pressure differential up to 300 psi. For the same strokes a plunger design varies from 50 to 78%, and at over 300 psi differential the efficiencies are 41–67% [30]. Steam required is approximately 120 lb/h per BHP.

**PUMP SELECTION**

Reciprocating pump selection follows the fundamentals of centrifugal pumps:

1. Evaluate suction side head loss.
2. Evaluate discharge side head loss.
3. Determine system static pressure.
4. Determine total differential head across pump.
5. Determine the NPSH<sub>A</sub> on suction of pump.
6. From manufacturer's performance tables, select pump nearest to GPM and head requirements.
7. Contact manufacturer for final recommendations, give complete system requirements and physical properties of liquid. Figure 5-51 will serve this purpose.

<b><u>HORIZONTAL DIRECT ACTING STEAM PUMP or POWER PUMP</u></b>		ITEM _____			
Apparatus _____ Operate at _____ # GA; Max WP _____ # GA & Max T _____ °F _____ Date _____ By _____ Based on Steam at _____ lb/in. <sup>3</sup> GA and _____ °F _____ Dated _____ Item No. _____ Size and Type _____					
<b>A. CONDITIONS OF SERVICE</b>		<b>DATA</b>			
Operation _____					
Pumped Material _____					
°API at 60/60°F _____					
Specific Gravity at 60°F _____					
Specific Gravity at P.T. _____					
Viscosity at P.T. _____					
Pumping Temperature °F _____					
US GPM at 60°F _____					
US GPM at P.T. _____					
Discharge Pressure, lb/in. <sup>2</sup> GA _____					
Suction Pressure, lb/in. <sup>2</sup> GA _____					
Approx. Vapor Pressure of Liquid _____					
Exhaust Steam _____					
<b>B. PUMP SPECIFICATIONS</b>					
Suction Valves – Number _____					
Suction Valves – Size: in. _____					
Suction Valves – Area each, in. <sup>2</sup> _____					
Discharge Valves – Number _____					
Discharge Valves – Size, in. _____					
Discharge Valves – Area each, in. <sup>2</sup> _____					
Suction Connection _____	Size: _____	Series: _____			
Discharge Connection _____	Size: _____	Series: _____			
Type of Rings–Steam End: _____	Pump end: _____				
<b>C. MATERIALS</b>		<b>STEAM END</b> <b>LIQUID END</b>			
Cylinders _____					
Liners _____					
Pistons _____					
Piston Rods _____					
Valves _____					
Valve Seats _____					
Valve Springs _____					
Packing _____					
<b>D. PERFORMANCE</b>					
Piston Speed – ft/min _____	Required HP _____	Installed HP _____			
RPM _____	Driver RPM _____				
Stalling Pressure – lb/in. <sup>2</sup> GA _____					
Steam Consumption _____					
Copy to _____	Date _____	Checked _____	Date _____	Approved _____	Date _____
Remarks: _____					

**Figure 5-51** Horizontal direct-acting steam pump or power pump.

## 5.20 SELECTION RULES-OF-THUMB

Every pump has a specific curve that relates flow, head, power,  $NPSH_R$ , and efficiency for specific impeller diameters for that particular unit. This allows correct selection of the impeller diameter. Therefore, during specification, the objective is to select a pump with a rated or design point as close as possible to the best efficiency point (BEP), as determined by the pump manufacturer [31].

The following are general guidelines for proper selection [31].

1. Select the pump based on rated conditions.
2. The BEP should be between the rated point and the normal operating point.
3. The head/capacity characteristic curve should continuously rise as flow is reduced to shutoff (or zero flow).
4. The pump should be capable of a head increase at rated conditions by installing a larger impeller.
5. The pump should not be operated below the manufacturer's minimum continuous flowrate.

The pump has a specific  $NPSH_R$ , which varies depending on the head and flow. Once the specific pump model and size have been determined from the basic process information, the materials of construction must be chosen. Selection depends on fluid properties (e.g., viscosity, corrosiveness, and erosiveness) and the presence of dissolved gases. In general, adequate knowledge of the chemical composition of the fluid helps to ensure proper material selection of the pump and its shaft seal.

## NOMENCLATURE

$a$	Area of piston or plunger, in. <sup>2</sup>	$E_n$	Fraction entrained gas by volume at atmospheric pressure
$B_d$	Bell diameter of vertical sump pump, ft	$E_v$	Volumetric efficiency ratio of actual pump capacity to volume displaced per unit of time
BHP	Brake horsepower	$E_w$	Pump efficiency with water, %
$BHP_{vis}$	Brake horsepower when handling viscous material	$E_{vis}$	Pump efficiency with viscous fluid, %
$C_E$	Viscosity correction for efficiency to convert to water performance	$e$	Pump efficiency, fraction
$C_H$	Viscosity correction for head, to convert to water performance	$e_w$	Pump efficiency with water, fraction
$C_Q$	Viscosity correction for capacity, to convert to water performance	$e_{vis}$	Pump efficiency with viscous fluid, fraction
$c_p$	Specific heat of liquid, Btu/lb –°F	$e_{vol}$	Volumetric efficiency, fraction
$D$	Height of liquid (static) above (+) or below (–) the centerline of the pump on discharge side, ft	$e_M$	Minimum safe flowing efficiency, overall pump fraction
$D'$	Incremental height of liquid (static) above normal $D$ level, to establish "worst case" condition, ft	$g$	Acceleration of gravity, 32 ft/s <sup>2</sup> (9.81 m/s <sup>2</sup> )
$D''$	Theoretical displacement volume displaced per revolution(s) of driving rotors, in. <sup>3</sup> /rev	$H$	Total head developed by a pump ft (m) of fluid; or total head/stage, ft, or,
$d$	Impeller diameter, in.	$H$	Static head discharge ft (m)
$d_p$	Diameter of piston or plunger, in.	$H_{so}$	Head at no flow, or shutoff, ft
$d_r$	Diameter of piston rod, in.	$H_{vis}$	Head of viscous fluid, ft
$d'$	Liquid displacement, ft <sup>3</sup> /min	$H_w$	Water equivalent head, ft
$d''$	Theoretical displacement, ft <sup>3</sup> /min	$h_d$	Discharge head on a pump, ft of fluid
EHP	Electrical horsepower	$h_s$	Suction head (or suction lift) on a pump, ft of fluid
$E$	Efficiency, %	$h_{SL}, h_{DL}$	Friction losses in pipe and fittings: subscript SL for suction line; and DL for discharge line, ft of fluid
		$h_v$	Velocity head, ft of fluid
		$L = S$	Static head, suction side, ft
		$I$	Water depth in sump, ft
		$N_s$	Specific speed, dimensionless
		$N_p$	Number of pumps
		$n$	Rotative speed, revolutions per minute = RPM = rpm
		$P$	Positive external pressure on surface of liquid (+) or partial vacuum on surface of liquid (–)
		$P_a$	Atmospheric pressure or absolute pressure in vessel, psia
		$P_{so}$	Brake horsepower at shutoff or no flow
		$P_{td}$	Differential pressure between absolute pressures at outlet and inlet to pump, psi
		$P_{vp}$	Vapor pressure of liquid at pumping temperature, psia
		$p'$	Absolute pressure, in. Hg abs
		$p'_a$	Atmospheric pressure or absolute pressure in vessel expressed as ft of fluid
		$p'_{vp}$	Vapor pressure of liquid at pumping temperature expressed as ft of fluid
		$Q$	Flow rate, gpm
		$Q'$	Capacity of rotary pump, fluid plus dissolved gases/entrained gases at operating conditions, gpm
		$Q_M$	Minimum flow, gpm
		$Q_N$	Head at best efficiency point on pump curve, ft
		$Q_{vis}$	Viscous liquid capacity, gpm
		$Q_w$	Water capacity, gpm
		$S$	Suction static head, ft, or height of liquid (static) above (+) or below (–) the centerline of the pump on suction side, ft, or,
		$S_L$	Suction lift, negative suction head, ft
		$S'_L$	Worst case suction side static lift, ft
		$S''$	Slip, quantity of fluid that leaks through internal clearances of rotary pump per unit time, gpm
		sp gr	Specific gravity of liquid at pumping temperature referred to water = 1.0
		$s$	Stroke, in.
		$\Delta t$	Temperature rise, °F

$\Delta T_r$	Temperature rise, °F/min
$t$	Piston speed or travel, ft/min
$V$	Liquid velocity, ft/s
$v$	Average velocity, ft/s
$W$	Width of channel with series pump, ft
$W_1$	Weight of liquid in pump, lb
$WHP$	Water or liquid horsepower
$WHP_1$	Power imparted by pump to fluid discharged (also liquid horsepower)

L	Liquid
d	Discharge side of pump
$d_i$	Friction losses for pipe fittings and related items on discharge side of pump
$s_1$	Friction losses for pipe valves and other system losses, suction side of pump
R	Required by pump (NPSH)
s	Suction side of pump

**SUBSCRIPTS**

- 1, 2 Refer to first and second condition respectively  
 A Available from pump system (NPSH)

**GREEK SYMBOLS**

$$\rho = \text{Fluid mass gravity, lb/ft}^3$$

**APPENDIX****A SECURE LUBRICATION**

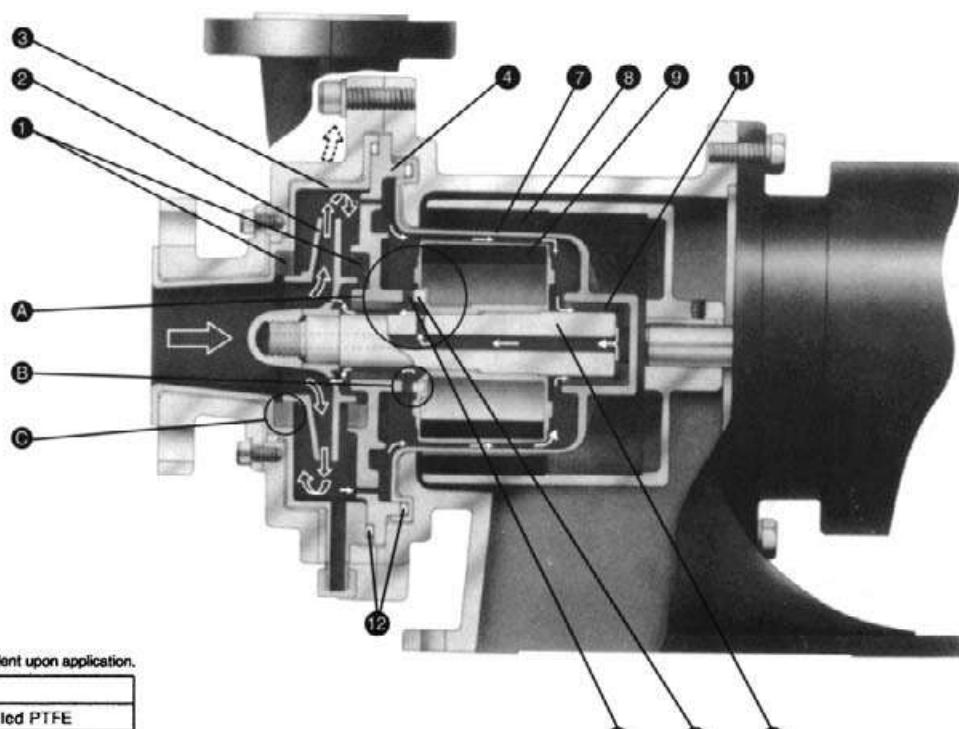
Liquid from the casing flows between the rear casing and the magnet lining which prevents clogging. The fluid is then forced to flow to the rear bushing then through the shaft to the front bushing. This flow guarantees perfect lubrication for the bushing.

**B LONG LIFE THRUST RING**

The thrust ring is connected to the magnet not to the impeller, which reduces its rotating velocity. The lower velocity increases the life of the thrust ring.

**C NO ADHESIVE**

All of the bushings as well as the liner ring and thrust ring are not attached by adhesives. Thus the parts are easily replaced and the parts are free from any weakness of the adhesive.



Standard materials: other materials available dependent upon application.

No.	Part Name	Material
1	Liner Ring	Carbon-filled PTFE
2	Impeller	PVDF (KYNAR)**
3	Casing	PVDF (KYNAR)
4	Bushing Plate	PVDF (KYNAR)
5	Front Bushing	Carbon-filled PTFE
6	Thrust Ring	CERAMIC
7	Rear Casing	Carbon-filled PVDF
8	Outer Magnet	
9	Inner Magnet	
10	Shaft	CERAMIC
11	Rear Bushing	Carbon-filled PTFE
12	"O" Ring	VITON

\*\* Kynar is a registered trademark of Pennwalt Corp.

**Figure 5-52** Sealless magnetic drive centrifugal pump, no seals, no leakage, no coupling. Chemical resistance depends on materials of construction (see Table 5-2). (By permission from LaLabour Pump Co.)

Fluid at approximately 60% of discharge pressure is circulated through the bearings and over the rotor for cooling and lubrication and returns through the hollow shaft to suction pressure.

**Terminal Plate**  
O-ring sealing for positive secondary fluid containment.

**Bearing Monitor**  
The standard bearing monitor solves the most basic problem common to all sealless pumps—detecting normal bearing wear so that routine maintenance can be accomplished before serious motor damage occurs. It responds to bearing wear in both the axial and radial directions and is over 98% effective on 70,000 operational units.

In addition, the monitor is useful in detecting corrosion of the stator liner and rotor sleeve since the contact tip is supplied in the same metallurgy but one-half the thickness of those components.

**Shaft Sleeves**  
Available in a variety of surface treatments to suit the specific fluid applications. Replaced when bearings are changed for like new wear surfaces and clearances.

**Bearings**  
Available in a variety of materials to suit the specific fluid application. Oversized for minimum loading.

**Dry Stator**  
Eliminates oil pressure relief valve required for oil filled stators. Integrity of secondary leak containment shell is maintained.

**Flanges**  
150 and 300 psig rating (raised face). Self venting centerline discharge. Compatible with ANSI B73.1 dimensions.

**Impeller**  
High efficiency design, open and closed configurations.  
(Optional flow inducers available for minimum NPSH requirements.)

**Hollow Shaft (Basic, HB and HX Models Only)**  
Assures complete self-venting and prevents vapor collection at the bearings.

**Thermostats**  
Embedded in the hot spot of the windings for protection against overheating.

**Thrust Washers**  
Absorb thrust loads during upset conditions and provide back-up to hydraulic thrust balancing.

## Motors

In the Sundyne Canned Motor design, the entire outside of the motor is enclosed in a secondary leakage containment shell or can. Primary leakage protection is provided by corrosion resistant liners which are seal welded and 100% leak checked to assure that pumped fluid does not contact the stator windings or rotor core. There is no shaft protrusion to seal and thus no seals to leak.

Pumped fluid is circulated inside the

stator liner to cool the motor, and lubricate the bearings.

Motor windings and insulation systems are specially designed, developed and applied as an integral part of the pump so that design life is at least as great as for conventional air cooled motors. Winding temperature is primarily influenced by pumped fluid temperature and secondarily by use of cooling jacket. Fluid temperature is considered in pump

application to assure full winding life. Thermostats are embedded in the hot spots of windings for shutdown in case of overheating.

Motors are suitable for use in general purpose areas and in Class I, Division 2, Group C and D areas for a wide range of pump fluid temperatures. For Class I, Division 1 Group C and D; U.L. listed, explosion proof motors are available.

Figure 5-53 Sealless canned centrifugal pump, primary and secondary leakage containment (see Table 5-2). (Courtesy of Sundstrand Fluid Handling Co.)

**MODIFIED STUFFING BOX** . . . available with John Crane Type 1 mechanical seals. These mechanical seals are easy to install in the field and are carried in stock.

**CASING** is horizontally split to permit removal of top half without disturbing piping. Suction and discharge connections are in lower half of casing.

**STANDARD DEEP STUFFING BOX** carries generous packing; is easily accessible. Stuffing box contains split seal cages and split glands. Gland bolts are completely removable. Bushing at bottom of stuffing box is close fitting and readily replaceable.

**SHAFT** is heat treated, extra heavy to take maximum radial thrust. Shaft supports impeller between bearings for longer bearing, wearing ring and packing life.

**WEARING RINGS** protect casing, are easily replaced, assure continuous high efficiency. Impeller rings and tongue and groove casing rings are available at nominal extra cost.

**IMPELLER**, hydraulically balanced — double suction enclosed type for better performance under critical suction conditions.

**BALL BEARINGS**, grease-lubricated, held in bearing housing of heavy one-piece construction — are exceptionally well sealed against moisture. Brackets holding housing are cast integral with the casing. Bearing covers are also of heavy one-piece construction and are interchangeable end for end . . . and contain stationary member of labyrinth seal.

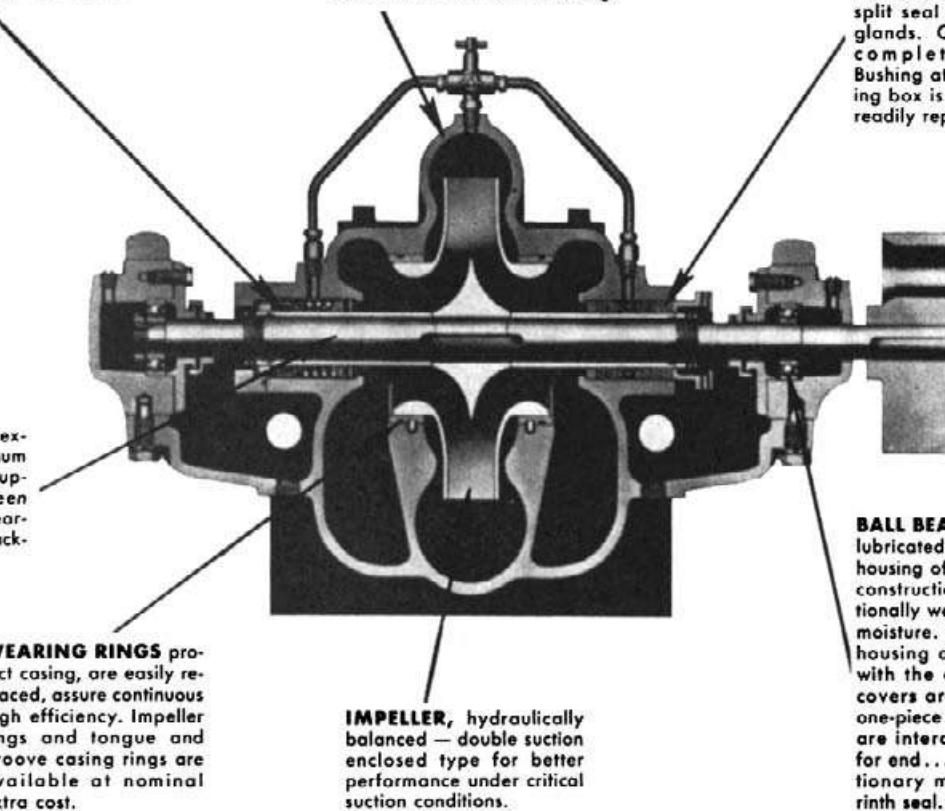


Figure 5-54 Centrifugal pump, double suction-stage impeller. (Courtesy of Allis-Chalmers Mfg. Co.)

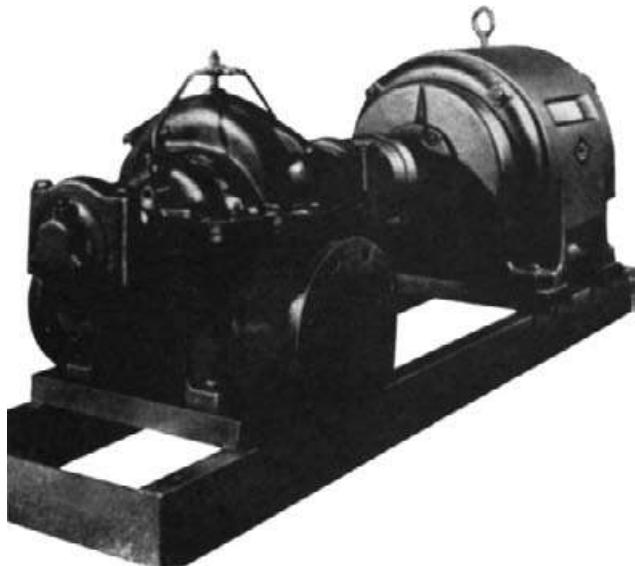
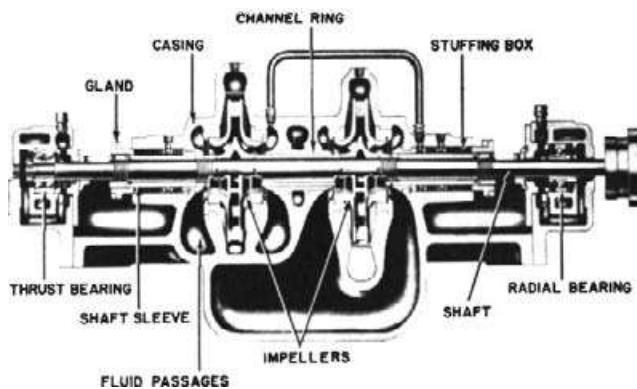
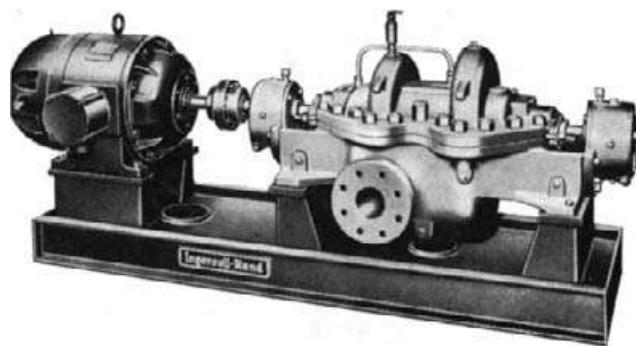


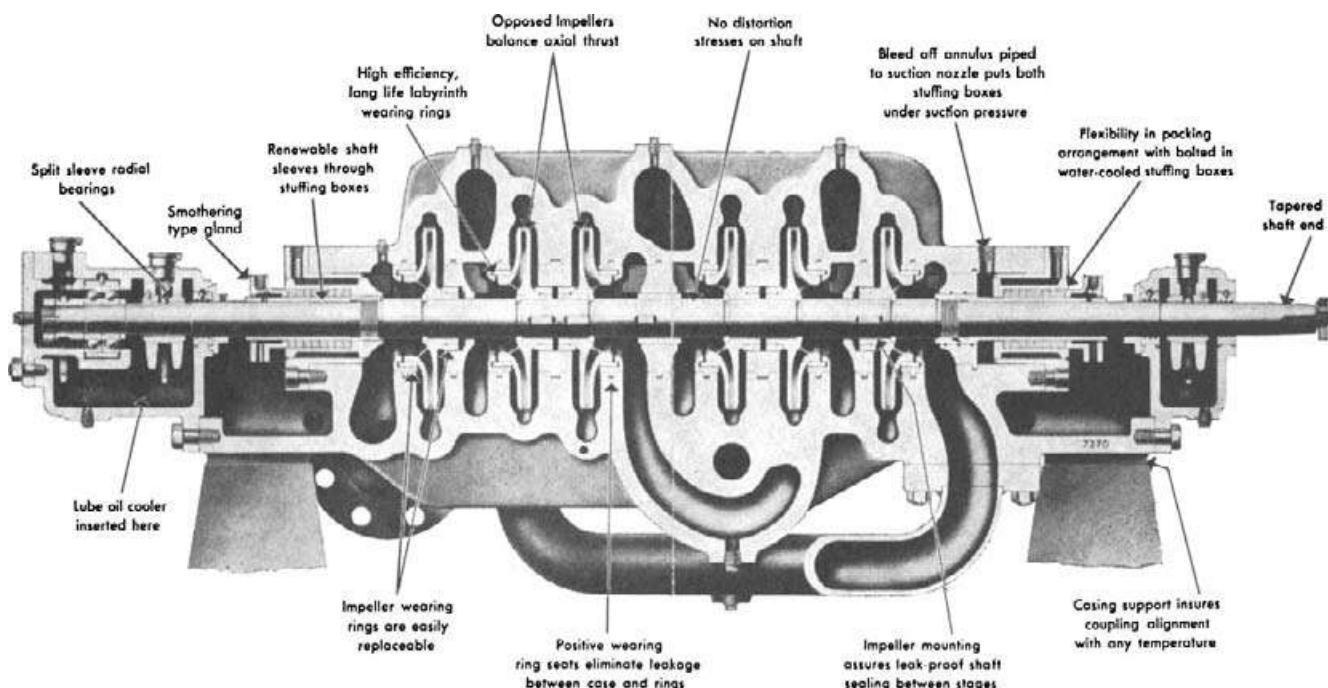
Figure 5-55 External view double suction single-stage pump. (Courtesy of Allis-Chalmers Mfg. Co.)



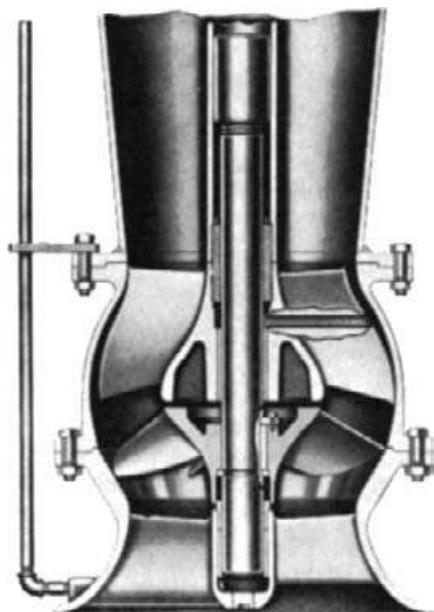
**Figure 5-56** Cross-section horizontal two-stage horizontal split case centrifugal pump. (Courtesy of Ingersoll-Rand Co.)



**Figure 5-57** Exterior view of horizontal two-stage split case centrifugal pump. (Courtesy of Ingersoll-Rand Co.)



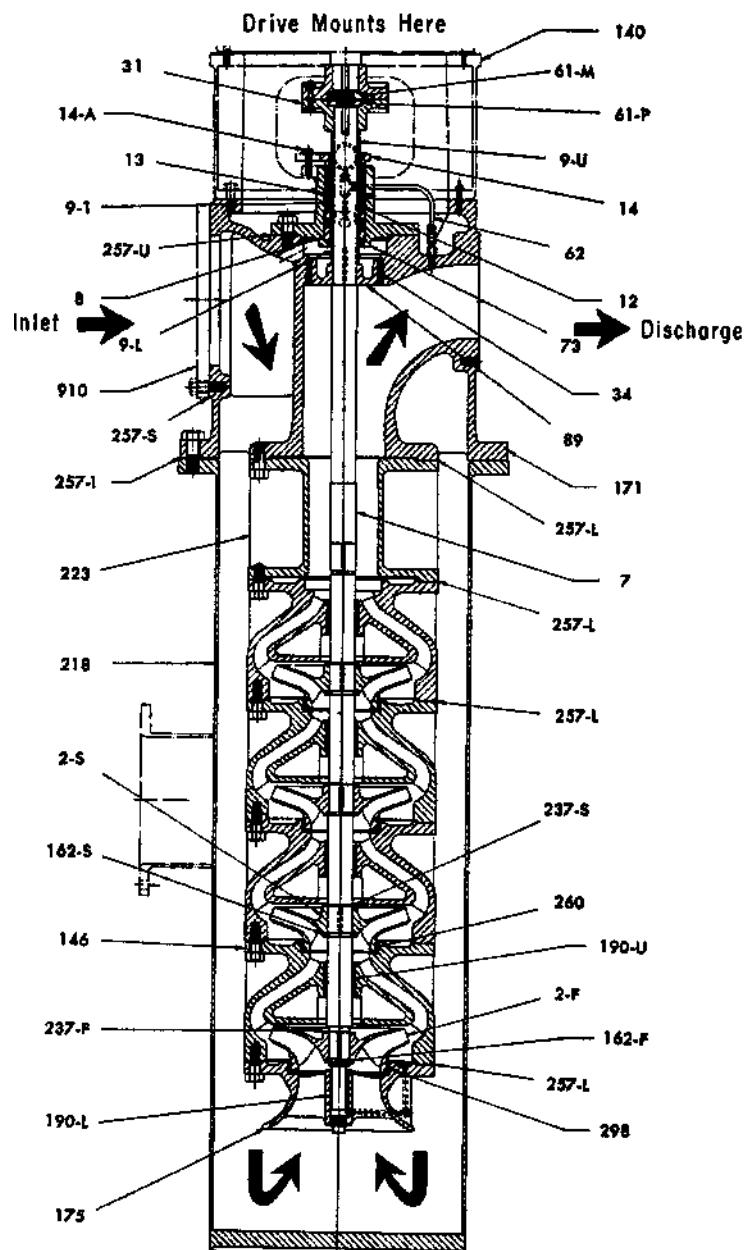
**Figure 5-58** Refinery oil and boiler feed high pressure centrifugal pump. (Courtesy of Delaval Steam Turbine Co., currently Transamerica Delaval, Inc.)



**Figure 5-59** Vertical single-stage mixed flow type pump, liquid inlet and impeller. (Courtesy of Peerless Pump Div., FMC Corp.)

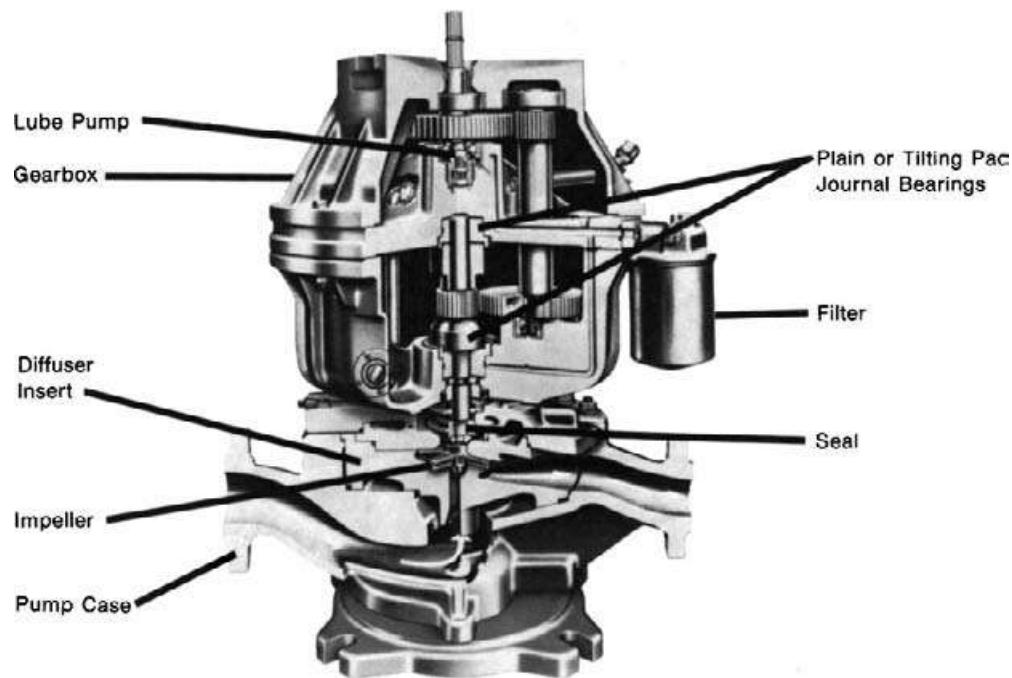


**Figure 5-60** Vertical propeller type pump. (Courtesy of Peerless Pump Div., FMC Corp.)



PARTS LIST	
Cat. No.	Part Name
2-F	First Stage Impeller
2-S	Second Stage Impeller and Above
34	Nozzle Head Bushing
260	Diffuser Ring
298	Suction Bell Ring
8	Stuffing Box Bushing
9-L	Lower Shaft Sleeve
9-I	Intermediate Shaft Sleeve
9-U	Upper Shaft Sleeve
12	Packing
13	Seal Cage
14	Gland
14-A	Gland Bolt
31	Complete Coupling
61-P	Pump Half Coupling Lock Nut
61-M	Motor Half Coupling Lock Nut
62	Complete Piping
73	Stuffing Box
89	Balance Disk
140	Motor Support Column
146	Diffuser
162-F	1st. Stg. Impeller Retaining Collar
162-S	2nd Stg. and Above Impeller Retaining Collar
171	Nozzle Head
175	Suction Bell
190-L	Lower Sleeve Bearing
190-U	Upper Sleeve Bearing
218	Tank
223	Spacer Column
237-F	First Stage Snap Ring
237-S	Second Stage Snap Ring and Above
257-L	Lower Gasket
257-I	Intermediate Gasket
257-U	Upper Gasket
257-S	Blind Flange Gasket
7	Shaft with Keys
910	Blind Flange

Figure 5-61 Vertical multistage centrifugal pump with barrel casing. (Courtesy of Allis-Chalmers Mfg. Co.)



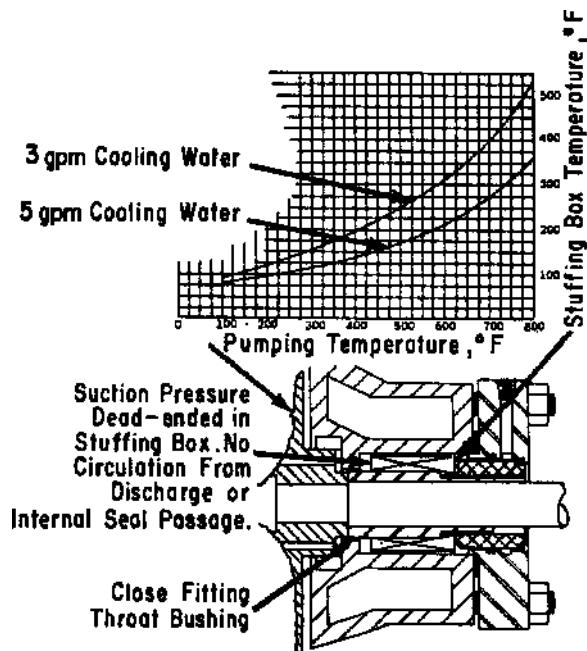
**Figure 5-62** Single-stage, high speed (>5000 ft) centrifugal process pump. Pumps are high speed, gear driven, and especially suited for applications in light hydrocarbon liquid service. (Courtesy of Sundstrand Fluid Handling, Inc.)



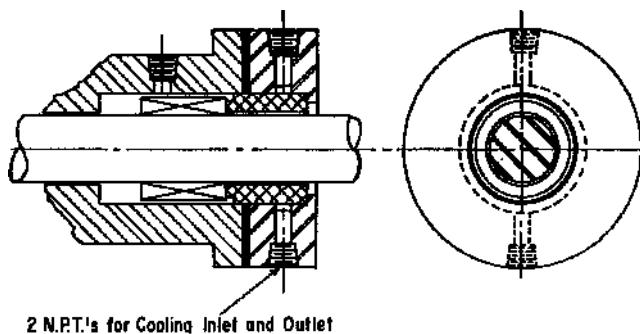
**Impeller:** Since the Sundyne straight, radial-bladed impeller does not require close running clearances, it eliminates the necessity for oversizing to compensate for performance deterioration common to conventional pumps. Clearance of the Sundyne impeller is 0.030 to 0.070 inch.

**Inducer:** An optional helical inducer is available to substantially reduce the pump NPSH requirement.

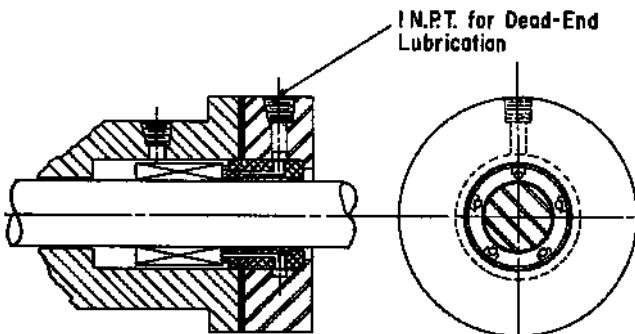
**Figure 5-63** Impeller and inducer (optional) for pumps in Figure 5-12. (Courtesy of Sundstrand Fluid Handling, Inc.)



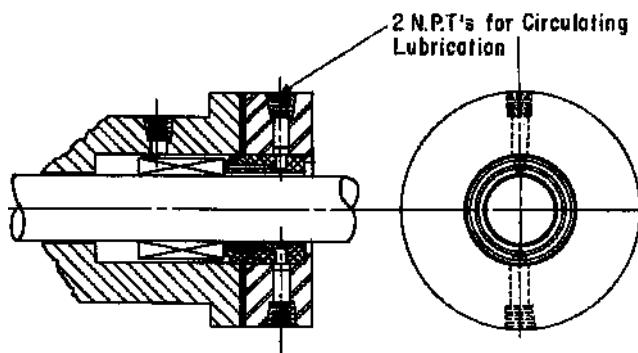
**Figure 5-64** Water-jacketed stuffing box. (By permission from Hummer, H.P. and W.J. Ramsey, Bulletin SD 752, O & T, Durametallic Corp.)



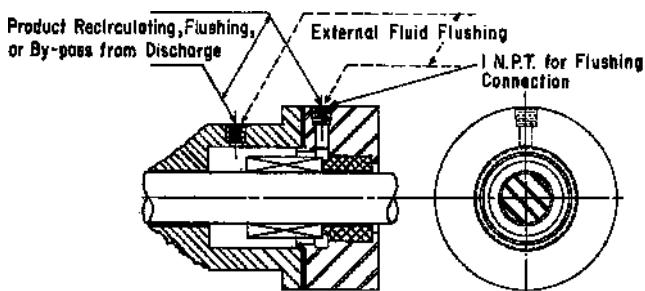
**Figure 5-65** Gland plate cooling. (By permission from Hummer, H.P. and W.J. Ramsey, Bulletin SD 752, O & T, Durametallic Corp.)



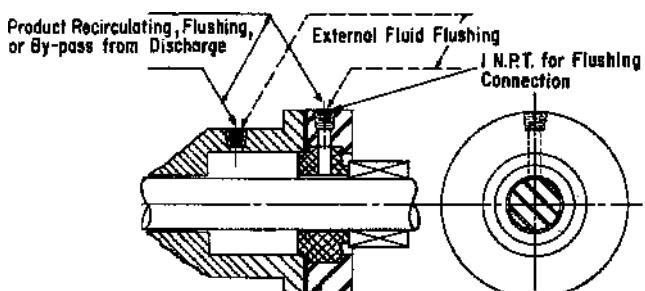
**Figure 5-66** Dead-end lubrication. (By permission from Hummer, H.P. and W.J. Ramsey, Bulletin SD 752, O & T, Durametallic Corp.)



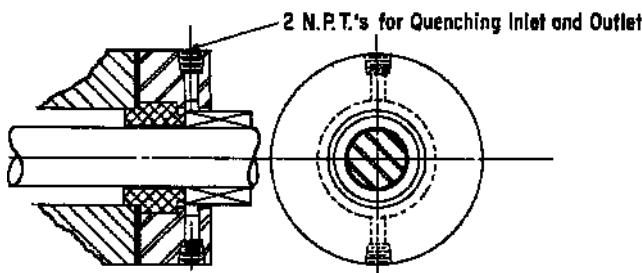
**Figure 5-67** Circulating lubrication. (By permission from Hummer, H.P. and W.J. Ramsey, Bulletin SD 752, O & T, Durametallic Corp.)



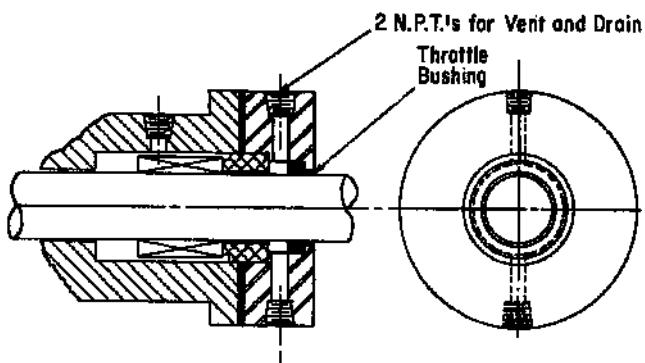
**Figure 5-68** Flushing inside seal. (By permission from Hummer, H.P. and W.J. Ramsey, Bulletin SD 752, O & T, Durametallic Corp.)



**Figure 5-69** Flushing outside seal. (By permission from Hummer, H.P. and W.J. Ramsey, Bulletin SD 752, O & T, Durametallic Corp.)



**Figure 5-70** Quenching outside seal. (By permission from Hummer, H.P. and W.J. Ramsey, Bulletin SD 752, O & T, Durametallic Corp.)



**Figure 5-71** Vent and drain. (By permission Hummer, H.P. and W.J. Ramsey, Bulletin SD 752, O & T, Durametallic Corp.)

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## MECHANICAL SEPARATIONS

**P**ractically every process operation requires the separation of entrained material or two immiscible phases in a process. This may be either as a step in the purification of one stream, or a principal process operation [1]. These separations may be as follows:

1. liquid particles from vapor or gas
2. liquid particles from immiscible liquid
3. dust or solid particles from vapor or gas
4. solid particles from liquid
5. solid particles from other solids.

These operations may sometimes be better known as mist entrainment, decantation, dust collection, filtration, centrifugation, sedimentation, screening, classification,

scrubbing, and so on. They often involve handling relatively large quantities of one phase in order to collect or separate the other. Therefore the size of the equipment may become very large. For the sake of space and cost it is important that the equipment be specified and rated to operate as efficiently as possible [2]. This subject will be limited here to the removal or separation of liquid or solid particles from a vapor or gas carrier stream (points 1 and 3 above) or separation of solid particles from a liquid (point 4). Svarosky's review is helpful [3].

Other important separation techniques such as pressure-leaf filtration, centrifugation, rotary drum filtration, and others all require technology very specific to the equipment and cannot be generalized in many instances.

### 6.1 PARTICLE SIZE

The particle sizes of liquid and solid dispersoids will vary markedly depending upon the source and nature of the operation generating the particular particles. For design of equipment to reduce or eliminate particles from a fluid stream, it is important either to know from data the range and distribution of particle sizes, or be in a position to intelligently estimate the normal and extreme expectancies. Figures 6-1a and b give a good overall picture of dimensions as well as the descriptive terminology so important to a good understanding of the magnitude of a given problem. The significant laws governing particle performance in each range are also shown.

Particle sizes are measured in microns ( $\mu$ ). A micron is  $1/1000$  mm. or  $1/25,400$  in. A millimicron ( $m\mu$ ) is  $1/1000$  of a micron, or  $1/1,000,000$  mm. Usually particle size is designated as the average diameter in microns, although some literature reports particle radius. Particle concentration is often expressed as grains per cubic foot of gas volume. One grain is  $1/7000$  of a pound.

The mechanism of formation has a controlling influence over the uniformity of particle size and the magnitude of the dimensions. Thus, sprays exhibit a wide particle size distribution, whereas condensed particles such as fumes, mists, and fog are particularly uniform in size. Table 6-1 gives the approximate average particle sizes for dusts and mists which might be generated around process plants. Figure 6-2 indicates the size ranges for some aerosols, dusts, and fumes. Table 6-2 gives typical analysis of a few dusts, and Table 6-3 gives screen and particle size relationships. Table 6-4 gives approximate mean particle size for water spray from a nozzle.

### 6.2 PRELIMINARY SEPARATOR SELECTION

The Sylvan Chart [7] of Figure 6-3 is useful in preliminary equipment selection, although arranged primarily for dust separations; it is applicable in the appropriate parts to liquid separations. Perry [8] presents a somewhat similar chart that is of different form but contains much of the same information as Figures 6-1a and b.

#### EXAMPLE 6-1

##### Basic Separator Type Selection [7, 9]

A suitable collector will be selected for a limekiln to illustrate the use of the Sylvan Chart (Figure 6-3). Referring to the chart, the concentration and mean particle size of the material leaving the kiln can vary between  $3$  and  $10$  grs/ $ft^3$  with  $5$ – $10$   $\mu$  of mass mean particle size. Assume an inlet concentration of  $7.5$  gr/ $ft^3$  and inlet mean particle size of  $9 \mu$ . Projection of this point vertically downward to the collection efficiency portion of the chart will indicate that a low resistance cyclone will be less than 50% efficient; a high efficiency centrifugal will be 60–80% efficient, and a wet collector, fabric arrester, and electrostatic precipitator will be 97% + efficient. The last three collectors are often preceded by a pre-cleaner so a high efficiency centrifugal will be selected. Using the average line of this group, the efficiency will be 70%. Therefore, the effluent from this collector will have a concentration of  $7.5(1.00 - 0.70) = 2.25$  gr/ $ft^3$ .

Draw a line through the initial point with a slope parallel to the lines marked "industrial dust." Where deviation is not known, the average of this group of lines will normally be sufficiently accurate to predict the mean particle size in the collector effluent. Where this line intersects the horizontal line marked  $2.25$  gr/ $ft^3$ , a vertical line through the point will indicate the effluent mean particle size of  $6.0 \mu$ .

A projection of this point of collector effluent vertically downward shows that a second high efficiency centrifugal will be less than 50% efficient. A wet collector, fabric arrester, and electrostatic precipitator will be not less than 93% efficient. Selection of a good wet collector will show an efficiency of 98%. The effluent leaving this collector will have a concentration of  $2.25(1.00 - 0.98) = 0.045$  gr/ $ft^3$ . Using the line initially drawn, the point where it intersects the line of  $0.045$  gr/ $ft^3$  will indicate a mean particle size in the effluent of  $1.6 \mu$ .

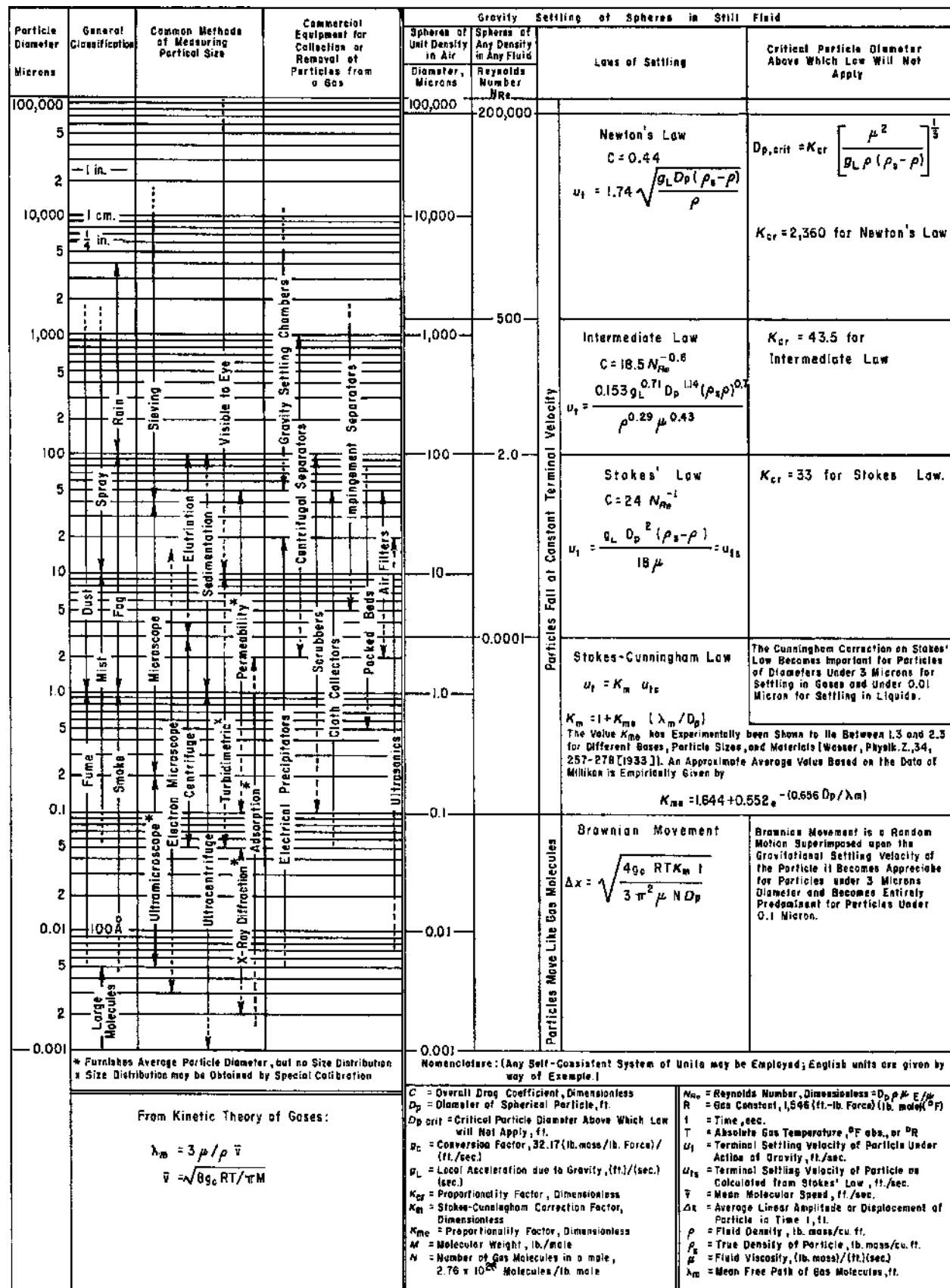
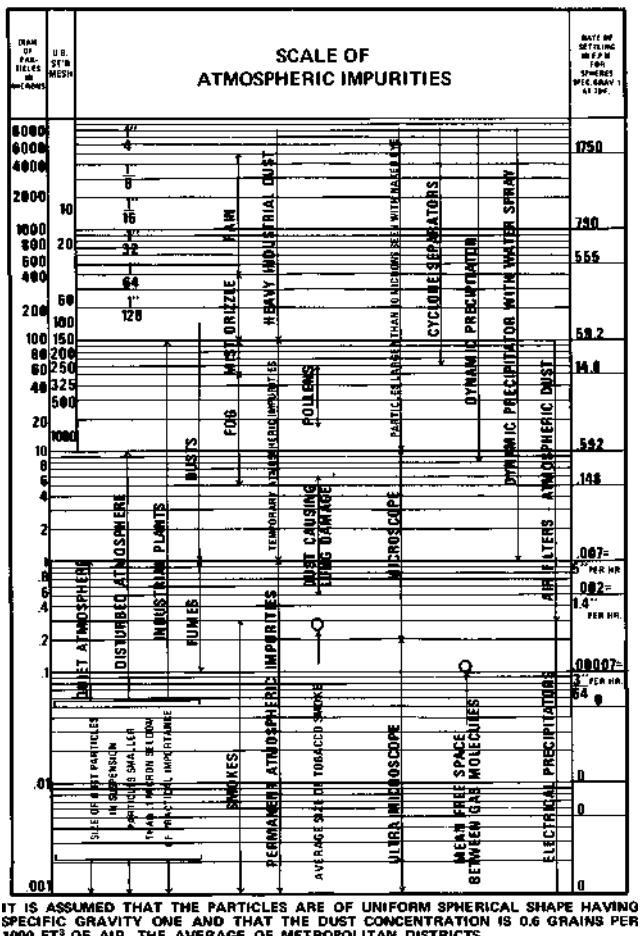


Figure 6-1a Characteristics of dispersed particles. (By permission from Perry, J.H., ed., *Chemical Engineers Handbook*, 3rd ed., 1950, McGraw-Hill Company, Inc.)



**Figure 6-1b** Size and characteristics of air-borne solids. (By permission from *Hoffman Handy Engineering Data*, Hoffmann Air and Filtration Systems, Inc.)

### 6.3 GUIDE TO DUST SEPARATOR APPLICATIONS

Table 6-5 [10] summarizes dry dust particle separators as to general application in industry; Table 6-6 and Figures 6-4 and 6-5 [11] compare basic collector characteristics. Figure 6-5 presents a typical summary of dust collection equipment efficiencies which have not changed significantly for many years except for specialized equipment to specialized applications.

### 6.4 GUIDE TO LIQUID-SOLID PARTICLE SEPARATORS

Table 6-7 summarizes liquid particle separators as to the general process-type application.

### 6.5 GRAVITY SETTLERS

The use of these settlers is not usually practical for most situations. The diameters or cross-sectional areas become too large for the handling of anything but the very smallest of flowing vapor streams. In general, gravity settlers of open box or tank design are not economical for particles smaller than 325 mesh or 43  $\mu$  [8]. They are much more practical for solids or dusts, although even for these situations the flow quantities must be small if the sizes are not to become excessive. With unusually heavy and/or large particles the gravity separator can be used to advantage.

**TABLE 6-1 Sizes of Common Dusts and Mists**

Dust or Mist	Average Particle Diameter, Microns ( $\mu$ )
Human Hair (for comparison)	50-200
Limit of visibility with naked eye	10-40
<i>Dusts</i>	
Atmospheric dust	0.5
Aluminum	2.2
Anthracite coal mining	
Breaker air	1.0
Mine air	0.9
Coal drilling	1.0
Coal loading	0.8
Rock drilling	1.0
Alkali fume	1-5
Ammonium Chloride fume	0.05-0.1-1
Catalyst (reformer)	0.5-50
Cement	0.5-40-55
Coal	5-10
Ferro-manganese, or silicon	0.1-1
Foundry air	1.2
Flour mill	15
Fly ash (Boiler flue gas)	0.1-3
Iron (Gray Iron Cupola)	0.1-10
Iron oxide (steel open hearth)	0.5-2
Lime (Limekiln)	1-50
Marble cutting	1.5
Pigments	0.2-2
Sandblasting	1.4
Silica	1-10
Smelter	0.1-100
Taconite iron ore (crushing and screening)	0.5-100
Talc	10
Talc milling	1.5
Tobacco smoke	0.2
Zinc oxide fume	0.05
Zinc (sprayed)	15
Zinc (condensed)	2
<i>Mists</i>	
Atmospheric fog	2-15
Sulfuric Acid	0.5-15

(Source: Compiled from [4-6].)

The fundamentals of separation for a particle moving with respect to a fluid are given by the drag coefficient of Figure 6-6.

The motion of particle and fluid are considered relative, and the handling of the relations is affected only by conditions of turbulence, eddy currents, and so on.

### 6.6 TERMINAL VELOCITY

When a particle falls under the influence of gravity it will accelerate until the frictional drag in the fluid balances the gravitational forces. At this point it will continue to fall at constant velocity. This is the terminal velocity or free-settling velocity. The general formulae for particle of any shape are as follows [5].

$$u_t = \sqrt{\frac{2g_L m_p (\rho_s - \rho)}{\rho \rho_s A_p C}} \quad (6-1)$$

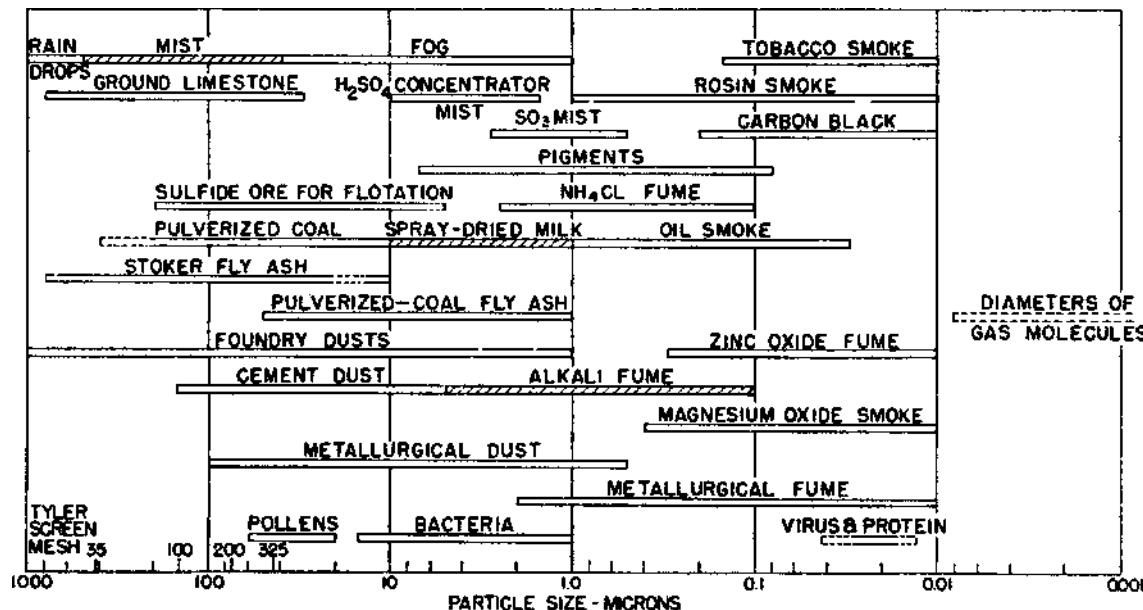


Figure 6-2 Particle size ranges for aerosols, dusts, and fumes. (Courtesy H.P. Munger, Battelle Memorial Institute.)

TABLE 6-2 Typical Dust Size Analysis

	Dust			
	Rock	Cement Kiln Exhaust	Foundry Sand	Limestone
Specific Gravity	2.63	2.76	2.243	2.64
Apparent Weight, lbs/ft <sup>3</sup>	61.3	52.0	45.9	72.0
<i>Screen Analysis (% passed)</i>				
100 Mesh	98.8	99.6	91.2	85.6
200 Mesh	92.8	92.2	78.4	76.4
325 Mesh	79.6	80.8	67.6	66.4
400 Mesh	70.8	73.2	64.4	63.2
<i>Elutriation Analysis (% under)</i>				
<i>Terminal Velocity</i>				
320 in./min	75.8	78.0	64.2	70.5
80 in./min	37.0	61.0	53.9	52.0
20 in./min	17.5	40.8	42.0	33.0
5 in./min	8.9	23.0	32.0	18.0

(Source: Compiled from Bulletin No. 1128, American Blower Corp., Detroit, MI.)

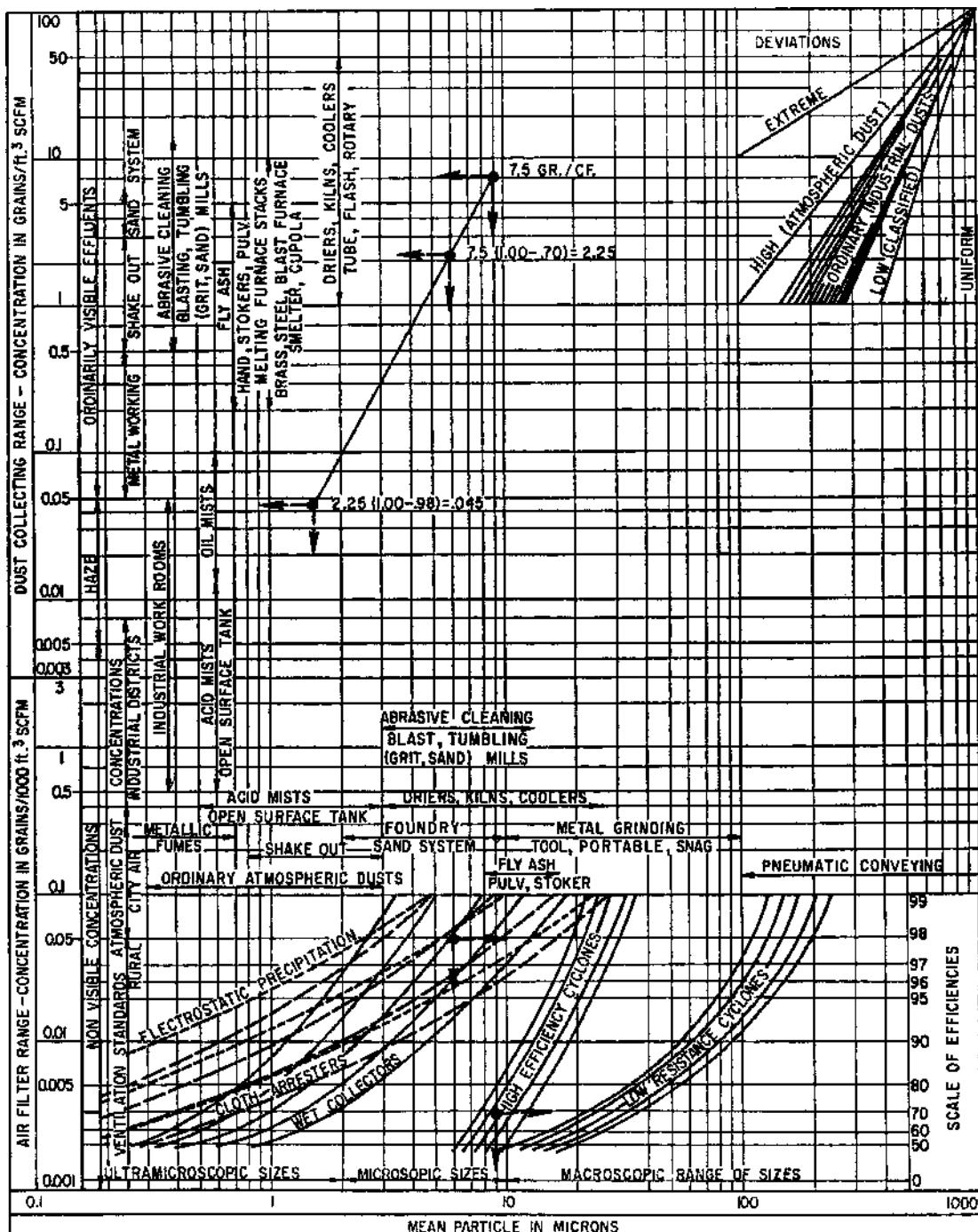
TABLE 6-4 Approximate Particle Sizes for Water Spray from Liquid Full Cone Spray Nozzles\*

Nozzle Size (in.)	Operating Pressure (psig)	Approximate Mean Particle Size ( $\mu$ )
1/2	15	1200
	60	750
3/4	10	1600
	40	1000
1	15	1750
	40	1250
1 1/2	15	2300
	60	1800
3	10	5300
	30	4300

\* Private communication, Spraying Systems Co., Bellwood, Ill.

TABLE 6-3 Dry Particle Screen Sizes

W.S. Tyler Screen Scale	Micron (approximate)
80	174
100	146
115	123
150	104
170	89
200	74
250	61
270	53
325	43



**Figure 6-3** Range of particle sizes, concentration, and collector performance. (Courtesy American Air Filter Co. Inc.)

where

$A_p$  = area of the particle projected on plane normal to direction of flow or motion,  $\text{ft}^2$

$C_d$  = overall drag coefficient, dimensionless

$C_d$  = overall drag coefficient, dimensionless

$m$  = mass of particle, lb m

$\rho$  = fluid density, lb/ft<sup>3</sup>

$\rho$  = particle density, lb/ft<sup>3</sup>

$u_t$  = terminal settling velocity determined by Stokes' Law of Particle under action of gravity, ft/s.

For spheres:

$$u_t = \sqrt{\frac{4g_L D_p (\rho_s - \rho)}{3\rho C}} \quad (6-2)$$

where  $D_p$  = diameter of particle, ft.

**TABLE 6-5 Applications of Dust Collections in Industry**

Operation	Concentration	Particle Sizes	Cyclone	High Efficient Centrifugal	Wet Collector	Fabric Arrester	Hi-Volt Electrostatic	See Remark No.
<i>Ceramics</i>								
a. Raw product handling	Light	Fine	Rare	Seldom	Frequent	Frequent	No	1
b. Fettling	Light	Fine to medium	Rare	Occasional	Frequent	Frequent	No	2
c. Refractory sizing	Heavy	Coarse	Seldom	Occasional	Frequent	Frequent	No	3
d. Glaze & Vitr. Enamel spray	Moderate	Medium	No	No	Usual	Occasional	No	
<i>Chemicals</i>								
a. Material handling	Light to moderate	Fine to medium	Occasional	Frequent	Frequent	Frequent	Rare	4
b. Crushing, grinding	Moderate to heavy	Fine to coarse	Often	Frequent	Frequent	Frequent	No	5
c. Pneumatic conveying	Very heavy	Coarse to fine	Usual	Occasional	Rare	Usual	No	6
d. Roasters, kilns, coolers	Heavy	Coarse to med-coarse	Occasional	Usual	Usual	Rare	Often	7
<i>Coal Mining and Power Plant</i>								
a. Material handling	Moderate	Medium	Rare	Occasional	Frequent	Frequent	No	8
b. Bunker ventilation	Moderate	Fine	Occasional	Frequent	Occasional	Frequent	No	9
c. Dedusting, air cleaning	Heavy	Med-coarse	Frequent	Frequent	Occasional	Often	No	10
d. Drying	Moderate	Fine	Rare	Occasional	Frequent	No	No	11
<i>Fly Ash</i>								
a. Coal burning-chain grate	Light	Fine	No	Rare	No	No	No	12
b. Coal burning-stoker fired	Moderate	Fine to coarse	Rare	Usual	No	No	Rare	
c. Coal burning-pulverized fuel	Heavy	Fine	Rare	Frequent	No	No	Frequent	13
d. Wood burning	Varies	Coarse	Occasional	Occasional	No	No	No	14
<i>Foundry</i>								
a. Shakeout	Light to moderate	Fine	Rare	Rare	Usual	Rare	No	15
b. Sand handling	Moderate	Fine to medium	Rare	Rare	Usual	Rare	No	16
c. Tumbling mills	Heavy	Med-coarse	No	No	Frequent	Frequent	No	17
d. Abrasive cleaning	Moderate to heavy	Fine to medium	No	Occasional	Frequent	Frequent	No	18
<i>Grain Elevator, Flour and Feed Mills</i>								
a. Grain handling	Light	Medium	Usual	Occasional	Rare	Frequent	No	19
b. Grain dryers	Light	Coarse	No	No	No	No	No	20
c. Flour dust	Moderate	Medium	Usual	Often	Occasional	Frequent	No	21
d. Feed mill	Moderate	Medium	Usual	Often	Occasional	Frequent	No	22
<i>Metal Melting</i>								
a. Steel blast furnace	Heavy	Varied	Frequent	Rare	Frequent	No	Frequent	23
b. Steel open hearth	Moderate	Fine to coarse	No	No	Doubtful	Possible	Probable	24
c. Steel electric furnace	Light	Fine	No	No	Considerable	Frequent	Rare	25
d. Ferrous cupola	Moderate	Varied	Rare	Rare	Frequent	Occasional	Occasional	26
e. Non-ferrous reverberatory	Varied	Fine	No	No	Rare	?	?	27
f. Non-ferrous crucible	Light	Fine	No	No	Rare	Occasional	?	28
<i>Metal Mining and Rock Products</i>								
a. Material handling	Moderate	Fine to medium	Rare	Occasional	Usual	considerable	?	29
b. Dryers, kilns	Moderate	Med-coarse	Frequent	Frequent	Frequent	Rare	Occasional	30
c. Cement rock dryer	Moderate	Fine to medium	Rare	Frequent	Occasional	No	Occasional	31
d. Cement kiln	Heavy	Fine to medium	Rare	Frequent	Rare	No	Considerable	32
e. Cement grinding	Moderate	Fine	Rare	Rare	No	Frequent	Rare	33
f. Cement clinker cooler	Moderate	Coarse	Occasional	Occasional	?	?	?	34
<i>Metal Working</i>								
a. Production grinding, scratch	Light	Coarse	Frequent	Frequent	Considerable	Considerable	No	35
b. Portable and swing frame	Light	Medium	Rare	Frequent	Frequent	Considerable	No	
c. Buffing	Light	Varied	Frequent	Rare	Frequent	Rare	No	36
d. Tool room	Light	Fine	Frequent	Frequent	Frequent	Frequent	No	37
e. Cast iron machining	Moderate	Varied	Rare	Frequent	Considerable	Considerable	No	38
<i>Pharmaceutical and Food Products</i>								
a. Mixers, grinders, weighing, blending, bagging, packaging	Light	Medium	Rare	Frequent	Frequent	Frequent	?	39
b. Coating pans	Varied	Fine to medium	Rare	Rare	Frequent	Frequent	No	40

(continued)

TABLE 6-5—(continued)

Operation	Concentration	Particle Sizes	Cyclone	High Efficient Centrifugal	Wet Collector	Fabric Arrester	Hi-Volt Electrostatic	See Remark No.
<i>Plastics</i>								
a. Raw material processing	(See comments under Chemicals)							41
b. Plastic finishing	Light to moderate	Varied	Frequent	Frequent	Frequent	Frequent	No	42
<i>Rubber Products</i>								
a. Mixers	Moderate	Fine	No	No	Frequent	Usual	No	43
b. Batchout rolls	Light	Fine	No	No	Usual	Frequent	No	44
c. Talc dusting and dedusting	Moderate	Medium	No	No	Frequent	Usual	No	45
d. Grinding	Moderate	Coarse	Often	Often	Frequent	Often	No	46
<i>Woodworking</i>								
a. Wood working machines	Moderate	Varied	Usual	Occasional	Rare	Frequent	No	47
b. Standing	Moderate	Fine	Frequent	Occasional	Occasional	Frequent	No	48
c. Waste conveying, hogs	Heavy	Varied	Usual	Rare	Occasional	Occasional	No	49

(Source: By permission from Kane, J. M. *Plant Engineering*, Nov, 1954.)

Remarks referred to in Table 6-5

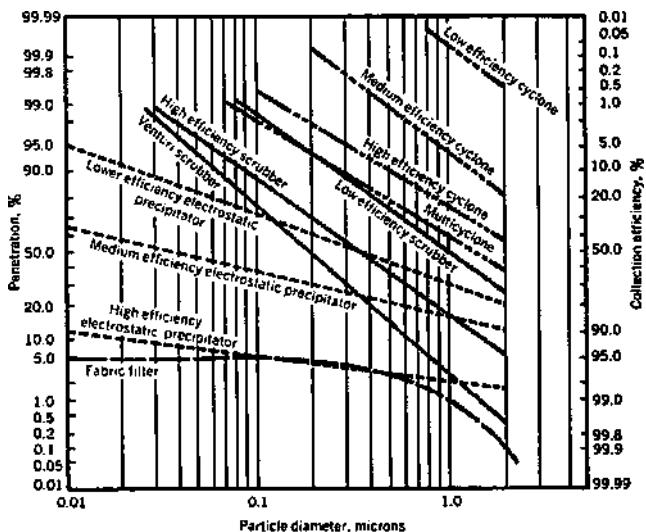
1. Dust released from bin filling, conveying, weighing, mixing, pressing, forming. Refractory products, dry pan and screening operations more severe.
2. Operations found in vitreous enameling, wall and floor tile, pottery.
3. Grinding wheel or abrasive cutoff operation. Dust abrasive.
4. Operations include conveying, elevating, mixing, screening, weighing, packaging. Category covers so many different materials that recommendation will vary widely.
5. Cyclone and high efficiency centrifugals often act as primary collectors followed by fabric or wet type.
6. Usual setup uses cyclone as product collector followed by fabric arrester for high overall collection efficiency.
7. Dust concentration determines need for dry centrifugal; plant location, product value determines need for final collectors. High temperatures are usual and corrosive gases not unusual.
8. Conveying, screening, crushing, unloading.
9. Remote from other dust producing points. Separate collector usually.
10. Heavy loading suggests final high efficiency collector for all except very remote locations.
11. Difficult problem but collectors will be used more frequently with air pollution emphasis.
12. Public nuisance from boiler-down indicates collectors are needed.
13. Higher efficiency of electrostatic indicated for large installations especially in residential locations. Often used in conjunction with dry centrifugal.
14. Public nuisance from sealed wood char indicates collectors are needed.
15. Hot gases and steam usually involved.
16. Steam from hot sand, adhesive clay bond involved.
17. Concentration very heavy at start of cycle.
18. Heaviest load from airless blasting due to higher cleaning speed. Abrasive shattering greater with sand than with grit or shot. Amounts removed greater with sand castings, less with forging scale removal, least when welding scale is removed.
19. Operations such as car unloading, conveying, weighing, storing.
20. Collection equipment expensive but public nuisance complaints becoming more frequent.
21. In addition to grain handling, cleaning, rolls, sifters, purifiers, conveyors, as well as storing, packaging operations are involved.
22. In addition to grain handling, bins, hammer mills, mixers, feeders, conveyors, bagging operations and control.
23. Primary dry trap and wet scrubbing usual. Electrostatic is added where maximum cleaning required.
24. Cleaning equipment seldom installed in past. Air pollution emphasis indicates collector use will be more frequent in future.
25. Where visible plume objectionable from air pollution standards, use of fabric arresters with greater frequency seems probable.
26. Most cupolas still have no collectors but air pollution and public nuisance emphasis is creating greater interest in control equipment.
27. Zinc oxide loading heavy during zinc additions. Stack temperatures high.
28. Zinc oxide plume can be troublesome in certain plant locations.
29. Crushing, screening, conveying, storing involved. Wet ores often introduce water vapor in exhaust air stream.
30. Dry centrifugals used as primary collectors, followed by final cleaner.
31. Collection equipment installed primarily to prevent public nuisance.
32. Collection usually permit salvage of material and also reduce nuisance from settled dust in plant area.
33. Salvage value of collected material high. Same equipment used on raw grinding before calcining.
34. Coarse abrasive particles readily removed in primary collector types.
35. Roof discoloration, deposition on autos can occur with cyclones and less frequently with dry centrifugal. Heavy duty air filters sometimes used as final cleaners.
36. Linty particles and sticky buffering compounds can cause trouble in high efficiency centrifugals and fabric arresters. Fire hazard is also often present.
37. Unit collectors extensively used, especially for isolated machine tools.
38. Dust ranges from chips to fine floats including graphitic carbon.
39. Materials involved vary widely. Collector selection may depend on salvage value, toxicity, sanitation yardsticks.
40. Controlled temperature and humidity of supplied air to coating pans makes recirculation from coating pans desirable.
41. Manufacture of plastic compounds involve operations allied to many in chemical field and vary with the basic process employed.
42. Operations are similar to woodworking and collection selection involves similar considerations. See Item 13.
43. Concentration is heavy during feed operation. Carbon black and other fine additions make collection and dust free disposal difficult.
44. Often no collection equipment is used where dispersion from exhaust stack is good and stack location favorable.
45. Salvage of collected material often dictates type of high efficiency collector.
46. Fire hazard from some operations must be considered.
47. Bulky material. Storage for collected material is considerable, bridging from splitters and chips can be a problem.
48. Production sanding produces heavy concentration of particles too fine to be effectively caught by cyclones or dry centrifugals.
49. Primary collector invariably indicated with concentration and particle size range involved, wet or fabric collectors when used are employed as final collectors.

**TABLE 6-6 Comparison of Some Important Dust Collector Characteristics**

Type	Higher Efficiency Range on Particles Greater than Mean Size in Microns	Pressure Loss (Inches Water)	Water (Gal Per 1000 cfm)	Space	Sensitivity To cfm Change		Humid Air Influence	Max Temp. (° F)(Standard Construction)
					Pressure	Efficiency		
Electro-Static	0.25	1/2	—	Large	Negligible	Yes	Improves efficiency	500
<i>Fabric</i>								
Conventional	0.4	3-6	—	Large	As cfm	Negligible	May make reconditioning difficult	180
Reverse Jet Wet	0.25	3-8	—	Moderate	As cfm	Negligible		200
Packed Tower	1-5	1 1/2-31/2	5-10	Large	As cfm	Yes	None	Unlimited
Wet Centrifugal	1-5	2 1/2-6	3-5	Moderate	As cfm <sup>2</sup>	Yes		
Wet Dynamic	1-2	Note 1	1/2-1	Small	Note 1	No		
Orifice Types	1-5	2 1/2-6	10-40	Small	As cfm or less	Varies with design		
<i>Higher Efficiency</i>								
Nozzle	0.5-5	2-4	5-10	Moderate	As cfm <sup>2</sup>	Slightly to Moderately	None	Note 2 Unlimited
Venturi	0.5-2	12-20	—	Small				
<i>Dry Centrifugal</i>								
Low Pressure Cycle	20-40	3/4-1 1/2	—	Large	As cfm <sup>2</sup>	Yes	May cause Condensation and plugging	750
High Efficient Centrifugal	10-30	3-6	—	Moderate	As cfm <sup>2</sup>	Yes		750
Dry Dynamic	10-20	Note 1	—	Small	Note 1	No		750
Louver	15-60	1-3	—	Small	As cfm <sup>2</sup>	Moderately		750

(Source: By permission from John M. Kane, "Operation, Application and Effectiveness of Dust Collection Equipment," *Heating and Ventilating*, Aug 1952, [10].)

Note: 1. A function of the mechanical efficiency of these combined exhauster and dust collectors. 2. Precooling of high temperature gases will be necessary to prevent rapid evaporation of fine droplets.



**Figure 6-4** Comparison chart showing ranges of performance of several collection/control devices in air streams. (By permission from Vandergrift et al. *Chemical Engineering*, Deskbook Issue, Jun 18, 1973, p. 109.)

(a) For spherical particles between 1500 and 100,000  $\mu$ ; by Newton's Law (with  $C = 0.445$ , average drag coefficient)

$$u_t = 1.74 \sqrt{\frac{g_L D_p (\rho_s - \rho)}{\rho}} \quad (6-3)$$

(b) For spherical particles between 100 and 1500  $\mu$  (with  $C = 18.5 N_{Re}^{-0.6}$  (see Figure 6-1)) [5],

$$u_t = \frac{0.153 g_L^{0.71} D_p^{1.14} (\rho_s - \rho)^{0.7}}{\rho^{0.29} \mu^{0.43}} \quad (6-4)$$

where

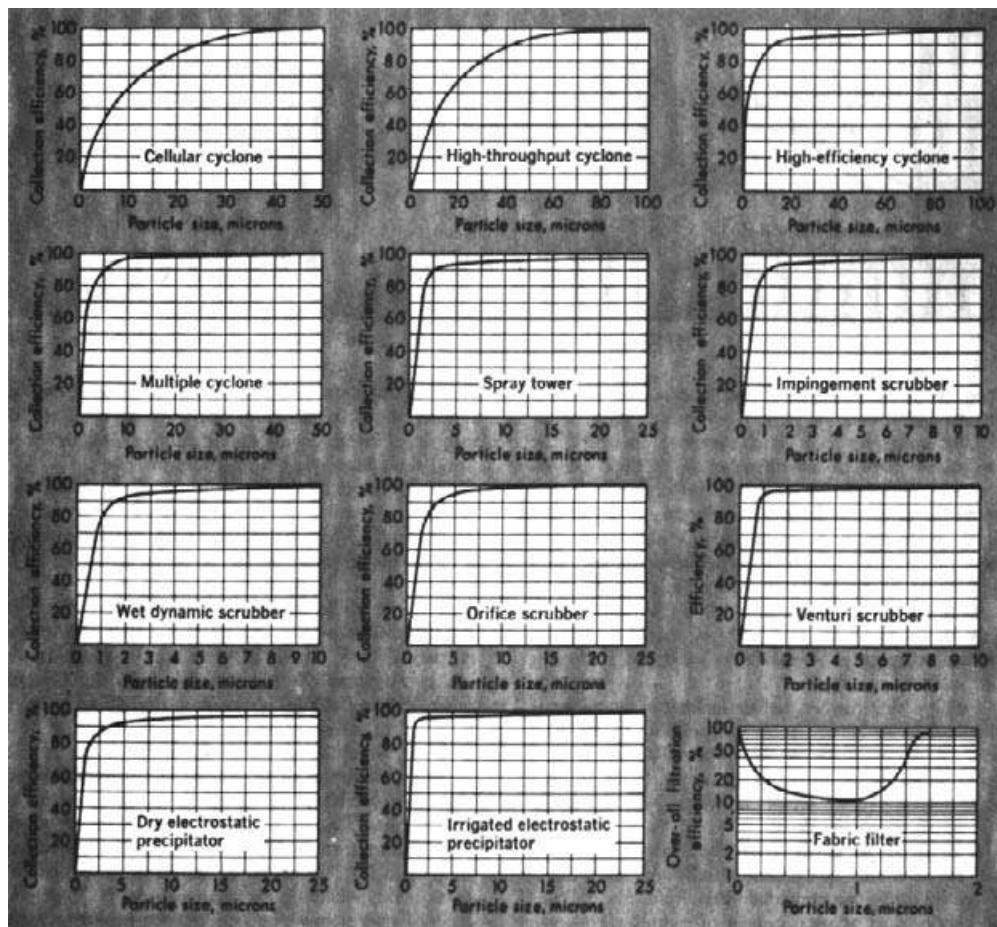
$\mu$  = fluid viscosity, cP ( $1 \text{ cP} = 6.72 \times 10^{-4} \text{ lb/ft.s}$ ).

(c) For spherical particles between 3 and 100  $\mu$  and Reynolds numbers between 0.0001 and 2.0, by Stokes' Law

with

$$C = 24 N_{Re}^{-1}$$

$$F_d = 3\pi\mu u D_p / g_L$$



**Figure 6-5** Efficiency curves for various types of dust collection equipment as of 1969. Only marginal improvements have been made since then. (By permission from Sargent, G.D., *Chemical Engineering*, Jan 27, 1969, p. 130.)

TABLE 6-7 General Applications of Liquid Particle Separators

Operation	Concentration	Particle Size	Gravity	Collector Types			
				Impingement	Cyclone	Scrubbers	Electrical
Pipeline entrained liquid	Light	Fine to coarse	No	Frequent	Yes	Occasional	Few
Compressor discharge liquid	Light	Fine	No	Frequent	Occasional	Occasional	Rare
Compressor oil haze	Vary light	Very fine	No	Frequent	Frequent	Frequent	Occasional
Flashing liquid	Light to medium	Fine to medium	No	Frequent	Frequent	Occasional	Rare
Boiling or bubbling	Light to heavy	Fine to coarse	Occasional	Frequent	Frequent	Occasional	Rare
Spraying	Light to heavy	Fine to coarse	No	Frequent	Frequent	Rare	Rare
Corrosive liquid particles	Light to heavy	Fine to heavy	Occasional	Frequent	Occasional	Frequent	Rare
Liquid plus solid particles	Light to heavy	Medium	Occasional	Occasional	Frequent	Frequent	Occasional

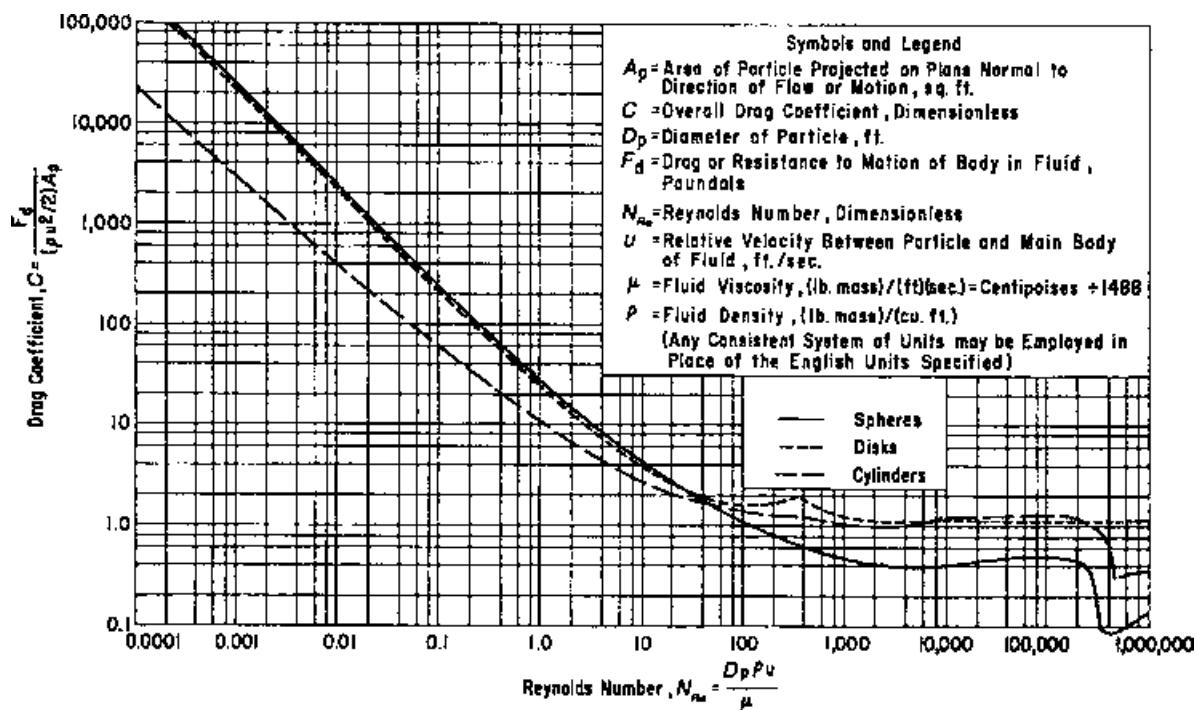


Figure 6-6 Drag coefficients for spheres, disks, and cylinders in any fluid. (By permission from Perry, J.H., ed., *Chemical Engineering Handbook*, 3rd ed., McGraw-Hill Company, 1950.)

where

$u$  = relative velocity particle and main body of fluid, ft/s.

$$u_t = g_L D_p^2 \frac{(\rho_s - \rho)}{18\mu} \quad (6-5)$$

For particles smaller than  $0.1 \mu$  the random Brownian motion is greater than the motion due to gravitational settling. Therefore the above relations based on Stokes' Law will not hold.

(d) For spherical particles between  $0.1$  and  $3 \mu$ , by Stokes-Cunningham Law [12]

$$u_t = K_m u_{ts} \quad (6-6)$$

where

$u_{ts}$  = terminal settling velocity determined by Stokes' Law, ft/s  
 $K_m$  = Stokes-Cunningham correction factor, dimensionless

**TABLE 6-8 Values of  $K_m$  for Air at Atmospheric Pressure [12]**

Particles diameter, $D_p$ ( $\mu$ )	70° F	212° F	500° F
0.1	2.8	3.61	5.14
0.25	1.682	1.952	2.528
0.5	1.325	1.446	1.711
1.0	1.160	1.217	1.338
2.5	1.064	1.087	1.133
5.0	1.032	1.043	1.067
10.0	1.016	1.022	1.033

$m$  = Exponent given by equations in Reynolds number table below

$V_t$  = Settling velocity for single spherical particle, ft/s and m/s (terminal)

$V_{ts}$  = Settling velocity for hindered uniform spherical particle, ft/s or m/s (terminal)

$c$  = Volume fraction solids

$K$  = Constant given by equation above

$N_{Re}$  = Reynolds number,  $D_p V_t \rho_f / \mu$

Values of $m$	$N_{Re}$
4.65	< 0.5
$4.375(N_{Re})^{-0.0875}$	$0.5 \leq N_{Re} \leq 1300$
2.33	$N_{Re} > 1300$

$\mu_{ts}$  = terminal settling velocity, as calculated from Stokes' Law, ft/s.

$$K_m = 1 + K_{me} (\lambda_m / D_p) \quad (6-7)$$

$$K_{me} = 1.64 + 0.552 e^{-(0.656 D_p / \lambda_m)} \quad (6-8)$$

This represents a correction on Stokes' Law and is significant for  $3\mu$  and smaller particles in gases and  $0.01\mu$  and smaller particles in liquids. Table 6-8 gives values of  $K_m$ .

When two free-settling particles of different dimensions,  $D'_{p1}$  and  $D'_{p2}$ , and different densities,  $\rho_{p1}$  and  $\rho_{p2}$ , fall through a fluid of density  $\rho_f$ , they will attain equal velocities when

$$\frac{D'_{p1}}{D'_{p2}} = \left( \frac{\rho_{p2} - \rho_f}{\rho_{p1} - \rho_f} \right)^n \quad (6-9)$$

where

$n = 1$  in eddy resistance zone (more turbulent) and  $n = 0.5$  in streamline fall.

## 6.7 ALTERNATE TERMINAL VELOCITY CALCULATION

In contrast to individual particles settling in a very dilute solution/fluid is the case of sedimentation where particles must settle in more concentrated environment, and hence particles influence adjacent particles. This is often termed "hindered settling" [8, 13]. Depending upon the particles concentration, the hindered terminal settling velocity will generally be somewhat lower than for the terminal settling velocity of a single desired particle in the same medium.

According to Carpenter,  $K$  is a dimensionless number that establishes the regime of settling class, reference to the settling laws [13]:

$$K = 34.8 D'_p \left[ \frac{\rho_f (\rho_p - \rho_f)}{\mu^2} \right]^{1/3} \quad (6-10)$$

If  $K < 3.3$ , then Stokes' law applies; if  $3.3 \leq K \leq 43.6$ , the intermediate law applies; if  $K > 43.6$ , Newton's law applies; and if  $K > 2360$ , the equations should not be used. The corresponding  $b_1$  and  $a$  values for these  $K$  values are tabulated below.

Law	Range	$b_1$	$a$
Stokes'	$K < 3.3$	24.0	1.0
Intermediate	$3.3 \leq K \leq 43.6$	18.5	0.6
Newton's	$K > 43.6$	0.44	0

The terminal settling velocity for single spheres can be determined using the contrasts for the flow regime.

$$V_t = \left[ \frac{4a_e D_p^{(1+n)} (\rho_p - \rho_f)}{3b_1 \mu^n \rho_f^{(1-n)}} \right]^{1/(2-n)}, \text{ ft/s} \quad (6-11)$$

For hindered particle settling in a "more crowded" environment, using spherical particles of uniform size,

$$V_{ts} = V_t (1 - c)^m, \text{ ft/s} \quad (6-12)$$

where

$c$  = volume fraction solids

$D'_p$  = diameter of particle, in. or mm

$a_e$  = acceleration due to gravity, 32.2 ft/s<sup>2</sup> or 9.8 m/s<sup>2</sup>

$\rho_p$  = density of particles, lb/ft<sup>3</sup> or kg/m<sup>3</sup>

$\rho_f$  = density of fluid, lb/ft<sup>3</sup> or kg/m<sup>3</sup>

$\mu$  = viscosity of fluid, cP

$b_1$  = constant given above

$K$  = constant given by equation above

$m$  = exponent given by equations in Reynolds number table below

$n$  = constant given in text

$V_t$  = settling velocity for single spherical particle, ft/s and m/s (terminal)

$V_{ts}$  = settling velocity for hindered uniform spherical particle, ft/s or m/s (terminal).

Referring the above to other than uniform spherical particles does not create a significant loss in accuracy for industrial applications. For higher concentration, the values of  $V_{ts}$  are lower than  $V_t$ . In large particles in small vessels, the wall effect can become significant [8].

For a single particle,  $D_p$  can be taken as 2 (hydraulic radius) and as the Sauter mean diameter for hindered particles.

Reynolds number  $N_{Re}$ :

$$N_{Re} = \frac{D_p V_t \rho_f}{\mu}, \text{ dimensionless} \quad (6-13)$$

Values of $m$	$N_{Re}$
4.65	< 0.5
$4.375(N_{Re})^{-0.0875}$	$0.5 \leq N_{Re} \leq 1300$
2.33	$N_{Re} > 1300$

**EXAMPLE 6-2****Hindered Settling Velocities**

Using the example of Carpenter [13]:

$$\rho_f = \text{fluid density} = 0.08 \text{ lb/ft}^3$$

$$\mu = \text{viscosity} = 0.02 \text{ cP}$$

$$\rho_p = 500 \text{ lb/ft}^3$$

$$D_p' = \text{particle diameter} = 0.01 \text{ in.}$$

$$c = \text{volume fraction solids, } 0.1.$$

Solving equation for  $K$ , for unhindered particle,

$$K = 34.81(0.01) \left[ \frac{0.08(500 - 0.08)}{(0.02)^2} \right]^{1/3}$$

$$K = 16.16.$$

Then, for  $K = 16.16$  (intermediate range),  $b = 18.5$ ;  $n = 0.6$ .

Solving Eq. (6-11) for settling velocity,  $V_t$ ,

$$V_t = \left[ \frac{4(32.2)(0.01)^{(1+0.6)}(500 - 0.08)}{3(18.5)(0.02)^{0.6}(0.08)^{(1-0.6)}} \right]^{1/(2-0.6)} \quad V_t = 8.81 \text{ ft/s}$$

Reynolds number,

$$N_{Re} = \frac{D_p V_t \rho_f}{\mu}$$

$$= \left( \left[ \frac{0.01}{(12)} \right] \frac{(8.81)(0.08)}{(0.02)(6.72 \times 10^{-4})} \right) \left( \text{in.} \cdot \frac{\text{ft}}{\text{in.}} \cdot \frac{\text{ft}}{\text{s}} \cdot \frac{\text{lb}}{\text{ft}^3} \cdot \frac{1}{\frac{\text{lb}}{\text{ft s}}} \right)$$

$$= 43.70$$

$$\mu = (\text{cP})(6.72 \times 10^{-4}), \text{lb/ft s}$$

$$\text{Then, } m = 4.375(N_{Re})^{-0.0875} = 4.375(43.70)^{-0.0875} = 3.1437$$

For 0.1 volume fraction solids the hindered settling velocity is as follows:

$$V_{ts} = V_t(1 - c)^m = 8.81(1 - 0.1)^{3.1437} = 6.33 \text{ ft/s}$$

(e) Particles under 0.1 micron Brownian movement becomes appreciable for particles under  $3\mu$  and predominates when the particle size reaches  $0.1\mu$  [5]. This motion usually has little effect in the average industrial process settling system except for the very fine fogs and dusts. However, this does not mean that problems are not present in special applications.

Figure 6-1a gives the limiting or critical diameter above which the particular settling law is not applicable. Figure 6-7 gives terminal velocities for solid particles falling in standard air ( $70^\circ\text{F}$  and 14.7 psia), and Figure 6-8 gives terminal velocities of particles falling through water. If a particle (liquid or solid) is falling under the influence of gravity through a vapor stream, the particle will continue to fall until or unless the vapor flow rate is increased up to or beyond the terminal velocity value of the particle. If the vapor velocity exceeds this, then the particle will be carried along with the vapor (entrained).

**PRESSURE DROP**

Pressure drop through gravity settlers is usually extremely low due to the very nature of the method of handling the problem.

Figure 6-9 is convenient for quick checks of terminal settling velocities of spherical particles in air and in water [8].

## 6.8 AMERICAN PETROLEUM INSTITUTE'S OIL FIELD SEPARATORS

The American Petroleum Institute (API) Manual on Disposal of Refinery Wastes provides specific design and construction standards for API-separators that are often used in oilfield waste disposal [14].

## LIQUID/LIQUID, LIQUID/SOLID GRAVITY SEPARATIONS, DECANTERS, AND SEDIMENTATION EQUIPMENT

**Lamella Plate Clarifier.** This angle plate gravity separator (Figure 6-10) removes suspensions of solids from a dilute liquid.

The unit is more compact than a box-type settler due to the increased capacity achieved by the multiple parallel plates. The concept is fairly standard (US Patent 1458,805—year 1923) but there are variations in some details. For effective operation, the unit must receive the mixture with definite particles having a settling velocity. The units are not totally effective for flocculants or coagulated masses that may have a tendency to be buoyant.

Flow through the plates must be laminar, and the critical internal areas are [15–17] as follows.

- distribution of the inlet flow
- flow between each parallel plate surface
- collection of clarified water

Because these units are essentially open, the “standard” design does not fit into a closed process system without adding some enclosures, and certainly is not suitable for a pressurized condition. Although the main suspended flow is through a top mixing chamber, the mixture flows around the angled ( $\text{avg } 55^\circ$ ) parallel plate enclosure and begins its settling path from the bottom of the plates, flowing upward while depositing solids that slide countercurrently into the bottom outlet. The purified liquid flows overhead and out into the top collecting trough.

Manufacturers should be contacted for size/rating information, because the efficiency of a design requires proper physical property information as well as system capacity and corrosion characteristics.

**THICKENERS AND SETTLERS**

Generally, large volume units for dewatering, settling of suspensions, thickening and concentration of solids, and clarification must be designed by the specific manufacturer for the process conditions and physical properties. Some typical processes involving this class of equipment are lime slurring, ore slurring, ore clarifying, wastewater clarification, and thickening. A good summary discussion is given by Fitch [18].

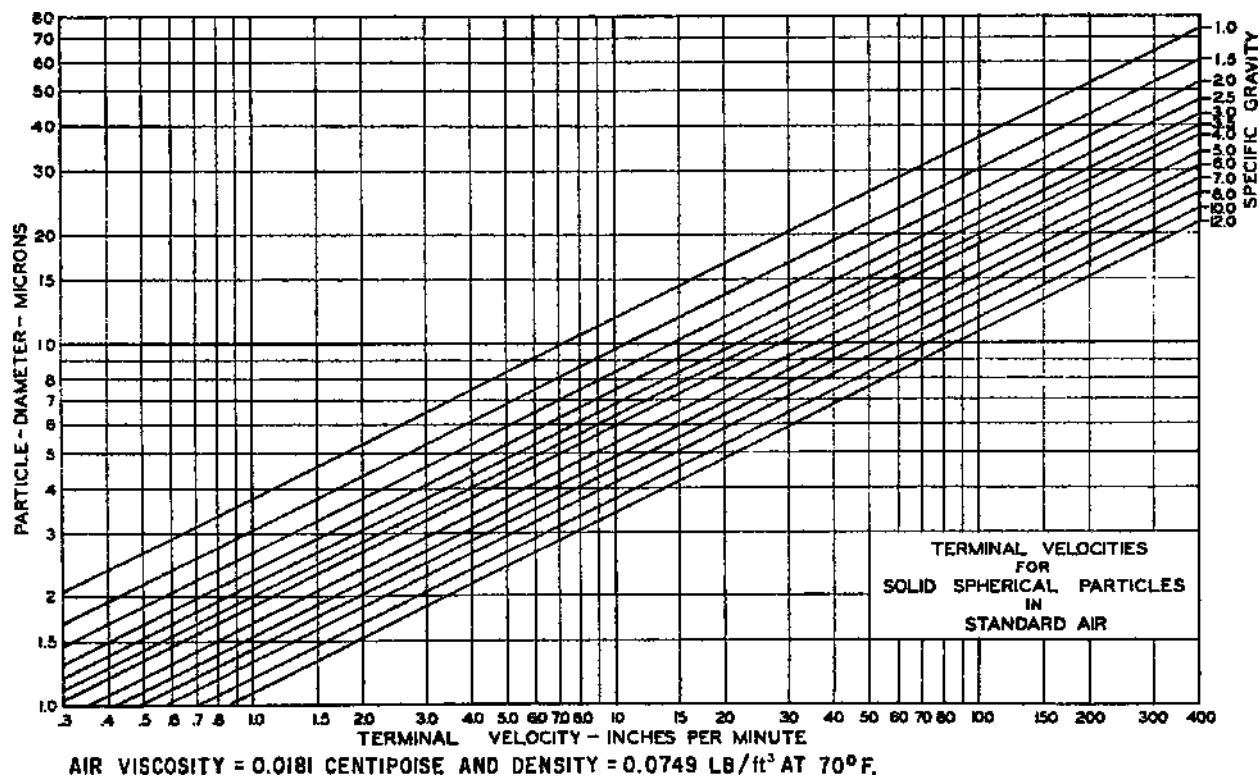


Figure 6-7 Terminal velocities of solid particles in standard air. (Courtesy of American Blower Div. American Radiator and Standard Sanitary Corporation.)

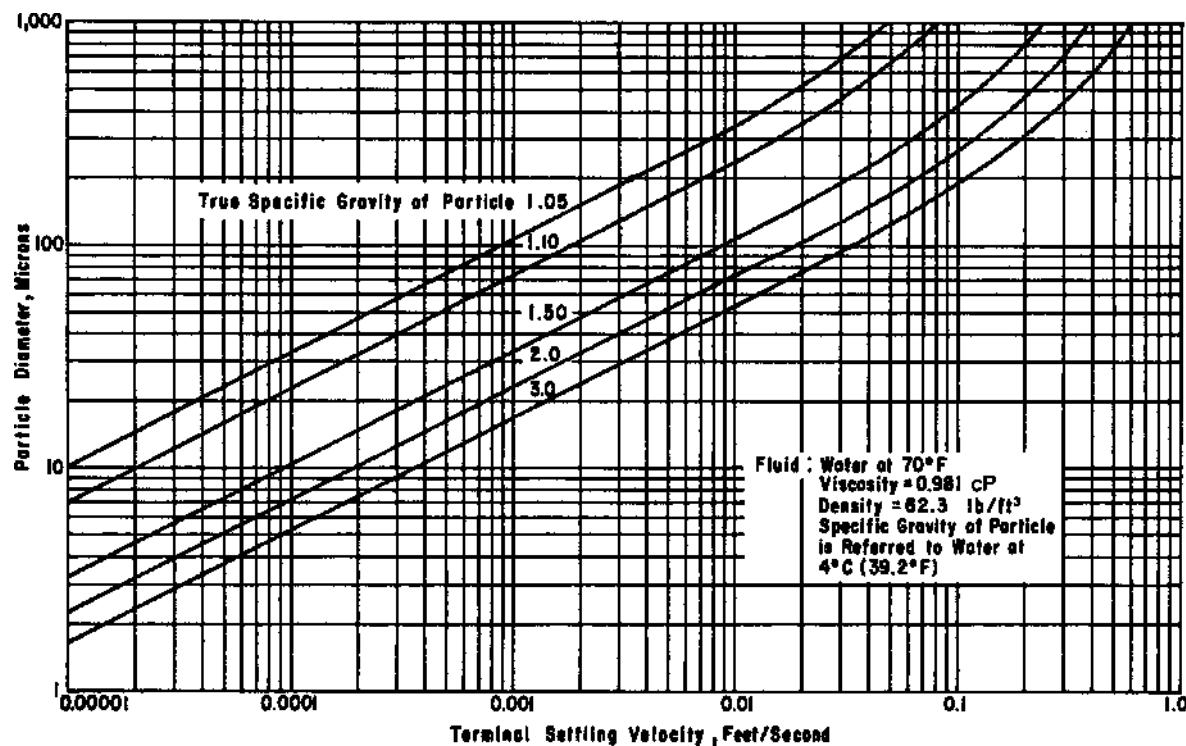
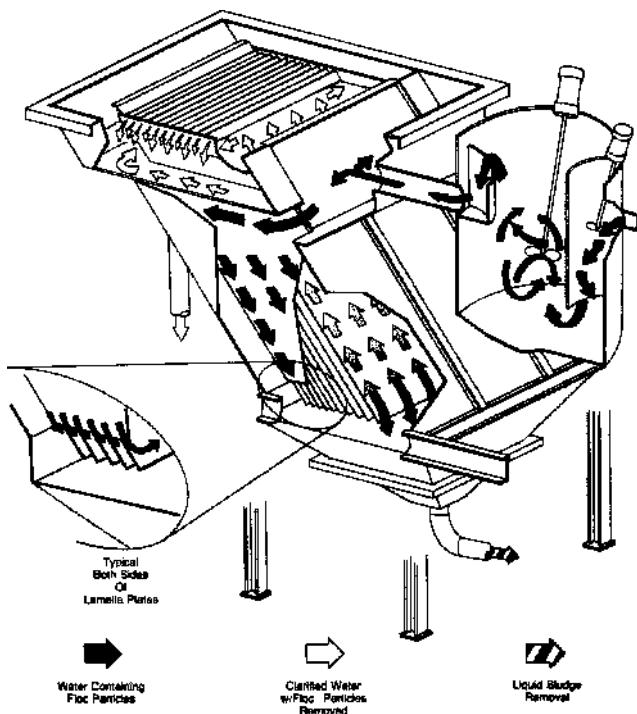
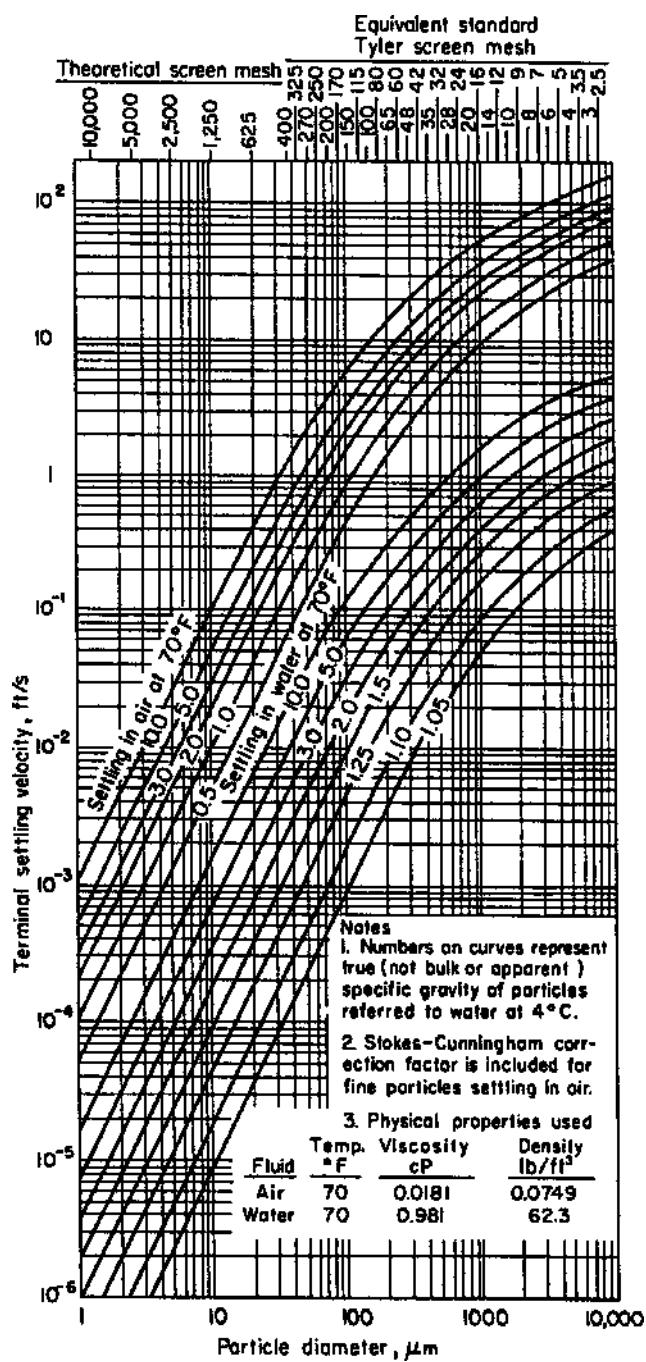


Figure 6-8 Terminal settling velocity of particles in water. (By permission from Lapple, C.E., *Fluid and Particle Mechanics*, 1st ed., Univ. of Delaware, Newark, 1954.)



**Figure 6-10** Lamella Clarifier. (By permission from Graver Water Div. of the Graver Co.)

the aqueous phase into the hydrocarbon and vice versa. Horizontal units are usually best for settling and possibly vented units for decantation (but not always).

Residence time of the mixture in the vessel is a function of the separation or settling rate of the heavier phase droplets through the lighter phase. Most systems work satisfactorily with a 30 min to 1 h residence time, but this can be calculated [19]. After calculation, give a reasonable margin of extra capacity to allow for variations in process feed rate and in the mixture phase composition.

From Stokes' Law, the terminal settling velocity is as follows.

$$V_t = \frac{g D_p^2 (\rho_p - \rho)}{18 \mu}, \text{ ft/s} \quad (6-14)$$

where

$\mu$  = viscosity of surrounding fluid, lb/ft s

$D_p$  = diameter of particle, ft.

For assumed spherical particles in a surfactant-free system [19], the minimum particle diameter for many fine dispersions is  $100\text{ }\mu$ ; however, Hooper and Jacobs [20] have reviewed a wide variety of liquid drop data and suggest that a good choice is  $150\text{ }\mu$  or  $0.15\text{ cm}$  or  $0.0005\text{ ft}$ . This is also the particle size used in the API Design Manual [14]. Using too large an assumed particle diameter will cause the settler unit to become unreasonably small.

Assuming a horizontal unit, as illustrated in Figure 6-11, has a segment of a circle equal to 25–75% [18, 19] of the circular area (the highest of this segment will be about 30–70% of the diameter), then height of the interface will be as follows [19].

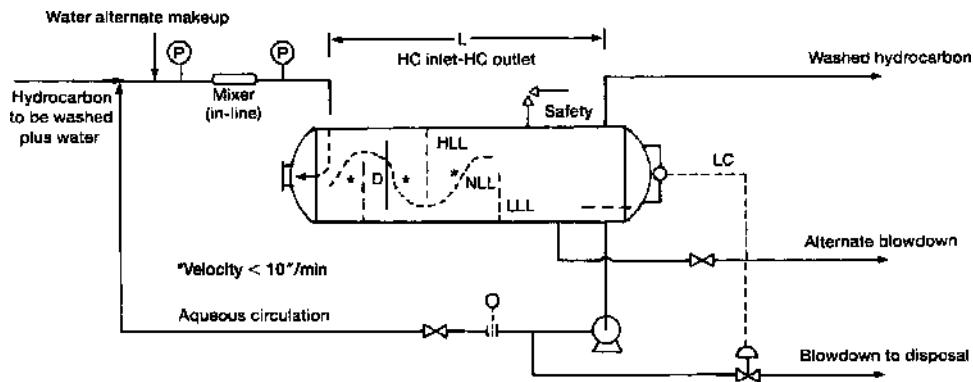
$$H/D = \frac{0.8A}{(\pi D^2/4)} + 0.1 \text{ ft} \quad (6-15)$$

or

$$h_c = 38.4A/(\pi D) + 1.2D \text{ in.} \quad (6-15a)$$

## HORIZONTAL GRAVITY SETTLERS OR DECANTERS, LIQUID/LIQUID

Many processes require the separation of immiscible liquid/liquid streams; that is, water/hydrocarbon. The settling unit must be of sufficient height (diameter) and length to prevent entrainment of



**Figure 6-11** Settler vessel; runs full. (Adapted by permission from Abernathy, M.W., *Encyclopedia of Chemical Processing and Design*, Vol. 25, J.J. McKetta, ed., Marcel Dekker, 1987, p. 77 and *Hydroc. Proc.*, Sep 1977, p. 199 [15] and Ludwig [21].)

where

$H$  = height of segment of a circle, ft

$h_c$  = height of segment of a circle, in.

$D$  = diameter of vessel, ft

$A$  = area of segment of circle,  $\text{ft}^2$ .

The average volumetric residence time in the settler is

$$t_{\text{avg}} = 7.48(V_{\text{set}}/F) \text{ min} \quad (6-16)$$

where

$V_{\text{set}}$  = active volume of settler occupied by one of the phases,  $\text{ft}^3$

$F$  = flow rate of one phase, gpm

$t_{\text{avg}}$  = average residence time based on liquid flow rate and vessel volume, min.

The minimum residence time as determined by Stokes' Law terminal settling velocity is given as follows:

$$t_{\text{min}} = h_c/v_t, \text{ min} \quad (6-17)$$

where

$h_c$  = height of segment of circle, in.

$v_t$  = terminal settling velocity of a particle under the action of gravity, ft/s.

Average residence time related to minimum residence time is

$$t_{\text{ave}} = (f)(t_{\text{min}}) \quad (6-18)$$

where

$f$  = factor relating average velocity to maximum velocity.

This relationship is related to the viscosities of the hydrocarbon and aqueous phases at the interface. Based on data from different systems:

$$f = 2.0 \text{ (use for design)}$$

The active volume occupied by either phase is

$$V = AL, \text{ ft}^3 \quad (6-19)$$

$L$  = length of vessel, ft, inlet to outlet

$$h_c = 7.48ALv_t/(fF) \quad (6-20)$$

For an aqueous hydrocarbon or organic solvent mixture, the top layer will be hydrocarbon, with the aqueous layer droplets settling through the hydrocarbon. The terminal velocity is

$$v_{\text{hc}} = 12.86(\Delta s \text{ gr})/\mu_{\text{hc}}, \text{ in./ min} \quad (6-21)$$

where

$v_{\text{hc}}$  = terminal settling velocity of aqueous droplets in hydrocarbon phase in the top of the vessel, in./min

$\Delta s \text{ gr}$  = differences in specific gravity of the particle and surrounding fluid

$\mu_{\text{hc}}$  = viscosity of surrounding fluid, cP.

*Height of hydrocarbon layer to the interface:*

$$h_t = (7.48) \frac{(A_t L v_{\text{hc}})}{(f_{\text{hc}} F_{\text{hc}})} \quad (6-21a)$$

where

$F_{\text{hc}}$  = hydrocarbon phase flow rate, gpm.

Also,

$$h_t = 38.4 A_t / (\pi D) + 1.2 D \quad (6-21b)$$

where

$h_t$  = height of continuous hydrocarbon phase in the top of the vessel, in.

then,

$$A_t = \frac{1.2 D}{\left[ \frac{7.48 L v_{\text{hc}}}{(f_{\text{hc}} F_{\text{hc}})} - \frac{38.4}{(\pi D)} \right]} \quad (6-22)$$

where

$A_t$  = cross-sectional area at the top of the vessel occupied by the continuous hydrocarbon phase,  $\text{ft}^2$

$A_b$  = cross-sectional area at the bottom of the vessel occupied by continuous aqueous phase,  $\text{ft}^2$ .

*For the bottom aqueous phase:* hydrocarbon droplets settle out of the continuous aqueous phase. The terminal settling velocity for hydrocarbon droplets is

$$v_{\text{aq}} = 12.86(\Delta s \text{ gr})/\mu_{\text{aq}}, \text{ in./ min} \quad (6-23)$$

where

$v_{\text{aq}}$  = terminal settling velocity of hydrocarbon droplets in aqueous phase in the bottom of the vessel, in./min

$\mu_{\text{aq}}$  = viscosity of aqueous phase, cP.

*Height of aqueous layer to the interface:*

$$h_b = (7.48)(A_b Lv_{aq})/(f_{aq} F_{aq}) \quad (6-23a)$$

Also,

$$h_b = 38.4 A_b / (\pi D) + 1.2 D \quad (6-23b)$$

where

$h_b$  = height of continuous aqueous phase in the bottom of the vessel, in.

$A_b$  = cross-sectional area at the bottom of the vessel occupied by continuous aqueous phase,  $\text{ft}^2$

$f_{aq}$  = factor relating aqueous phase average velocity to maximum velocity

$F_{aq}$  = aqueous phase flow rate, gpm

$L$  = length of vessel from hydrocarbon inlet to hydrocarbon outlet, ft

$v_{aq}$  = terminal settling velocity of hydrocarbon droplets in aqueous phase in the bottom of the vessel, in./min.

$$A_b = \frac{1.2 D}{\left[ \frac{7.48 L v_{aq}}{(f_{aq} F_{aq})} - \frac{38.4}{(\pi D)} \right]} \quad (6-23c)$$

*Optimum vessel diameter:*

Assume 20% cross-sectional area is occupied by an emulsion and is recognized as a "dead volume." This is actually the height over which the interface level will vary during normal operations [19].

$$A_t + A_b = 0.8\pi D^2/4 \quad (6-24)$$

where

$A_t$  = cross-sectional area at the top of the vessel occupied by continuous hydrocarbon phase,  $\text{ft}^2$

$A_b$  = cross-sectional area at the bottom of the vessel occupied by continuous aqueous phase,  $\text{ft}^2$

$D$  = diameter of vessel, ft.

$$D = \pm \left[ \frac{a}{2} \pm \frac{\sqrt{(a^2 - 4b)}}{2} \right]^{0.5}, \text{ ft} \quad (6-25)$$

where

$$a = (1.889)(v_{hc} f_{aq} F_{aq} + v_{aq} f_{hc} F_{hc}) / (r v_{hc} v_{aq}) \quad (6-26)$$

$$b = (3.505)(f_{hc} F_{hc} f_{aq} F_{aq}) / (r^2 v_{hc} v_{aq}) \quad (6-27)$$

where

$r$  = economical vessel radius ( $= L/D$ ), ft

$v_{hc}$  = terminal settling velocity of aqueous droplets in hydrocarbon phase in the top of the vessel in./min.

## 6.9 MODIFIED METHOD OF HAPPEL AND JORDAN [22]

This method is a modification of the earlier method [23] by Abernathy [19], as follows, and can be less conservative [19] than the original method [23]. A basic assumption is that particles must rise/fall through one-half of the drum vertical cross-sectional area [19].

$$t = h/v$$

$$t = (1/2)(7.48)[0.8\pi D^2 L/4]F_t \quad (6-28)$$

where

$D$  = diameter of vessel, ft

$L$  = length of vessel from hydrocarbon inlet to hydrocarbon outlet, ft

$F_t$  = total flow rate of both phases, gpm

$v_t$  = terminal settling velocity of a particle under the action of gravity, ft/s.

This assumes 20% of the cross-sectional even as "dead volume." The height from the interface can be determined by combining the above equations:

$$h = (0.748)\pi D^2 Lv/F_t \quad (6-29)$$

The height for each interface is

$$h_t = (0.748)\pi D^2 L h_{hc}/F_t \quad (6-30)$$

where

$h_t$  = height of continuous hydrocarbon phase in the top of the vessel, in.

$$h_b = (0.748)\pi D^2 Lv_{aq}/F_t \quad (6-31)$$

where

$h_b$  = height of continuous aqueous phase in the bottom of the vessel, in.

$$A_t = \left[ \frac{(0.748)\pi D^2 Lv_{hc}}{F_t - 1.2D} \right] \frac{\pi D}{38.4} \quad (6-32)$$

where

$A_t$  = cross-sectional area at the top of the vessel occupied by continuous hydrocarbon phase,  $\text{ft}^2$ .

$$A_b = \left[ \frac{(0.748)\pi D^2 Lv_{aq}}{F_t - 1.2D} \right] \frac{\pi D}{38.4} \quad (6-33)$$

where

$A_b$  = cross-sectional area at the bottom of the vessel occupied by continuous aqueous phase,  $\text{ft}^2$ .

## 6.10 DECANTER [25]

In most general applications, a decanter is a continuous gravity separation vessel that does not run full, as contrasted to a settler that usually runs full, with one stream exiting at or near the top of a horizontal vessel. For most decanters, one phase of a two-plane mixture overflows out of the vessel (Figure 6-12). The concept of the decanter involves the balancing of liquid heights due to differences in density of the two phases, as well as settling velocity of the heavier phase falling through the lighter, or the lighter rising through the heavier.

Terminal Settling Velocity [25]:

$$v_d = gd^2 \frac{(\rho_d - \rho_c)}{18\mu_c}, \text{ ft/s} \quad (6-34)$$

### EXAMPLE 6-3 Horizontal Gravity Settlers

Using the data from Sigales [24] and following the design of [19], data for propane/caustic wash is as follows. where

$$\begin{aligned} F_{hc} &= 95 \text{ gpm} \\ F_{aq} &= 39 \text{ gpm} \\ v_{aq} &= 5 \text{ in./ min} \\ v_{hc} &= 120 \text{ in./ min} \\ r &= 3.4. \end{aligned}$$

The terminal (highest calculated) settling velocity of the aqueous droplet in/through the hydrocarbon phase is

$$v_{hc} = (1.2)(5 \text{ in./ min})(95 \text{ gpm}/39 \text{ gpm}) = 14.6 \text{ in./ min}$$

Because this is more than the 10 in./min recommended earlier, use

$$v_{hc} = 10 \text{ in./ min}$$

Assume for design,  $f_{hc} = f_{aq} = 2$  (from earlier discussion).

Then, using Eq. (6-26),

$$a = (1.889) \frac{[(10)(2)(39) + (5)(2)(95)]}{[(3.4)(10)(5)]} = 19.22$$

From Eq. (6-27),

$$b = \frac{(3.505)(2)(95)(2)(39)}{[(3.4)^2(10)(5)]} = 89.87$$

From Eq. (6-25),

$$D = \pm \left[ \frac{19.22}{2} \pm \frac{\sqrt{(19.22^2 - 4(89.87))}}{2} \right]^{0.5}, \text{ ft}$$

$D = 3.34 \text{ ft}$  or  $-2.83 \text{ ft}$  (latter is an unreal negative number, so use  $3.34 \text{ ft}$ ) where

$A_t$  = Area of segment at the top of the vessel

Substituting these values in Eq. (6-22) gives

$$A_t = \frac{1.2 \times 3.34}{\left[ \frac{7.48 \times 3.4 \times 3.34 \times 10}{(2 \times 95)} - \frac{38.4}{(\pi \times 3.34)} \right]} = 4.945 \text{ ft}^2$$

where

$v_d$  = terminal settling velocity of a droplet, ft/s

$g$  = acceleration due to gravity,  $32.17 \text{ ft/s}^2$

$d$  = droplet diameter, ft ( $1 \text{ ft} = 304,800 \mu\text{m}$ , or  $1 \mu\text{m} = 0.001 \text{ mm}$ )

$\rho_d$  = density of fluid in the droplet, lb/ft<sup>3</sup>

$\rho_c$  = density of fluid continuous phase, lb/ft<sup>3</sup>

$\mu_c$  = viscosity of the continuous phase, lb/ft s.

Note:  $1 \text{ cP} = 6.72 \times 10^{-4} \text{ lb/ft s}$

For a decanter that operates under gravity flow with no instrumentation flow control, the height of the heavy phase liquid leg

Using  $L/D = 3.4$ , for the bottom segment of the vessel, aqueous layer

$$A_b = \frac{1.2 \times 3.34}{\left[ \frac{7.48 \times 3.4 \times 3.34 \times 5}{(2 \times 39)} - \frac{38.4}{(\pi \times 3.34)} \right]} = 2.2448 \text{ ft}^2$$

Then, using Eq. (6-21a)

$$h_t = \frac{7.48(4.945)(3.4 \times 3.34)(10)}{(2 \times 95)} = 22.1 \text{ in.}$$

$$h_b = \frac{(7.48 A_b L v_{aq})}{(f_{aq} F_{aq})}$$

$$= \frac{7.48(2.2448)(3.4 \times 3.34)(5)}{(2 \times 39)} = 12.2 \text{ in.}$$

Then,

$$h_t/D = \frac{(22.1)}{(12)(3.34)} \times 100 = 55\%$$

$$h_b/D = \frac{(12.2)}{(12)(3.34)} \times 100 = 30\%$$

Since  $h_t$  and  $h_b$  are between 30 and 70% of the diameter, the solution is acceptable.

In summary:

Design Calculation	Practical Design Use
Diameter $3.34 \text{ ft}$ (40.08 in.)	$3.5 \text{ ft}$ (42 in.) or $3.83 \text{ ft}$ (46 in.)
Length $H_C$ inlet/outlet: 11 ft	12 ft or 14 ft

Abernathy [19] has compared several design methods as follows:

	Sigales	This Method	Modified Happel	Happel	Rule-of Thumb
Diameter	2.67 ft	3.34 ft	3.36 ft	4.01 ft	4.1 ft
$h_t$	10 in.	22 in.	22.6 in.	24 in.	32.5 in.
$h_b$	8 in.	12 in.	11.3 in.	24 in.	16.7 in.
Interface	14 in.	6 in.	6.4 in.	0 in.	0 in.
$H_C$ residence time	1.1 min	4.4 min	4.6 min	6.8 min	10 min

above the interface is balanced against the height of one light phase above the interface [8]. Figures 6-12 and 6-13 illustrate the density relationships and the key mechanical details of one style of decanter.

The same results can be achieved with internal flat plate baffles and outlet nozzles.

$$(z_h - z_i)\rho_H = (z_l - z_i)\rho_L \quad (6-35)$$

where

$z_h$  = heavy phase outlet dimension from the bottom of the horizontal decanter

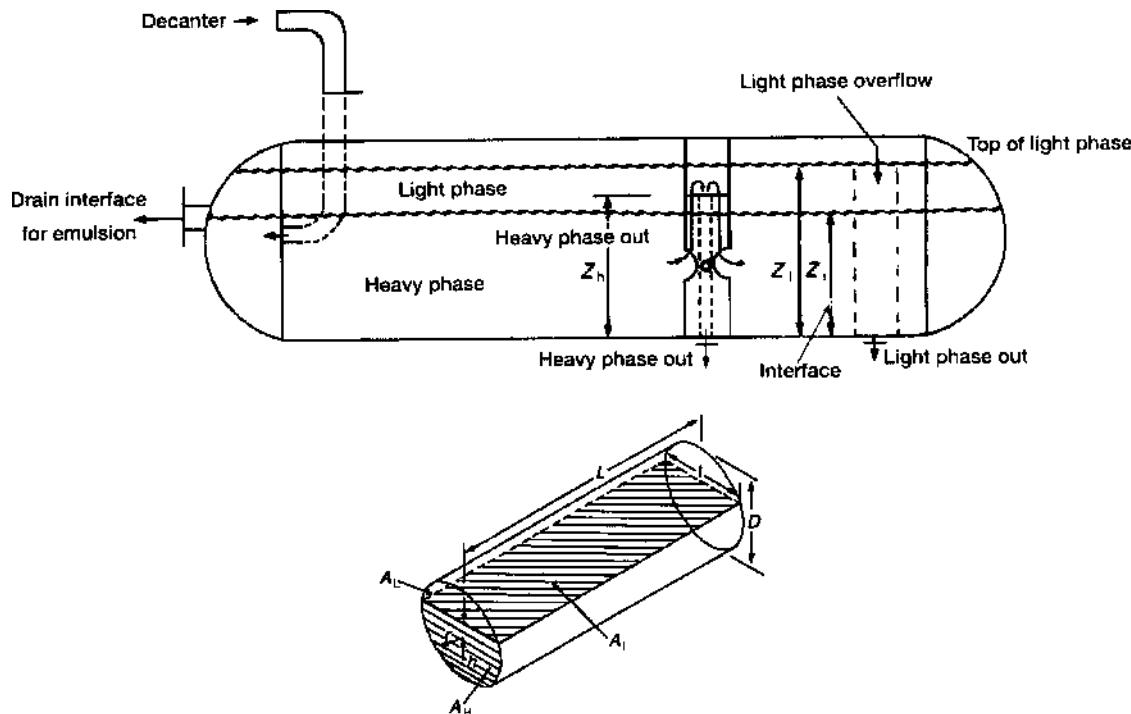


Figure 6-12 Gravity decanter basic dimensions. (Adapted by permission from Schweitzer, P.A., *Handbook of Separation Techniques for Chemical Engineers*, McGraw-Hill Book Co., 1979 [25].)

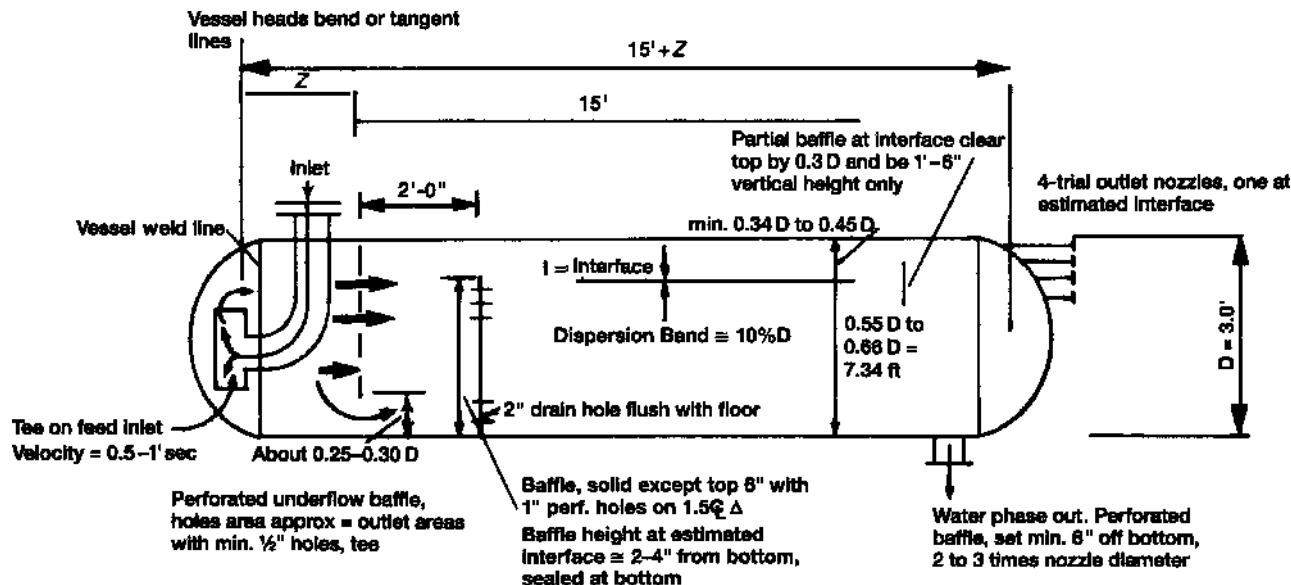


Figure 6-13 Decanter.

$z_i$  = interface measured from bottom

$z_l$  = light phase outlet measured from the bottom of the decanter.

Droplet diameter, when other data is not available:

$$d = 150 \mu\text{m} (0.0005 \text{ ft})$$

Schweitzer [25] recognizes that this is generally on the safe side, because droplets generated by agitation range from 500 to 5000  $\mu\text{m}$ , turbulent droplet range from 200 to 10,000  $\mu\text{m}$ . Due to limitations

of design methods, decanters sized for droplets larger than 300  $\mu\text{m}$  often result in being too small to work properly.

The continuous phase moves through the vessel on a uniform flow equal to the overflow rate. To identify which is the continuous phase (from Selker and Sleicher by Schweitzer),

$$\theta = \frac{Q_L}{Q_H} \left( \frac{\rho_L \mu_H}{\rho_H \mu_L} \right)^{0.3} \quad (6-36)$$

$\theta$	Result
< 0.3	light phase always dispersed
0.3–0.5	light phase probably dispersed
0.5–2.0	phase inversion probable, design for worst case
2.0–3.3	heavy phase probably dispersed
> 3.3	heavy phase always dispersed.

where

$$Q_L = \text{volumetric flow rate, ft}^3/\text{s, light phase}$$

$$Q_H = \text{volumetric flow rate, ft}^3/\text{s, heavy phase}$$

$$\rho_L = \text{density of light phase fluid, lb/ft}^3$$

$$\rho_H = \text{density of heavy phase fluid, lb/ft}^3$$

$$\mu_H = \text{viscosity of heavy phase, lb/ft s}$$

$$\mu_L = \text{viscosity of light phase, lb/ft s.}$$

There is a dispersion band through which the phases must separate. Good practice [25] normally keeps the vertical height of the dispersed phase  $H_D < 10\%$  of decanter height (normally a horizontal vessel), and

$$\frac{1}{2}H_D \frac{A_I}{Q_D} > 2-5 \text{ min}$$

where

$$A_I = \text{area of interface assuming flat interface, ft}^2$$

$$H_D = \text{height of the dispersion band, ft}$$

$$Q_D = \text{volumetric flow, dispersed phase, ft}^3/\text{s}$$

$$D = \text{decanter diameter, ft.}$$

Horizontal vessels as cylinders are generally more suitable for diameters up to about 8 ft than other shapes, or vertical, due in part to the increased interfacial area for interface formation. For a horizontal drum (see Figure 6-12),

$$I = 2(r^2 - h^2)^{1/2} \quad (6-37)$$

$$A_I = IL \quad (6-38)$$

where

$$h = \text{distance from center to given chord of a Vessel, ft}$$

$$I = \text{width of interface, ft}$$

$$L = \text{decanter length, ft.}$$

$$r = \text{vessel radius, ft.}$$

$$A_L = 1/2\pi r^2 - h(r^2 - h^2)^{1/2} - r^2 \arcsin(h/r) \quad (6-39)$$

or use the methods from the Appendix to calculate area of a sector of a circle. The arc is in radians.

$$\text{Radians} = (\text{degrees})(\pi/180)$$

$$A_H = \pi r^2 - A_L \quad (6-40)$$

$$D_L = \frac{4A_L}{(I+P)} \quad (6-41)$$

where  $D_L$  = diameter for light phase, ft.

$$D_H = \frac{4A_H}{(I+2\pi r - P)} \quad (6-42)$$

where

$$A_L = \text{cross-sectional area allotted to light phase, ft}^2$$

$$A_H = \text{cross-sectional area allotted to heavy phase, ft}^2$$

$$P = 2r \arccos(h/r)$$

Degree of turbulence [25]:

$$N_{Re} = \frac{v_c D_H \rho_c}{\mu_c} \quad (6-43)$$

$D_H$  = hydraulic diameter, ft = 4 (flow area for the phase in question/wetted perimeter of the flow channel)

$v_c$  = velocity down the flow channel for continuous phase.

## GUIDELINES FOR SUCCESSFUL DECANTERS [25]

$Re$	Results
< 5000	little problem
5000–20,000	some hindrance
20,000–50,000	major problem may exist
Above 50,000	poor separation

Velocities of both phases should be the same throughout the unit. By adjusting mechanical internals, a ratio of  $< 2:1$  is suggested (internals do not need to be equal) [25]. Velocities for entrance and exit at the vessel nozzle should be low, in the range of 0.5–1.5 ft/s. The feed must not “jet” into the vessel, and should be baffled to prevent impingement in the main liquid body, keeping turbulence to an absolute minimum to none. Baffles can/should be placed in the front half of the unit to provide slow flow of the fluids either across the unit or up/down paths followed by the larger stilling chamber, before fluid exits (see Figure 6-13).

## 6.11 IMPINGEMENT SEPARATORS

As the descriptive name suggests, the impingement separator allows the particles to be removed to strike some type of surface. This action is better accomplished in pressure systems where pressure drop can be taken as a result of the turbulence which necessarily accompanies the removal action.

Particle removal in streamline flow is less efficient than for turbulent flow, and may not be effective if the path of travel is not well baffled.

The “target” efficiency for impingement units expresses the fraction of the particles in the entraining fluid, moving past an object in the fluid, which impinge on the object. The target efficiencies for cylinders, spheres, and ribbon-like particles are given for conditions of Stokes’ Law in an infinite fluid by Figure 6-14.

If the particles are close enough together in the fluid to affect the path of each other, then Figure 6-14 gives conservative efficiencies. For particles differing considerably from those given in the curves, actual test data should be obtained.

There are basically three construction types for impingement separators:

1. Wire mesh
2. Plates (curved, flat, or special shaped)
3. Packed Impingement Beds.

## KNITTED WIRE MESH

A stationary separator element of knitted small diameter wire or plastic material is formed of wire 0.003–0.016 in. (or larger) in diameter into a pad of 4, 6, or 12 in. thickness and serves as the impingement surface for liquid particle separation. Solid particles can be separated, but they must be flushed from the mesh to prevent

plugging. Although several trade name units are available they basically perform on the same principle, and have very close physical characteristics. Carpenter [26] presented basic performance data for mesh units. Figure 6-15 shows a typical eliminator pad.

Figure 6-16 pictorially depicts the action of the wire mesh when placed in a vertical vessel.

**EXAMPLE 6-4****Decanter, using the method of Schweitzer [25]**

A plant process needs a decanter to separate oil from water. The conditions are as follows.

$$\text{Oil flow} = 8500 \text{ lb/h}$$

$$\rho = 56 \text{ lb/ft}^3$$

$$\mu = 9.5 \text{ cP}$$

$$\text{Water flow} = 42,000 \text{ lb/h}$$

$$\rho = 62.3 \text{ lb/ft}^3$$

$$\mu = 0.71 \text{ cP.}$$

*Units of conversion:*

$$Q_{\text{oil}} = (8500)/(56)(3600) = 0.0421 \text{ ft}^3/\text{s}$$

$$\mu_{\text{oil}} = (9.5)(6.72 \times 10^{-4}) = 63.84 \times 10^{-4} \text{ lb/ft-s}$$

$$Q_{\text{water}} = 42,000/(62.3)(3600) = 0.1873 \text{ ft}^3/\text{s}$$

$$\mu_{\text{w}} = (0.71)(6.72 \times 10^{-4}) = 4.77 \times 10^{-4} \text{ lb/ft-s.}$$

Checking dispersed phase, and using Eq. (6-36),

$$\theta = \frac{0.0421}{0.1873} \left[ \frac{(56)(4.77 \times 10^{-4})}{(62.3)(63.84 \times 10^{-4})} \right]^{0.3} \\ = 0.09997$$

Therefore, light phase is always dispersed since  $\theta$  is less than 0.3.

Settling rate for droplets of oil through water from Eq. (6-5):

(Assume droplet size is  $d = 0.0005 \text{ ft}$  (150  $\mu\text{m}$ ), as discussed earlier), and using Eq. (6-5),

$$V_{\text{oil}} = \frac{32.2 \times 0.0005^2 \times (56 - 62.3)}{(18 \times 4.77 \times 10^{-4})} \\ = -0.006 \text{ ft/s}$$

The (−) sign means the oil rises instead of settling.

*Overflow rate:*

Assume  $I$  (Figure 6-12) is 80% of diameter  $D$  of the vessel and that  $L/D = 5$ .

where

$L$  = length of the vessel, ft.

Then,  $Q_c/A_I < v_d$

$$A_I = IL = (0.8D)(5D) = 4D^2, \text{ then,}$$

$$Q_c/4D^2 < v_d$$

and

$$D \geq 1/2(Q_c/v_d)^{1/2} \geq 1/2(0.187/0.006)^{1/2}$$

$$D = 2.79 \text{ ft}$$

where  $v_d$  = terminal setting velocity of a droplet, ft/s

$$\text{Length, } L = 5D = 5(2.79) = 14 \text{ ft}$$

*Interface Level:*

Assume that interface is being held one foot below the top of the vessel to prevent interface from reaching the top oil outlet.

Then,

$$h = 0.5 \text{ ft}$$

$$r = 3.0/2 = 1.5 \text{ ft.}$$

From the Eq. (6-37)

$$I = 2(r^2 - h^2)^{1/2} = 2[(1.5)^2 - (0.5)^2]^{1/2} = 2.828 \text{ ft.}$$

And from Eq. (6-39)

$$A_{\text{oil}} = (1/2)(\pi)(1.5)^2 \\ - 0.5[(1.5)^2 - (0.5)^2]^{1/2} - (1.5)^2 \arcsin(0.5/1.5) \\ = 3.534 - 0.707 - 0.765 \\ = 2.062 \text{ ft}^2$$

*Note:* In radians,  $\arcsin(0.5/1.5) = (19.47/180)\pi = 0.3398$

$$A_{\text{water}} = \pi(1.5)^2 - A_{\text{oil}} = \pi(2.25) - 2.062 = 5.01 \text{ ft}^2$$

$$P = 2(1.5)[\arccos(0.5/1.5)] = 3.69 \text{ ft}^2$$

$$\text{Area interface, } A_I = (I)(L) = (2.828)(14) = 39.59 \text{ ft}^2$$

*Secondary settling:*

Continuous phase water droplets to resist the oil overflow rate if it gets on the wrong side of interface.

$$v_{\text{water}} \leq Q_{\text{oil}}/A_I = 0.0421/39.59 = 0.00106 \text{ ft/s.}$$

Then, from settling-velocity equation, the droplet diameter  $d$  is

$$d = \left[ \frac{18v_t \mu}{g(\rho - \rho_s)} \right]^{0.5}$$

$$d = \left[ \frac{18 \times 6.384 \times 10^{-3} \times 0.00106}{32.17 \times (62.3 - 56)} \right]^{0.5}$$

$$d = 0.0007752 \text{ ft} (236 \mu\text{m})$$

*Note:*  $1 \mu\text{m} = 39.37 \times 10^{-6} \text{ in} = 3.2808 \times 10^{-6} \text{ ft}$

(continued)

**EXAMPLE 6-4—(continued)**

Checking coalescence time, assume  $H_D$  = height of dispersion band = 10% of  $D = 0.3$  ft

Time available to cross the dispersed band =  $1/2(H_D A_l / Q_D)$  should be > 2–5 min

$$= 1/2[(0.279)(39.59)/(0.0421)] \\ = 131 \text{ s, which is } 2.2 \text{ min}$$

should be acceptable, but is somewhat low.

Then

$$D_{\text{oil}} = 4(2.062)/(2.828 + 3.69) = 1.265 \text{ ft}$$

$$V_{\text{oil}} = Q_{\text{oil}} / A_{\text{oil}}$$

$$V_{\text{oil}} = 0.0421 / 2.062 = 0.0204 \text{ ft/s}$$

The Reynolds number is

$$N_{Re} = \frac{D_{\text{oil}} V_{\text{oil}} \rho_{\text{oil}}}{\mu_{\text{oil}}}$$

$$N_{Re} = \frac{1.2654 \times 0.0204 \times 56}{63.84 \times 10^{-4}} = 226.4$$

$$D_{\text{water}} = 4(5.01) / [2.828 + 2\pi(1.5) - 3.69] = 2.34 \text{ ft}$$

$$v_{\text{water}} = \frac{Q_{\text{water}}}{A_{\text{water}}} \text{ ft/s}$$

$$= 0.187 / 5.01 = 0.0373 \text{ ft/s}$$

The Reynolds number is

$$N_{Re(\text{water})} = \frac{(0.0373)(2.34)(62.3)}{4.77 \times 10^{-4}} = 11,399$$

The degree of turbulence would be classified as acceptable, but the unit must not be increased in capacity for fear of creating more water phase turbulence.

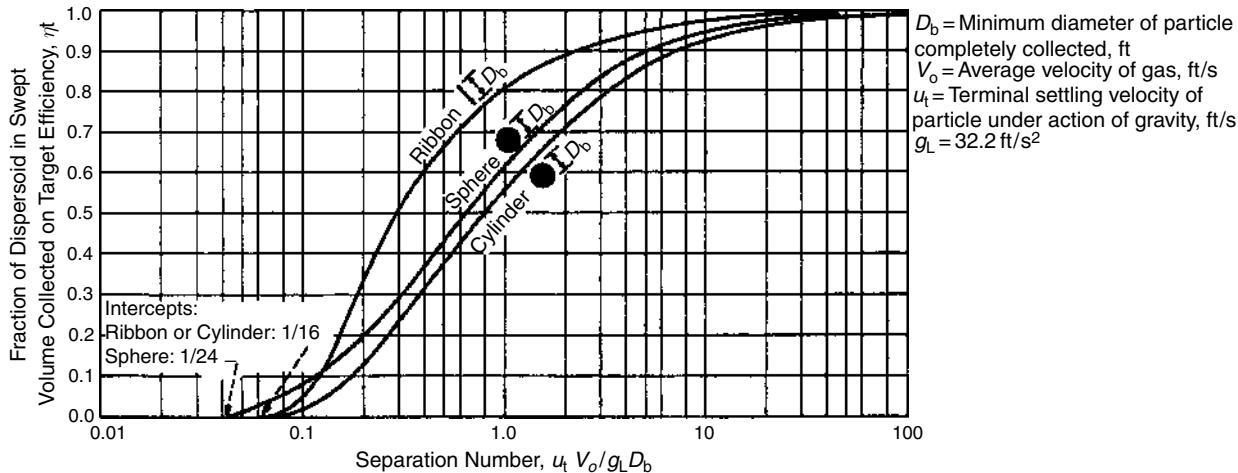
Referring to Figure 6-16, the typical situation represents a vapor disengaging from a liquid by bursting bubbles and creating a spray of liquid particles of various sizes. Many of these particles are entrained in the moving vapor stream. The largest and heaviest particles will settle by gravity downward through the stream and back to the bottom of the vessel or to the liquid surface. The smaller particles move upward, and if not removed will carry along in the process stream. With wire mesh in the moving stream, the small particles will impinge on the wire surfaces, coalesce into fluid films and then droplets, run to a low point in their local system, and fall downward through the up-flowing gas stream when sufficiently large. The gas leaving is essentially free from entrained liquid unless the unit reaches a flooding condition.

For special applications the design of a mist eliminator unit may actually be an assembly in one casing of wire mesh and fiber packs/pads or in combination with Chevron-style mist elements (Figures 6-17a–c.) This can result in greater recovery efficiencies

for small particles and for higher flow rates through the combined unit. Refer to the manufacturers for application of these designs.

**Mesh Patterns.** There are several types of mesh available, and these are identified by mesh thickness, density, wire diameter, and weave pattern. Table 6-9 identifies most of the commercial material now available. The knitted pads are available in any material that can be formed into the necessary weaves; this includes stainless steels, monel, nickel, copper, aluminum, carbon steel, tantalum, Hastelloy, Saran, polyethylene, fluoropolymer, and glass multifilament.

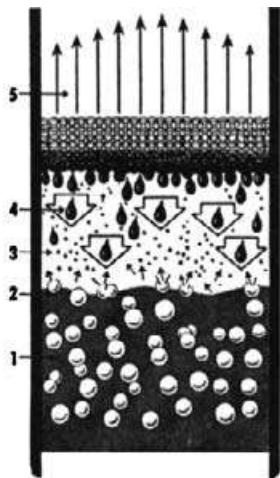
**Capacity Determination.** The usual practice in selecting a particular mesh for a given service is to determine the maximum allowable velocity and from this select a vessel diameter. In the case of existing vessels where mesh is to be installed, the reverse procedure is used, that is, determine the velocity conditions which will prevail and select a mesh to fit as close to the conditions as possible. The procedure is outlined below.



**Figure 6-14** Target efficiencies for spheres, cylinders, and ribbon-like particles. (By permission from Perry, J.H., *Chemical Engineers' Handbook*, 3rd ed., McGraw-Hill Company, 1950 [5].)



Figure 6-15 Eliminator Pad. (Courtesy of Otto H. York Co.)



When a gas is generated in, or passes through, a liquid (1), the gas on bursting from the liquid surface (2) carries with it a fine spray of droplets—liquid entrainment—which are carried upward in the rising gas stream (3). As the gas passes through the mist eliminator, these droplets impinge on the extensive surface of the wire, where they are retained until they coalesce into large drops. When these liquid drops reach sufficient size, they break away from the wire mesh (4) and fall back against the rising gas stream. In this way, the entrained droplets are literally “wiped out” of the gas which, freed from liquid entrainment, (5) passes on unhindered through the mesh.

Figure 6-16 Diagram of action of wire mesh in liquid-vapor separation. (Courtesy of Metal Textile Corp., Bulletin ME 9-58.)

Allowable vapor velocity (mesh in horizontal position):

$$V_a = k \sqrt{\frac{\rho_L - \rho_v}{\rho_v}}$$

where

$V_a$  = maximum allowable superficial vapor velocity across inlet face of mesh, ft/s

$k$  = constant based on application (Table 6-10); average for free flowing system = 0.35 for 9–12 lb/ft<sup>3</sup> mesh

$\rho_L$  = liquid density, lb/ft<sup>3</sup>

$\rho_v$  = vapor density, lb/ft<sup>3</sup>.

For other mesh densities, use  $k$  [29] of 0.4 for 5 lb/ft<sup>3</sup> mesh (high capacity), and 0.3 for plastic mesh such as Teflon® and polypropylene.

*Fleximesh Design Manual* [29] suggests “dry” mesh pressure drop of

$$\Delta p_D = \left( \frac{f_c l a \rho_v V_s}{g_c \varepsilon^3} \right) \left( \frac{27.7}{144} \right) \quad (6-44)$$

$$\Delta p_T = \Delta p_D + \Delta p_L \quad (6-45)$$

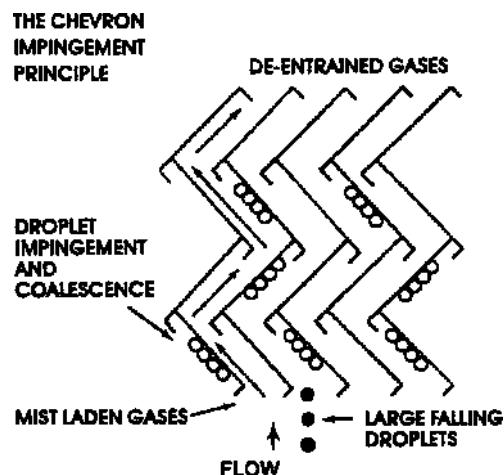


Figure 6-17a Separation/Impingement action of Chevron-style mist eliminators. Flow is up the V-shaped plates assembly. (Courtesy of Bulletin KME-12, Koch Engineering Co.)

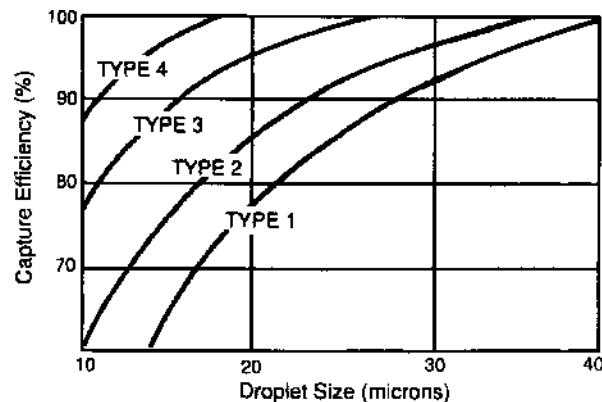
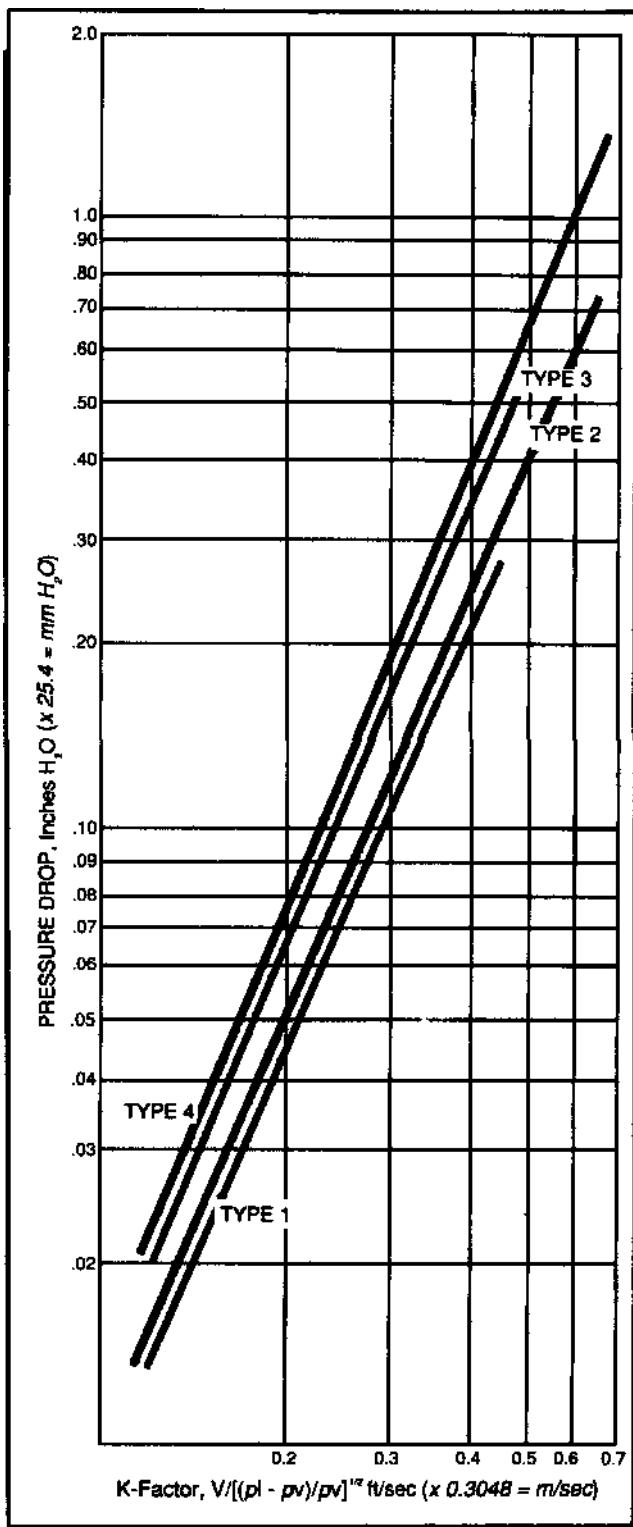


Figure 6-17b Capture efficiency vs particle size for four standard York-Vane mist eliminators. (By permission from Otto H. York Co Inc.)



**Figure 6-17c** Pressure drop vs K-factor for standard York-Vane mist eliminators, air-water system. (By permission from Otto H. York Co., Inc.)

For  $\Delta p_L$  see manufacturer's curves.

A rough approximation of operating mesh pressure drop is 1 in. water or less. The calculated pressure drop at the maximum allowable velocity is close to 1.5 in. of water. Therefore,

$$\Delta p_T = 1.5 \left( \frac{V_{act}}{V_{max}} \right)^2 \quad (6-46)$$

where

- $a$  = specific surface area,  $\text{ft}^2/\text{ft}^3$
- $f_c$  = friction factor, dimensionless
- $g_c$  = gravitational constant,  $32.17(1\text{bm}/1\text{bf})(\text{ft}/\text{s}^2)$
- $l$  = wire mesh thickness, ft
- $\Delta p_D$  = pressure drop, no entrainment, in. of water
- $\Delta p_L$  = pressure drop, due to liquid load, in. of water
- $\Delta p_T$  = pressure drop, total across wet pad, in. of water
- $V_s$  = superficial gas velocity, ft/s
- $\varepsilon$  = void fraction of wire mesh, dimensionless
- $\rho_L$  = liquid density,  $\text{lb}/\text{ft}^3$
- $\rho_v$  = vapor density,  $\text{lb}/\text{ft}^3$
- $f$  = generally ranges from 0.2 to 2 for dry mesh.

Subscript:

act = actual

max = maximum.

The correlation factor  $k$  is a function of the liquid drop size, liquid viscosity, liquid load, disengaging space, type of mesh weave, and so on. And  $k$  varies somewhat with system pressure; as pressure increases the  $k$  value decreases. The manufacturers should be consulted for final design  $k$  values for a system, because the wire style, size, and material also affect the value. For pressures below 30 psig,  $k = 0.35$  avg; above 30 psig,  $k$  value decreases with pressure with an approximate value of 0.30 at 250 psig and 0.275 at 800 psig. Certain values have been found satisfactory for estimating systems described in Tables 6-10 and 6-11.

For conditions of high liquid loading, use caution in design. Use the high velocities for very fine mist to remove the small particles, and use two mesh pads in series with the second mesh operating at a lower velocity to remove the larger drops re-entrained from the first mesh. Systems involving high viscosity fluids should be checked with the various manufacturers for their case history experience. Lower  $k$  values are used for systems with high vacuum, high viscosity liquids, low surface tension liquids, and systems with very bad fouling conditions. Table 6-11 indicates the effect of disengaging height on the allowable  $k$  value. Similar relations should hold for other mesh densities.

**Velocity Limitations.** Very low velocities will allow particles to drift through the mesh and be carried out with the leaving vapor. Also, very high velocities will carry liquid to the top of the mesh, establish a "flooding" condition, and then re-entrain the liquid from the surface of the mesh. For most situations very good performance can be expected for all velocities from 30 to 100% of the optimum allowable design velocity. The minimum allowable safe design velocity is 10% of the value calculated by the equation. The flooding velocity of the mesh is usually about 120–140% of the maximum allowable velocity.

Generally the maximum allowable velocities are lower under conditions of pressure, and higher under conditions of vacuum. The limits and ranges of each area are determined by the relative operating densities of the vapor and liquid, the nature of the entrainment, and the degree of separation required.

When the mesh is installed with the pad vertical or inclined, the maximum allowable velocity is generally used at 0.67 times the allowable value for the horizontal position.

**TABLE 6-9 Identification of Wire Mesh Types**

General Type	Density, (lb/ft <sup>3</sup> )*	Surface Area (ft <sup>2</sup> /ft <sup>3</sup> )	Thickness (in.)**	Min Efficiency (wt. %)	Application
High Efficiency	12	115	4+	99.9+	Relatively clean, moderate velocity
Standard Efficiency	9	85	4+	99.5+	General purpose
Optimum Efficiency or VH Efficiency, and Wound type	13–14	120	4+	99.9+	For very high efficiency
Herringbone, High Through-put or Low Density	5–7	65±	4–6+	99.0+	For services containing solids, or "dirty" materials

\*If the mesh is made of nickel, monel, or copper, multiply the density values by 1.13, referenced to stainless steel.

\*\* 4" is minimum recommended thickness; 6" is very popular thickness; 10" and 12" recommended for special applications such as fine mists, oil vapor mist.

(Source: Compiled from [27, 28].)

**TABLE 6-10 "k" Values of Knitted Mesh**

Bottom of mesh at least 12 in. above liquid surface		
Service Conditions	"k"	General Type Mesh
Clean fluids, moderate liquid load, fits 90% of process situations	0.35–0.36	Standard
	0.35	High efficiency
	0.25	Very high efficiency
High viscosity, dirty suspended solids	0.40	Low density or Herringbone, high Throughput
<i>Vacuum operations</i>		
2 in. Hg abs	0.20	Standard or
16 in. Hg abs	0.27	High efficiency
Corrosive Chemical	0.21	Plastic-coated wire, or plastic strand

(Source: Compiled from various manufacturer's published data.)

Note: k values for estimating purposes, not final design unless verified by manufacturer. Unless stated, all values based on stainless steel wire.

**Design Velocity.** To allow for surges, variations in liquid load and peculiarities in liquid particle size and physical properties use

$$V_D = 0.75 V_a \quad (6-47)$$

for the design of new separators.  
where

$V_D$  = design vapor velocity, ft/s

$V_a$  = maximum allowable superficial vapor velocity across inlet face of mesh, ft/s

When checking existing vessels to accept wire mesh, some variation may have to be accepted to accommodate the fixed diameter

**TABLE 6-11 Variation of k with Disengaging Height**

Disengaging Height Above Mesh (in.)	Allowable k Value
3	0.12
4	0.15
5	0.19
6	0.22
7	0.25
8	0.29
9	0.32
10	0.35
11	0.38
12	0.40
13	0.42
14	0.43

(Source: By permission from York [28].)

Note: Values based on 12 lb/ft<sup>3</sup> wire mesh. Design practice normally does not exceed k of 0.4 even for higher disengaging height.

condition, but this is no great problem since the range of good operation is so broad.

**Efficiency.** For most applications the efficiency will be 98–99%+ as long as the range of operating velocity is observed. The typical performance curves for this type of material are given in Figures 6-17b, 6-18, and 6-19. For hydrocarbon liquid–natural gas system, guarantees are made that not more than 0.1 gal. of liquid will remain in the gas stream per million cubic feet of gas. Special designs using a 3-ft thick pad reduce radioactive entrainment to one part per billion [28].

For the average liquid process entrainment the mesh will remove particles down to 4–6 μ at 95%+ recovery efficiencies (see Figure 6-19). Particles smaller than this usually require two mesh pads or the fiber pack style discussed later. Carpenter [26, 30] shows the calculated effect of decreasing particle size on percent

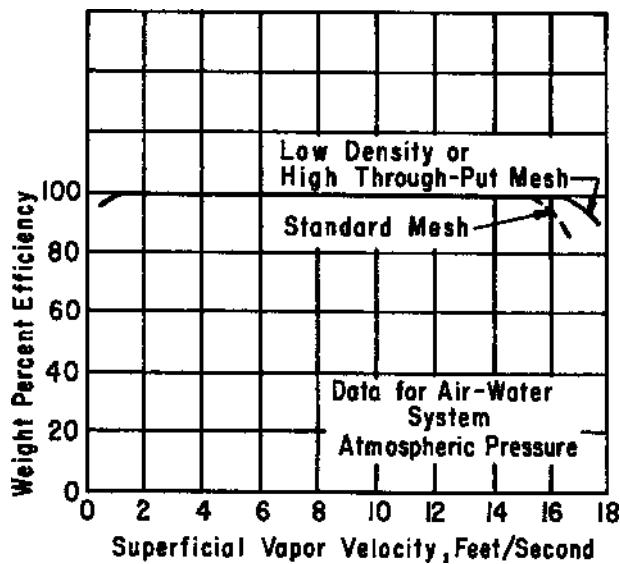


Figure 6-18 Typical wire mesh efficiency.

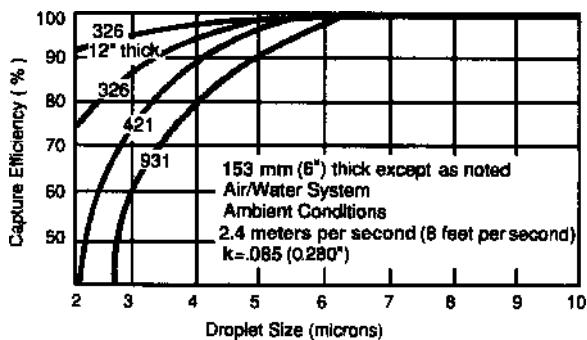


Figure 6-19 Capture efficiency vs particle size for four types of DEMISTER® knitted mesh mist eliminators. (By permission from Otto H. York Co. Inc.)

entrainment removed at various linear velocities. For water particles in air at atmospheric pressure, the  $8\text{ }\mu$  particles are 99% removed at 3.5 ft/s, the  $7\text{ }\mu$  at 5 ft/s, and the  $6\text{ }\mu$  at 6.8 ft/s. An excellent performance may be obtained in most systems for velocities of 30–110% of calculated values [31].

**Pressure Drop.** Pressure drop through wire mesh units is usually very low, in the order of 1-in. water gauge for a 4-in. or 6-in. thick pad. For most pressure applications this is negligible. If solids are present in the particle stream, then solids build-up can become appreciable, and is usually the guide or indicator for cleaning of the mesh. A 12-in. pad may require a 3-in. water drop. Figures 6-20 and 6-21 present the range of expected pressure drops for a spread of 3–1600 lb/h-ft<sup>2</sup> for liquid rates. Although this is for air-water system at atmospheric pressure it will not vary much unless the physical properties of the vapor and liquid deviate appreciably from this system, in which case the general Fanning equation can be used to approximate the pressure drop under the new conditions. Approximate values based upon air-water tests suggest these relations [27].

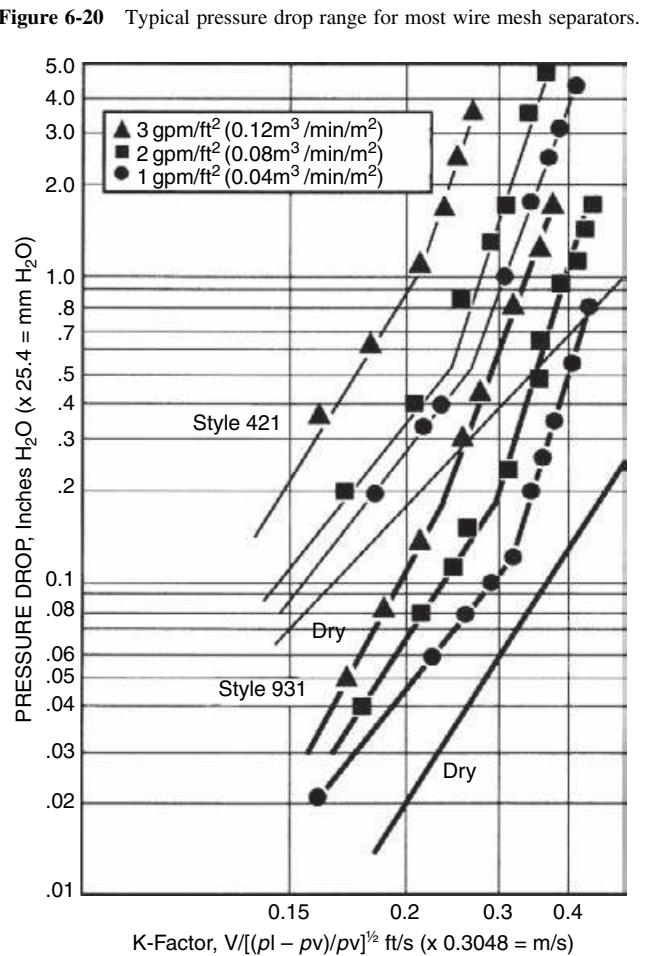
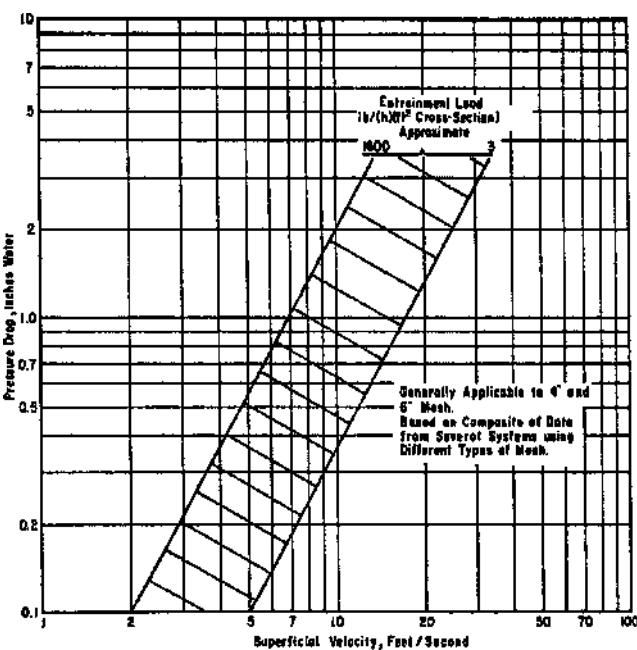


Figure 6-21 Typical wire mesh mist eliminator pressure drop curves for one style of mesh at three different liquid loadings. Others follow similar pressure drop patterns. (By permission from Otto H. York Co., Inc.)

For the standard weave, 4 in. thick,

$$\Delta p = 0.2V_D^2\rho_v \text{, in. water} \quad (6-48)$$

For the low density weave (high through-put), 6 in. thick,

$$\Delta p = 0.12V_D^2\rho_v \quad (6-49)$$

**Installation.** The knitted mesh separator unit may be placed in a pipe in which case a round flat rolled unit is usually used, or it may be placed in a conventional vessel. Although the vessel may be horizontal or vertical, the mesh must always be in a horizontal plane for best drainage. Some units in special situations have been placed at an angle to the horizontal, but these usually accumulate liquid in the lower portion of the mesh. Since the material is not self-supporting in sizes much over 12 in. diameter, it requires support bars at the point of location in the vessel. In most instances it is wise to also install hold-down bars across the top of the mesh in accordance with manufacturers' instructions as the material will tend to blow upward with a sudden surge or pulsation of vapor in the system. Many early installations made without the bars on top were soon found ineffective due to blowout holes, and wire particles were found in pipe and equipment downstream of the installation.

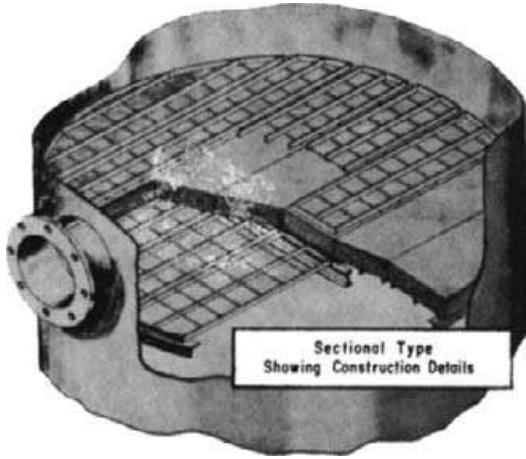


Figure 6-22 Typical installation of mesh strips in vertical vessel.  
(Courtesy of Otto H. York Co., Inc.)

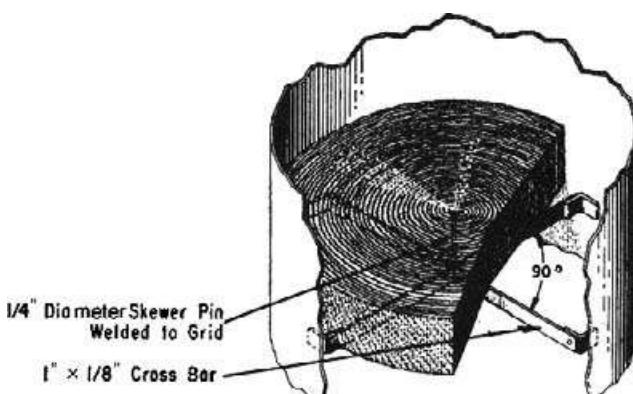


Figure 6-23 Typical installation of wound mesh pads in vertical vessel.  
(Courtesy of Metal Textile Corp., Bulletin ME 9-58.)

Figures 6-22 and 6-23 show a typical installation arrangement in a vertical vessel. The mesh is wired to the bottom support bars and the hold-down on top.

A few typical arrangements of mesh in vessels of various configurations are shown in Figure 6-24. Note that in some units of Figure 6-24 the mesh diameter is smaller than the vessel. This is necessary for best operating efficiency under the system conditions, and applies particularly when using an existing vessel.

When placing mesh in small diameter vessels it is important to discount the area taken up by the support ring before determining the operating velocity of the unit. For small 6-, 8-, and 12-in. vessels (such as in-line, pipe-with-mesh units) it is usual practice to use meshes of 6- or 8-in. thickness for peak performance.

Provide at least 6-12 in. minimum (preferably 18 in. minimum) of disengaging space ahead of the inlet face of the mesh, that is, above any inlet nozzle bringing the liquid-carrying vapors to the vessel, or above any liquid surface held in the vessel. Leave 12 in. minimum of disengaging space above the mesh before the vapors enter the vessel vapor exit connection. The mesh may be installed in horizontal, vertical, or slanting positions in circular, rectangular, or spherical vessels. For locations where the liquid drains vertically through the mesh pad perpendicular or angular to its thickness dimension, care must be taken to keep velocities low and not to force or carry the liquid through to the downstream side of the mesh.

The check or specification form of Figure 6-26 is necessary and helpful when inquiring wire mesh entrainment units, either as the mesh alone, or as a complete turnkey unit including vessel. For services where solids are present or evaporation of droplets on the mesh might leave a solid crust, it is usual practice to install sprays above or below the mesh to cover the unit with water (or suitable solvent) on scheduled (or necessary) operating times, as the plugging builds up. This is checked by a manometer or other differential pressure meter placed with taps on the topside and the bottom side of the mesh installation.

A few case examples for guidance include the following:

- 2-3% caustic solution with 10% sodium carbonate. This condition might plug the wire mesh. Sprays would be recommended.
- Raw river water. This presents no plugging problem.
- Light hydrocarbon mist. This presents no plugging problem.
- Heavy oil with suspended matter. This might plug. A light oil or solvent spray would be recommended for flushing the mesh.

#### FIBER BEDS/PADS IMPINGEMENT ELIMINATORS

The use of fiber packing held between wire mesh containing screens is best applied in the very low micron range, generally 0.1 to  $> 3\mu$  with recoveries of entrained liquid of up to 99.97% (by weight). Figures 6-27a-c illustrate the design concept, and its corresponding data table indicates expected performance. The fibers used mean the bed packing can be fabricated from fine glass, polypropylene fibers, or can be selected to be the most resistant to the liquid mist entering the unit from corrosive plant operations such as sulfuric acid, chlorine, nitric acid, ammonia scrubbing for sulfur oxides control, and many others. The entrained particles are removed by direct interception, inertial impaction, and Brownian capture.

The design rating for this equipment is best selected by the manufacturers for each application.

The concept of removal of entrained liquid particle is essentially the same as for wire mesh designs, except the particle size removed may be much smaller. Just as for other types of mist eliminators, the performance is affected by the properties of the liquid particles, entraining gas, temperature, pressure, liquid viscosity, particle size distribution of entrained material and the quantity of

**EXAMPLE 6-5****Wire Mesh Entrainment Separator**

Design a flash drum to separate liquid ethylene entrainment for the following conditions:

Volume of vapor = 465 cfm at  $-110^{\circ}\text{F}$  and 35 psig

Density of vapor,  $\rho_v = 0.30 \text{ lb/ft}^3$

Density of liquid ethylene,  $\rho_L = 33 \text{ lb/ft}^3$ .

Allowable velocity for wire mesh:

$$V_a = k \sqrt{\frac{\rho_L - \rho_v}{\rho_v}}$$

Use  $k = 0.35$  for clean service, moderate liquid loading and

$$V_a = 0.35 \sqrt{\left(\frac{33 - 0.3}{0.3}\right)}$$

= 3.66 ft/s, allowable loading velocity

Use  $V_D = 0.75 V_a$

Design velocity:

$$V_D = 0.75(3.66) = 2.74 \text{ ft/s}$$

Required vessel cross-sectional area:

$$A = 465/(60)(2.74) = 2.83 \text{ ft}^2$$

Vessel diameter:

$$D = \sqrt{\frac{2.83(4)}{\pi}} = 1.898 = 1' - 11"$$

First, try 2'-0" ID vessel

Deduct 4 in. from effective diameter for 2-in. support ring inside.

$$24'' - 4'' = 20''$$

total entrainment, and the desired process removal requirement. Some designs of these units provide excellent performance removal efficiencies at a wide range of rates (turndown), even at low gas rates.

Pressure drop is usually low depending on many factors, but can be expected in the range of 2–20 in. of water [32].

**Baffle Type Impingement.** There are many baffle type impingement separators. The efficiency of operation for entrainment is entirely a function of the contacting action inside the particular unit. There are no general performance equations which will predict performance for this type of unit; therefore manufacturers' performance data and recommendations should be used. A few of the many available units are shown in Figures 6-28–6-31. Many use the Chevron-style vertical plates as shown in Figures 6-17a and 6-30.

**Baffles (Chevron/Vanes).** One of the common impingement plate assemblies is of the Chevron "zig-zag" style of Figures 6-17a and 6-30. This style of impact separation device will tolerate higher gas velocities, high liquid loading, viscous liquids, reasonable solids, and relatively low-pressure drops. The collected liquid

Net area:

$$A = \frac{\pi (20)^2}{4(144)} = 2.18 \text{ ft}^2$$

$$\text{Actual velocity at ring} = 2.74 \left(\frac{2.83}{2.18}\right) = 3.56 \text{ ft/s.}$$

This is 97% of maximum allowable design, too high.

Second, try to increase diameter to next standard dimension, 2 ft, 6 in. Although intermediate diameters could have been selected, the heads normally available for such vessels run in 6-in. increments (either OD or ID).

Net inside diameter at support ring:

$$30''\text{OD} - 4'' - 3/4'' = 25 \frac{1}{4}''$$

Note that vessel wall was assumed as 3/8 in. thick. Net area =  $3.46 \text{ ft}^2$ .

$$\text{Actual velocity at ring} = \left(\frac{2.83}{3.46}\right)(2.74) = 2.24 \text{ ft/s}$$

Percent design velocity:  $2.24(100)/3.66 = 61.3\%$ . This is acceptable operating point. Note that if 28-in. OD  $\times$  3/8-in. wall pipe is available this could be used with weld cap ends, or dished heads. The percent design velocity would be 71.8%. This is also an acceptable design.

Pressure drop is in the order of 0.1–0.5 in. water.

*Note:* Since this vessel will operate as a flash drum with a liquid level at approximately 1/4 of its height up from bottom, place the inlet at about the center of the vessel (Figure 6-25). Use stainless 304 mesh for low temperature operation. Carbon steel is too brittle in wire form at this temperature.

droplets run down on the plate surfaces countercurrently to the up-flowing gas stream (see McNulty [33] for performance study).

Spacing of the plates and their angles is a part of the design using the manufacturers' data. Multiple pass designs can result in higher recovery efficiencies. The units can be designed/installed for vertical or horizontal flow.

Some of the same physical properties of the liquid and gas phases as well as temperature, pressure, the amount of entrained liquids (or solids if present), and the expected particle size and its distribution control the design and performance of these units also.

For preliminary selection:

$$V_D = k [(\rho_1 - \rho_v) / \rho_v]^{1/2} \quad (6-50)$$

where

$\rho_v$  = vapor density,  $\text{lb/ft}^3$  at actual conditions

$\rho_L$  = liquid density,  $\text{lb/ft}^3$  at actual conditions

$k$  = 0.40 for up-flow and 0.65 for horizontal flow.

For required flow area estimate only,

$$A = (\text{ACFS}) / V_D, \text{ ft}^2.$$

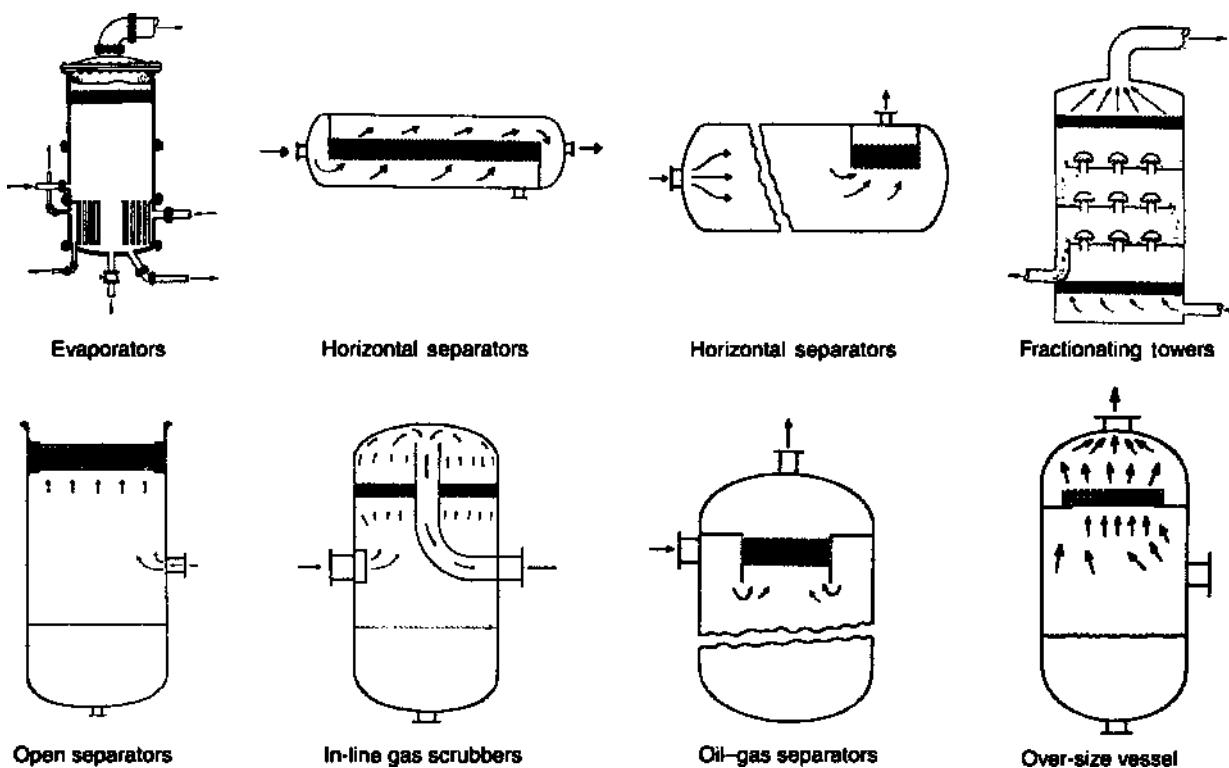


Figure 6-24 Typical mesh installations in process equipment. (Courtesy of Metal Textile Corp, Bulletin ME-7.)

where

$$A = \text{area, ft}^2$$

$$\text{ACFS} = \text{actual flow at conditions, ft}^3/\text{s}$$

$$V_D = \text{design vapor velocity, ft/s.}$$

Generally, this style of unit will remove particles of 12–15  $\mu$  efficiently. The typical droplet separator is shown for an air–water system in Figure 6-17a. This will vary for other systems with other physical properties. The variations in capacity (turndown) handled by these units is in the range of 3–6 times the low to maximum flow, based on  $k$  values [33].

A liquid–liquid separator used for removing small, usually 2% or less, quantities of one immiscible liquid from another is termed “coalescer”. This unit is not a gravity settler, but agglomerates the smaller liquid by passing through a surface contact medium such as excelsior, hay, cotton or wool bats, or cartridges of fibers similar in nature and weave to those of Tables 6-15a and b. Figure 6-32 illustrates a typical coalescer unit.

**Efficiency.** The efficiency of this type of unit varies, and is a function of the effectiveness of the impingement baffling arrangement. About 70% of separator applications can use the line-type unit; the other 30% require the vessel construction. The preference of the designer and problems of the plant operator are important in the final selection of a unit to fit a separation application. The efficiency for removal of liquid and solid suspended particles is 97–99%+ when handling 15  $\mu$  particles and larger. For steam service, a typical case would be 90% quality entering steam with 99.9% quality leaving.

Some units will maintain a reasonable efficiency of separation over a range of 60–120% of normal performance rating while

other types will not. This flexibility is very peculiar to the internal design of the unit. Some units are guaranteed to reduce mechanical entrainment loss to less than 0.1 gal. per million standard cubic feet of entraining gas.

**Pressure Drop.** Pressure drop in most units of this general design is very low, being in the order of 0.1–3 psi.

**How to Specify.** Manufacturers' catalogs are usually available and complete with capacity tables for the selection of a unit size. However, it is good practice to have this selection checked by the manufacturer whenever conditions will allow. This avoids misunderstandings and misinterpretations of the catalog, thereby assuring a better selection for the separation operation, and at the same time the experience of the manufacturer can be used to advantage.

With a manufacturer's catalog available, do the following:

1. Establish the minimum, normal, and maximum gas flow for the system where the unit will operate. This is usually in standard cubic feet per minute, per hour or per day. Note the catalog units carefully and also that the reference standard temperature is usually 60°F for gas or vapor flow.
2. Use the rating selection charts or tables as per catalog instructions. For specific gravity or molecular weight different from the charts or tables, a correction factor is usually designated and should be used.
3. Note that some units use pipe line size for the separator size designation, others do not.
4. From the system operating pressure, establish the pressure rating designation for the separator selection.

Job No. _____	SPEC. DWG. NO.				
	A.	Page	/ of	Pages	
B/M No. _____		Unit Price			
		No. Units	/		
		Item No. 0-3			
DRUM OR TANK SPECIFICATIONS					
Service Size	Flash Drum and Separator 30" O.D. x 10'-0" Bend line				
Type					
DESIGN DATA					
Operating Pressure	35	PSI	G	Operating Temp.	-110 °F
Design Pressure	100	PSI	G	Design Temp.	-130 °F
Code	PSME	Stamp	YES	Lethal Const.	No
Materials:	Steel	Low temperature steel	Heads	Low temp. steel	Density of Contents 33 Lbs/cc. ft
Lining:	Metal	No	Rubber or Plastic	No	Supports Carbon steel
	Brick	No		Cement	No
Internal Corrosion Allowance	1/64"	Self Supporting	YES	Insulations:	<input checked="" type="checkbox"/> Yes <input type="checkbox"/> No
				Class	-110°F
NOZZLES					
Service	No. Req'd	Size	Press Class	Facing	Mark No.
Inlet	1	6"	150	RTJ	A
Vapor Out	1	6"	150	RTJ	B
Liquid Out	1	3"	150	RTJ	C
Drain					
Safety Valve	1	4"	150	RTJ	D
Level Control	2	2"	150	RTJ	E
Pressure Top	1	1"	6000	Coupling	F
Vent					
Gage Glass	2	2"	150	RTJ	G
Manhole	1	2"	150		
High Level Alarm	1	2"	150	RTJ	H
Low Level Alarm	1	2"	150	RTJ	I
REMARKS					
Follow Code, Provide convenient to wire mesh installation					
Wire Mesh to be Standard Weave, 6" thick; Material Type 304 Stainless steel.					
Provide support and top head-down for mesh.					
By	Chk'd	Rev.	Rev.	Rev.	Rev.
Date					
P.O. Tel. _____					

Figure 6-25 Specification design sheet for separator using wire mesh.

5. Note that most separators for pressure system operations are fabricated according to the American Society of Mechanical Engineers (ASME) code.
6. Specify special features and materials of construction, such as alloy or nonferrous impingement parts, or entire vessel if affected by process vapor and liquid. Specify special liquid reservoir at the base of the unit if necessary for system operations.

Line units normally have dump traps or liquid outlet of separator, while vessel type often uses some type of liquid level control.

7. Prepare a specification sheet (Figure 6-33).

Note that these units should not be connected in lines larger than their pipe inlet, since inlet velocity conditions are very important,

*Wire entrainment Mesh Specifications*

## A. Application Service

1. Source of Entrainment: \_\_\_\_\_

2. Operating Conditions: Give (1) Normal (2) Maximum (3) Minimum, where possible

Temperature \_\_\_\_\_

Pressure \_\_\_\_\_

Vapor Phase

Flow Rate \_\_\_\_\_

\*Velocity \_\_\_\_\_

Density \_\_\_\_\_ at operating conditions

Molecular Weight \_\_\_\_\_

Composition or Nature of Phase \_\_\_\_\_

Liquid Entrainment Phase

Quantity (if known) \_\_\_\_\_

Density \_\_\_\_\_

Viscosity \_\_\_\_\_

Surface Tension \_\_\_\_\_

Composition or Nature of Entrainment \_\_\_\_\_

Droplet Sizes or distribution (if known) \_\_\_\_\_

Solids Content (Composition and Quantity) \_\_\_\_\_

Dissolved \_\_\_\_\_

Suspended \_\_\_\_\_

## 3. Performance

Allowable Total Separator Pressure Drop \_\_\_\_\_

Allowable Mesh Pressure Drop \_\_\_\_\_

Allowable Entrainment \_\_\_\_\_

Mesh Thickness Recommended \_\_\_\_\_

## B. Construction and Installation

## 1. Vessel

\*Diameter I.D.: \_\_\_\_\_ Length: \_\_\_\_\_

Construction Material \_\_\_\_\_

\*Position (Horizontal, Vertical, Inclined) \_\_\_\_\_

\*Shape (Circular, Square, etc.) \_\_\_\_\_

Type (Evaporator, Still, Drum, etc.) \_\_\_\_\_

Existing or Proposed \_\_\_\_\_

## 2. Entrainment Mesh

Construction Material

Separator Mesh \_\_\_\_\_

Support Grid \_\_\_\_\_

Installation Method (Dimensions)

Vessel Open End \_\_\_\_\_

Manhole (size) \_\_\_\_\_

## C. Special Conditions \_\_\_\_\_

\* Assumes vessel size fixed prior to mesh inquiry

**Figure 6-26** Wire entrainment mesh specifications.

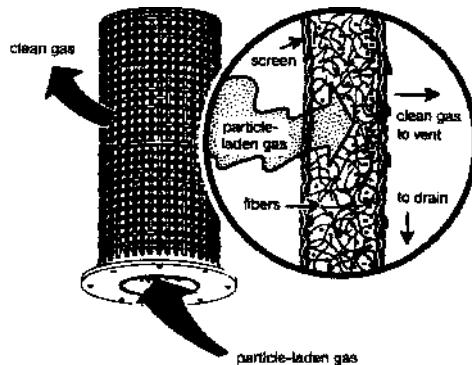
the swaging down or reduction tends to produce a jet effect by the gas upon the mist eliminator unit. This may erode the unit and cause erratic performance.

**Dry-Packed Impingement Beds.** Although this type of unit is not used as frequently as most of the others discussed, it does have some specific applications in sulfuric acid mist removal and similar very difficult applications. The unit consists of a bed of granular particles or ceramic packing, sometimes graduated in size, operating dry as far as external liquid application to aid in the separation. The superficial velocities of 0.5–8 ft/s through the unit are rather low for most separators; therefore the vessels become

large. Due to the packed heights of 2 ft (minimum) and higher, the pressure drop can be appreciable. Particle removal may be 0.5–5  $\mu$  at 99% efficiency for a good design. These units will plug on dust service and must be back washed to regain operability at reasonable pressure drops.

**6.12 CENTRIFUGAL SEPARATORS**

Centrifugal separators utilize centrifugal action for the separation of materials of different densities and phases. They are built in (a) stationary and (b) rotary types. Various modifications of stationary units are used more than any other kind for separation



#### Materials of construction

Packing of York-Fiberbed high efficiency mist eliminators consists of ceramic, glass, polypropylene, fluoropolymer fibers. Cages and frames are fabricated from all stainless steels and other weldable alloys as well as FRP.

**Figure 6-27a** Details of a cylindrical York-Fiberbed® mist eliminator. (Courtesy of Otto H. York Co. Inc., Bulletin in 55B.)

problems. The cost is moderate; it is simple in construction and is reasonably flexible in service, being useful for gas-liquid or gas-solid systems. In addition to serving as finishing separators centrifugal units are also used to take a "rough cut" into a separation problem. They may be followed by some additional unit of special cyclone action or filtration through woven cloth pads, and so on, to completely remove last traces of entrained particles.

#### STATIONARY VANE

The stationary vane type is quite popular and is adapted to many applications. It is used in vessels or pipe lines as illustrated in Figures 6-34–6-37d. They are usually of high efficiency for both liquid and solid particles such as rust, scale, dirt, and so on. When the system is dry with dust a special design is used.

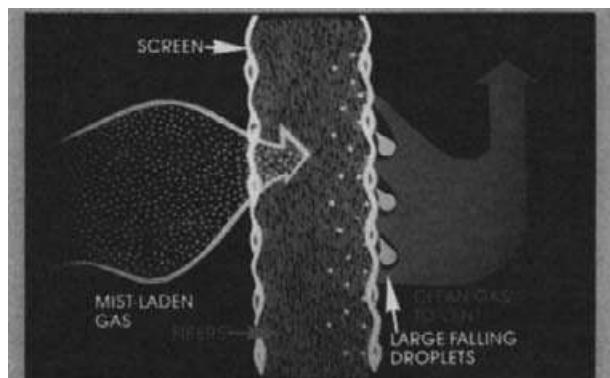
#### EFFICIENCY

The efficiency of centrifugal units is given below.

Type	Efficiency Range
High velocity	99% or higher, of entering liquid.
Stationary vanes	Residual entrainment 1 ppm or less
Cyclone	70–85% for 10 $\mu$ , 99% for 40 $\mu$ and larger. For high entrainment, efficiency increases with concentration.
Rotary	98% for agglomerating particles.

#### GAS-LIQUID SEPARATORS

The process engineer is often required to design separators or knockout drums for removing liquids from process gas streams. The vessels used for processing in the chemical process industry (CPI) are principally of two kinds: those that are without internals



#### Available FLEXIFIBER® Mist Eliminator Styles

KOCH Type	Primary Collection Mechanism	Collection Efficiency		Element Pressure Drop (Inches WG)	Bed Velocity (Ft/Min)
		Particle Size (Microns)	Efficiency (%)		
BD	Brownian Diffusion	>3	Essentially 100	2–20	5–40
		<3	Up to 99.95		
IC	Impaction	>3	Essentially 100	7–9	250–450
		1–3	95–99 +		
		>3	Essentially 100		
IP	Impaction	1–3	85–97	5–7	400–500
		0.5–1	50–85		
		>3	Essentially 100		
IS	Impaction	<3	15–30	1–2	400–500
		>3	Essentially 100		

NOTE: Since Types IC, IP and IS operate primarily by impaction, the above collection efficiencies drop off at gas flows below about 75% of design rates and depend on the specific gravity of the collected liquid.  
Flexifiber® Type BD: Normally cylindrical in shape and available in a wide variety of materials and sizes. Bulk packed elements are Mark I series.

Also available as wound beds (Mark II or III) for lower initial cost and ease of repacking, and with equivalent collection efficiency at equal or lower pressure drop.

Flexifiber® Type IC: Normally cylindrical in shape and available in a variety of materials and sizes.

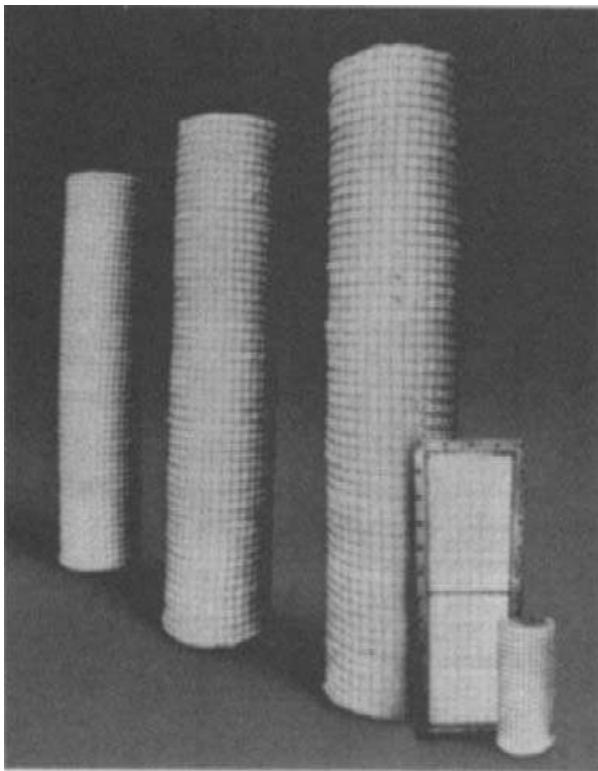
Flexifiber® Types IP and IS: Normally rectangular in shape and available in various metals.

**Figure 6-27b** Fiber-pack® mist eliminator pack separators. (By permission from Kock Engineering Co., Inc.)

Note that other manufacturers have basically the same concept; however, the identification of types are peculiar to each

and those with internals. Empty separators are drums that provide intermediate storage or surge of a process stream for a limited or extended period. Alternatively, they provide phase separation by settling.

The second category comprises equipment such as reactors, mixers, distillation columns, and heat exchangers. In some cases, it is important to separate liquid and gas flowing simultaneously through a pipe. This is because the conditions of the flowing mixture and the efficiency of separation may vary widely. Therefore, a separator for such duty must be adequate. In addition, there are constraints due to space or weight that often affect the choice of separators, the need to handle solids or effect a three-phase separation, and the requirements for liquid holdup. In practice, most separation problems are solved by the following types of separating equipment.

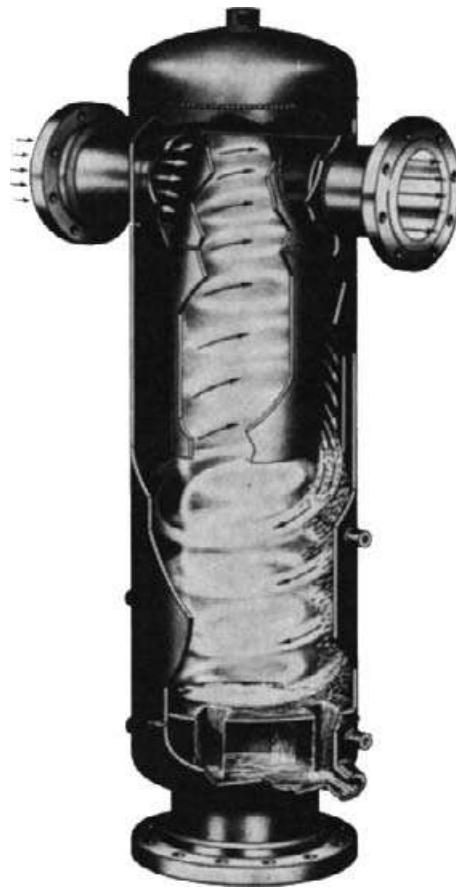


**Figure 6-27c** Typical Fiberbed mist eliminators are available in both candle and panel configurations. (By permission, Otto H. York Co. Inc.)

**Knockout or Surge Drums.** A knockout drum (Figures 6-38 and 6-39) is suitable for a bulk separation of gas and liquid, particularly when the liquid volume fraction is high with stratified or plug flow in the pipe. Also, it is useful when vessel internals are required to be kept to a minimum, for example, in relief systems or in fouling service. It is unsuitable if a mist is being separated or if high separating efficiency is required.

**Cyclones.** These are robust and not susceptible to fouling or wax (Figure 6-40). A multicyclone separator (Figure 6-41) is compact and is reasonably efficient for foam, but not for slugs. The efficiency of a cyclone decreases with increasing diameter, and cyclones are not applicable above 5 ft diameter. They possess a good efficiency for smaller flow rates. However, cyclones are expensive to operate, especially under vacuum conditions, because they have a higher pressure drop than either a knockout drum or a demister separator.

**Demister Separators.** A demister separator is fitted either with a vane demister package or with a wire demister mat. The latter type is much preferred, although it is unsuitable for fouling service. The wire mesh demister is a widely applied type of separator, and is adequate for all gas–liquid flow regimes over a wide range of gas flow rates. A knockout drum or demister separator may be either a vertical or a horizontal vessel. A vertical vessel is generally preferred because its efficiency does not vary with the liquid level. Alternatively, a horizontal vessel is chosen when it offers a clear size advantage, if the headroom is restricted or if a three-phase separation is required. Knockout drums and cyclones are recommended for waxy and coking feeds. Demister mats are not suitable because of the danger of plugging. Vane demister packages are used as alternatives, but provision should be made for cleaning.



**Figure 6-28** Wall-wiping centrifugal type separator. (Courtesy of Wright-Austin Co.)

#### SIZING OF VERTICAL AND HORIZONTAL SEPARATORS

**Vertical Separators.** Vertical liquid–vapor separators are used to disengage a liquid from a vapor when the volume of liquid is small compared with that of vapor volume. A typical example of a vertical liquid separator with small surge volume is a compressor knockout drum. The maximum allowable vapor velocity in a vertical separator that reduces the liquid carryover is dependent on the following:

- Liquid and vapor densities.
- A constant  $K$  based on surface tension, droplet size, and physical characteristics of the system.

The proportionality constant  $K$  is 0.35 for oil and gas systems with at least 10 in. disengaging height between the mist eliminator bottom and gas–liquid interface. For vertical vessels,  $K$  can vary between 0.1 and 0.35 if mist eliminators (demisters) are used to enhance disentrainment. The value of  $K$  also depends on the operating pressure of the vessel. At pressures above 30 psig, the  $K$  value decreases with pressure with an approximate value of 0.30 at 250 psig and 0.275 at 800 psig. Watkins [34] has developed a correlation between the separation factor and  $K$ . Figure 6-42 illustrates Watkins's vapor velocity factor chart, based on the fact that 5% of the liquid is entrained with the vapor.

Branan [35] has developed a polynomial equation using Watkins's data to calculate the  $K$  value for a range of separating factors between 0.006 and 5.0. Watkins proposed a method

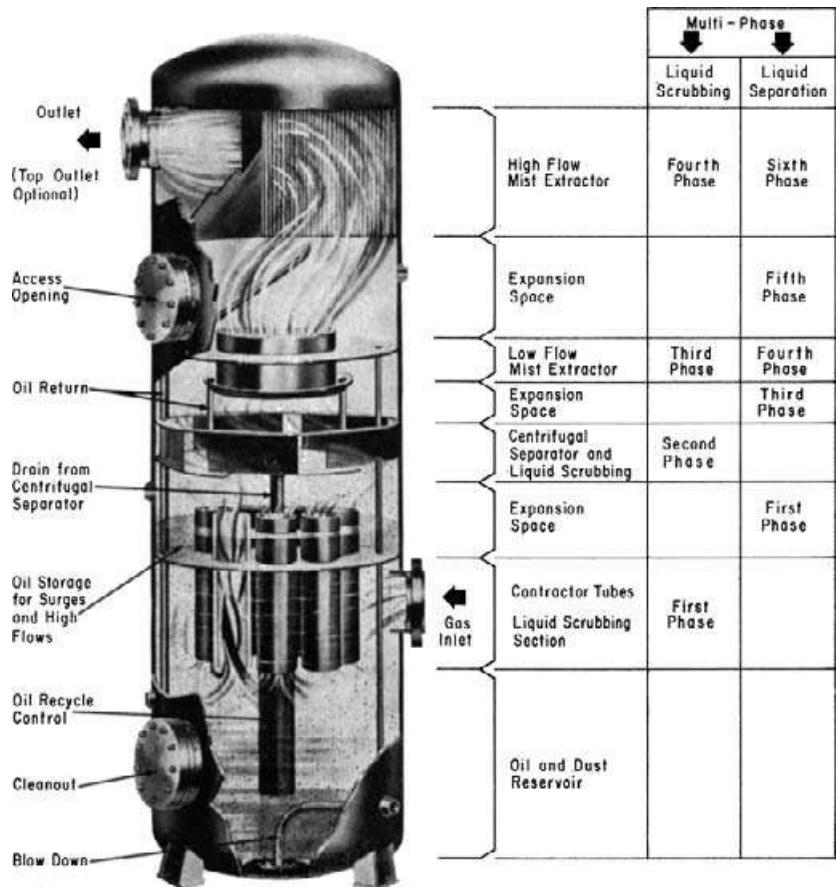


Figure 6-29 Multiphase gas cleaner. (Courtesy of Blaw-Knox Co.)

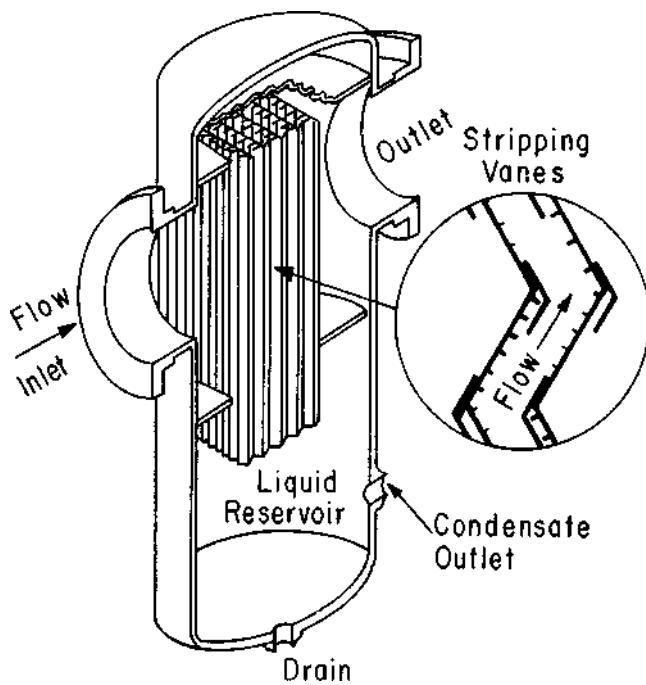


Figure 6-30 Impingement separator. (Courtesy of Peerless Manufacturing Co.)

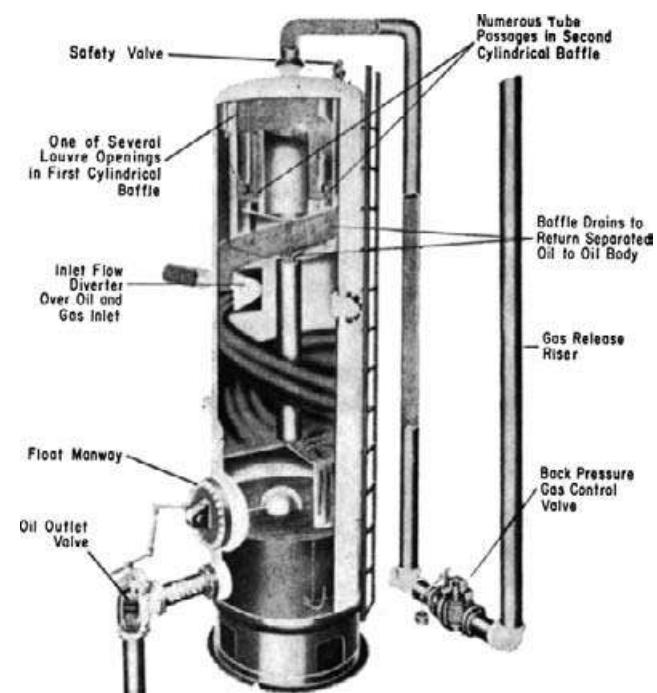


Figure 6-31 Combination separator. (Courtesy of National Tank Co.)

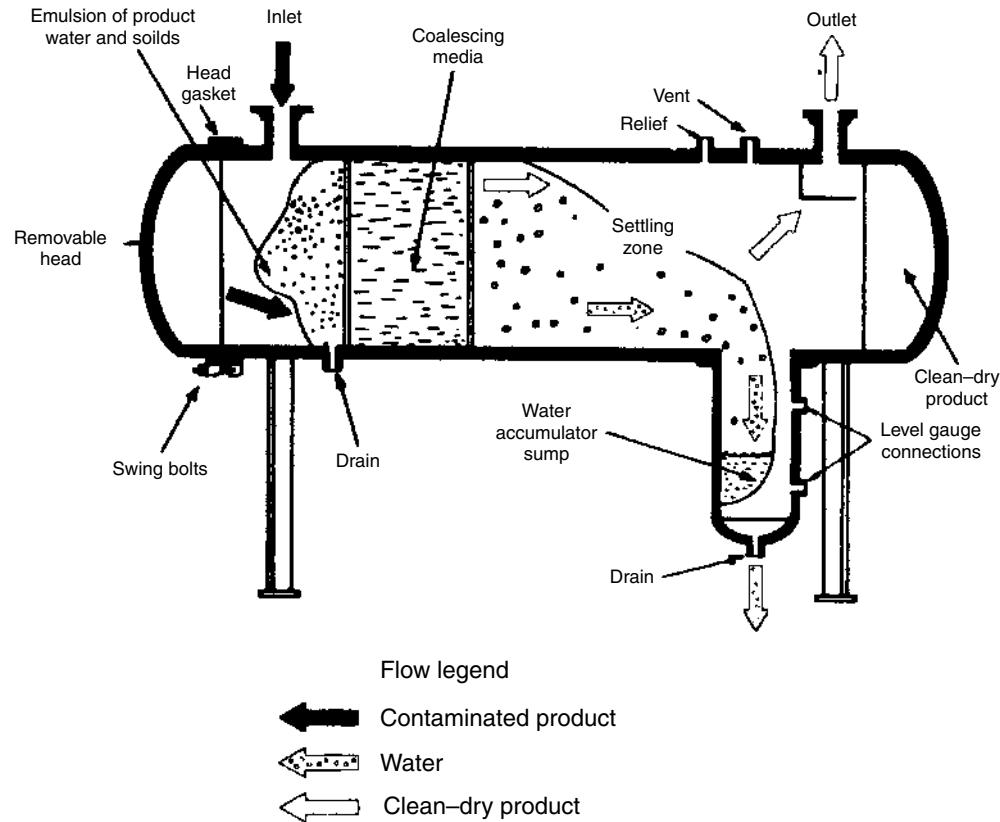


Figure 6-32 Typical coalescer unit. (By permission from Facet Enterprises Inc., Industrial Div.)

for sizing reflux drums based upon several factors, as illustrated in Tables 6-12 and 6-13. Table 6-12 gives the recommended design surge times. Table 6-13 gives the multiplying factors for various operator efficiencies. The operating factor is based upon the external unit and its operation, its instrumentation and response to control, the efficiency of labor, chronic mechanic problems, and the possibility of short- or long-term interruptions [34].

The multiplying factors  $F_1$  and  $F_2$  represent the instrument and the labor factors. A multiplying factor  $F_3$  is applied to the net overhead product going downstream. Multiplying factor  $F_4$  depends on the kind and location of level indicators. It is recommended that 36 in. plus half the feed nozzle OD (48 in. minimum) be left above the feed nozzle for vapor. Below the feed nozzle, allowance of 12 in. plus half the feed nozzle OD (18 in. minimum) is required for clearance between the maximum liquid level and the feed nozzle. At some value between  $L/D$  ratios of 3 and 5, a minimum vessel weight will occur resulting in minimum costs for the separator. Figure 6-43 shows the dimensions of a vertical separator.

#### CALCULATION METHOD FOR A VERTICAL DRUM

The following steps are used to size a vertical drum.

##### 1. Calculate the vapor–liquid separation factor.

$$S. \text{ Fac} = \frac{W_L}{W_V} \left( \frac{\rho_V}{\rho_L} \right)^{0.5} \quad (6-51)$$

2. From correlation, determine the design vapor velocity factor  $K_v$  and the maximum design vapor velocity [35].

$$X = \ln (S. \text{ Fac}) \quad (6-52)$$

$$K_v = \exp (A + BX + CX^2 + DX^3 + EX^4 + FX^5) \quad (6-53)$$

where

$$\begin{aligned} A &= -1.942936 \\ B &= -0.814894 \\ C &= -0.179390 \\ D &= -0.0123790 \\ E &= 0.000386235 \\ F &= 0.000259550 \end{aligned}$$

$$V_{\max} = K_v \left( \frac{\rho_L - \rho_V}{\rho_V} \right)^{0.5} \quad (6-54)$$

3. Calculate the minimum vessel cross-sectional area.

$$Q_v = \frac{W_V}{3600(\rho_V)}, \text{ ft}^3/\text{s} \quad (6-55)$$

where  $Q_v$  = volumetric flow rate,  $\text{ft}^3/\text{s}$ .

$$A_v = \frac{Q_v}{V_{\max}}, \text{ ft}^2 \quad (6-56)$$

*Baffle Type Separator Specifications*Separator Application (Give Service) \_\_\_\_\_Design Operating Conditions:

Main Stream Flow Rate \_\_\_\_\_ Sp. Gr. Or Mol. Wt. \_\_\_\_\_

Entrained Material rate (if known) Source of entrainment \_\_\_\_\_

Min. Pressure \_\_\_\_\_ psi (g) or (a), Max. Temp. \_\_\_\_\_ °F.

Max. Pressure \_\_\_\_\_ psi (g) or (a)

Entrained Particle size \_\_\_\_\_ (mesh) (Microns)

Vessel Specifications:

Design Pressure \_\_\_\_\_ psii \_\_\_\_\_ Design Temp. \_\_\_\_\_ °F.

Code: API-ASME \_\_\_\_\_ ASME 1949 Ed. \_\_\_\_\_ ASME 1950 Ed. \_\_\_\_\_

X-Ray \_\_\_\_\_ Stress Relief \_\_\_\_\_

Corrosion allowance \_\_\_\_\_

Dimensions \_\_\_\_\_ "O.D. x \_\_\_\_\_ " long bend line to bend line

Base Support \_\_\_\_\_

Mist Extractor \_\_\_\_\_, Mat'l. of Construction \_\_\_\_\_

Connections:

1. Gas inlet and outlet: (Size, ASA pressure rating, type flange)

2. Liquid Outlet \_\_\_\_\_

3. Liquid Level Gage \_\_\_\_\_

4. Liquid Level Control \_\_\_\_\_

5. Pressure Gauge \_\_\_\_\_

6. Relief Valve \_\_\_\_\_

7. Bursting Disc \_\_\_\_\_

8. High Level Alarm \_\_\_\_\_

9. Low Level Alarm \_\_\_\_\_

10. Thermometer \_\_\_\_\_

11. Equalizer \_\_\_\_\_

12. Drain \_\_\_\_\_

13. Others: \_\_\_\_\_ (specify) \_\_\_\_\_

Special Features: \_\_\_\_\_**Figure 6-33** Baffle type separator specifications.

- 4.** Set a vessel diameter based on 6-in. increments and calculate cross-sectional area.

$$D_{\min} = \left( \frac{4 \cdot A_v}{\pi} \right)^{0.5}, \text{ ft} \quad (6-57)$$

$D = D_{\min}$  to next largest, 6 in.

$$\text{Area} = \frac{\pi D_{\min}^2}{4} \quad (6-58)$$

- 5.** Estimate the vapor-liquid inlet nozzle based on the following velocity criteria.

$$(U_{\max})_{\text{nozzle}} = \frac{100}{\rho_{\text{mix}}^{0.5}}, \text{ ft/s} \quad (6-59)$$

$$(U_{\min})_{\text{nozzle}} = \frac{60}{\rho_{\text{mix}}^{0.5}}, \text{ ft/s} \quad (6-60)$$

where

 $U_{\max}$  = maximum velocity, ft/s $U_{\min}$  = minimum velocity, ft/s.

$$\rho_{\text{mix}} = \left( \frac{\frac{W_L + W_V}{W_L}}{\frac{W_L + W_V}{\rho_L}} \right), \text{ lb/ft}^3 \quad (6-61)$$

- 6.** From Figure 6-43, make a preliminary vessel sizing for the height above the centerline of a feed nozzle to top seam. Use 36 in. plus half the feed nozzle OD or 48 in. minimum. Use 12 in. plus half the feed nozzle OD or 18 in. minimum to determine the distance below the centerline of the feed nozzle to the maximum liquid level.
- 7.** From Table 6-12 or 6-13, select the appropriate full surge volume in seconds. Calculate the required vessel volume.

$$Q_L = \frac{W_L}{3600(\rho_L)}, \text{ ft}^3/\text{s} \quad (6-62)$$

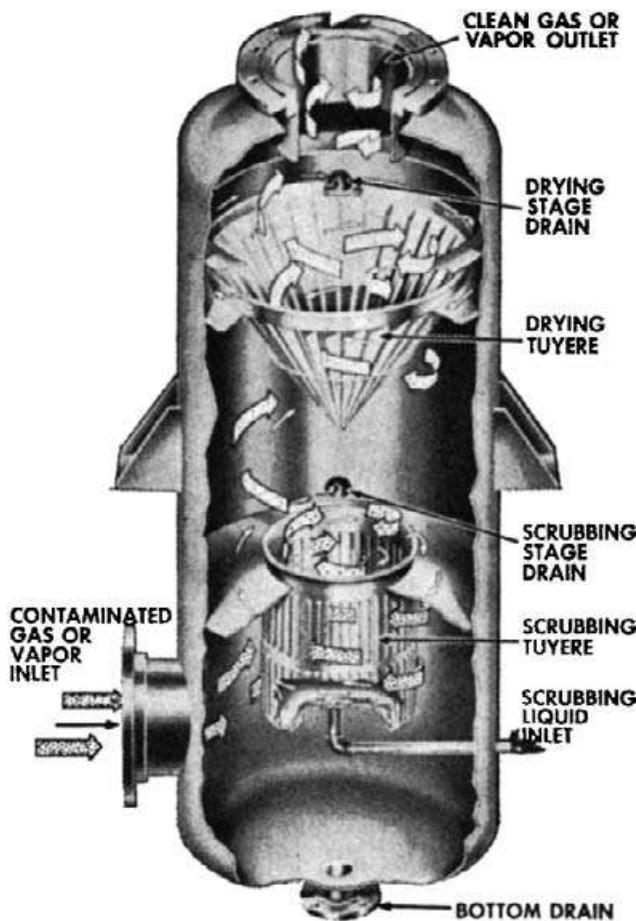


Figure 6-34 Scrubber with internal liquid feed. (Courtesy of Centrifix Corp.)

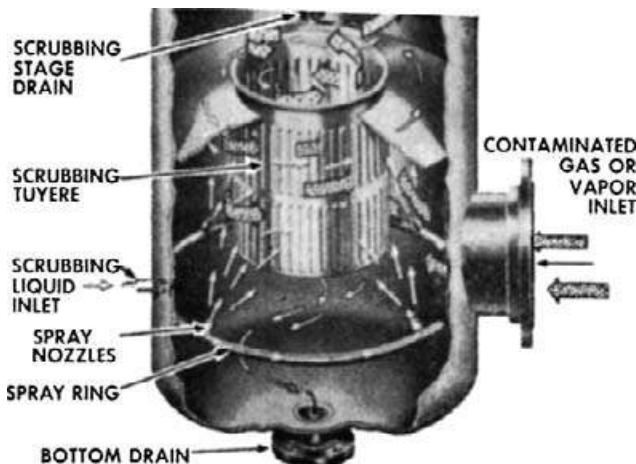


Figure 6-35 Scrubber with spray ring as alternate arrangement for Figure 6-34. (Courtesy of Centrifix Corp.)

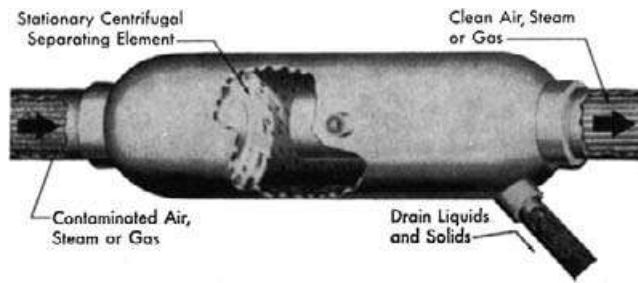


Figure 6-36 Line-type centrifugal separator. (Courtesy of V.D. Anderson Co.)

$$\begin{aligned} V &= (Q_L) (\text{design time to fill}) \text{ ft}^3 & (6-63) \\ &= (60) (Q_L) (T), \text{ ft}^3 \end{aligned}$$

The liquid height is

$$H_L = V \left( \frac{4}{\pi D^2} \right), \text{ ft} \quad (6-64)$$

8. Check geometry.  $(H_L + H_V)/D$  must be between 3 and 5, where  $H_V$  is the vapor height in feet. For small volumes of liquid, it may be necessary to provide more liquid surge than is necessary to satisfy the  $L/D > 3$ . Otherwise this criterion should be observed. If the required liquid surge volume is greater than that obtained in a vessel having  $L/D < 5$ , a horizontal drum must be provided.

#### CALCULATION METHOD FOR A HORIZONTAL DRUM

Horizontal vessels are used for substantial vapor-liquid separation where the liquid holdup space must be large. Maximum vapor velocity and minimum vapor space are determined as in the vertical drum, except that  $K_H$  for horizontal separators is generally set at 1.25  $K_v$ . The following steps are carried out in sizing horizontal separators.

1. Calculate the vapor-liquid separation factor by Eq. (6-51) and  $K_v$  by Eqs. (6-52) and (6-53).

2. For horizontal vessels

$$K_H = 1.25 K_v \quad (6-65)$$

3. Calculate the maximum design vapor velocity.

$$(U_V)_{\max} = K_H \left( \frac{\rho_L - \rho_V}{\rho_V} \right)^{0.5} \text{ ft/s} \quad (6-66)$$

4. Calculate the required vapor flow area.

$$(A_V)_{\min} = \frac{Q_v}{(U_V)_{\max}} \text{ ft}^2 \quad (6-67)$$

**EXAMPLE 6-6**

Size a vertical separator under the following conditions:

$$W_L = 5000 \text{ lb/h} \quad \rho_L = 61.87 \text{ lb/ft}^3$$

$$W_V = 37,000 \text{ lb/h} \quad \rho_V = 0.374 \text{ lb/ft}^3$$

The liquid surge time is 5 min.

*Solution*

Calculate the vapor–liquid separation factor using Eq. (6-51).

$$\text{S. Fac} = \frac{5000}{37,000} \left( \frac{0.374}{61.87} \right)^{0.5} = 0.0105$$

From correlation, determine the design vapor velocity factor  $K_v$  and the maximum design vapor velocity [35].

From Eq. (6-52),

$$\begin{aligned} X &= \ln(0.0105) \\ &= -4.55574 \end{aligned}$$

From Eq. (6-53),

$$\begin{aligned} K_v &= \exp[-1.942936 + (-0.814894)(-4.55575) \\ &\quad + (-0.17939)(-4.55575)^2 \\ &\quad + (-0.012379)(-4.55575)^3 \\ &\quad + (0.000386235)(-4.55575)^4 \\ &\quad + (0.00025955)(-4.55575)^5] \\ K_v &= [-1.942936 + 3.712453 - 3.723214 + 1.170483 \\ &\quad + 0.166376 - 0.509355] \\ K_v &= \exp[-1.126193] = 0.32426 \end{aligned}$$

From Eq. (6-54),

$$V_{\max} = 0.324 \left( \frac{61.87 - 0.374}{0.374} \right)^{0.5} = 4.158 \text{ ft/s}$$

Calculate the minimum vessel cross-sectional area using Eq. (6-55).

$$Q_v = \frac{37,000}{3600(0.374)} = 27.48 \text{ ft}^3/\text{s}$$

From Eq. (6-56),

$$A_v = \frac{27.48}{4.158} = 6.609 \text{ ft}^2$$

Set a vessel diameter based on 6-in. increments and calculate cross-sectional area.

From Eq. (6-57),

$$D_{\min} = \left( \frac{4 \times 6.609}{\pi} \right)^{0.5} = 2.9 \text{ ft}$$

$D = D_{\min}$  to next largest, 6 in.

The density of the gas–liquid mixture  $\rho_{\text{mix}}$  from Eq. (6-61) is

$$\rho_{\text{mix}} = \left( \frac{[5,000 + 37,000]}{\left[ \frac{5,000}{61.87} \right] + \left[ \frac{37,000}{0.374} \right]} \right), \text{ lb/ft}^3 = 0.424 \text{ lb/ft}^3$$

Estimate the vapor–liquid inlet nozzle using Eqs (6-59) and (6-60).

$$(U_{\max})_{\text{nozzle}} = \frac{100}{(0.424)^{0.5}} = 153,54 \text{ ft/s}$$

$$(U_{\min})_{\text{nozzle}} = \frac{60}{(0.424)^{0.5}} = 92.14 \text{ ft/s}$$

Calculate the required vessel volume using Eq. (6-62).

$$Q_L = \frac{5,000}{(3600 \times 61.87)} = 0.0224 \text{ ft}^3/\text{s}$$

From Eq. (6-63) the vessel volume  $V$  is

$$V = (60)(Q_L)(T) \text{ ft}^3$$

$$V = (60)(0.0224)(5)$$

$$= 6.73 \text{ ft}^3$$

The liquid height  $H_L$  is calculated using Eq. (6-64).

$$H_L = (6.72) \left( \frac{4}{\pi \times 2.9^2} \right) = 1.017 \text{ ft}$$

The Excel spreadsheet Example 6-6.xls has been developed for sizing gas–liquid vertical separator.

5. Select appropriate design surge times from Table 6-12 or Table 6-13 and calculate full liquid volume by Eqs (6-62) and (6-63). The remainder of the sizing procedure is carried out by trial and error as follows.

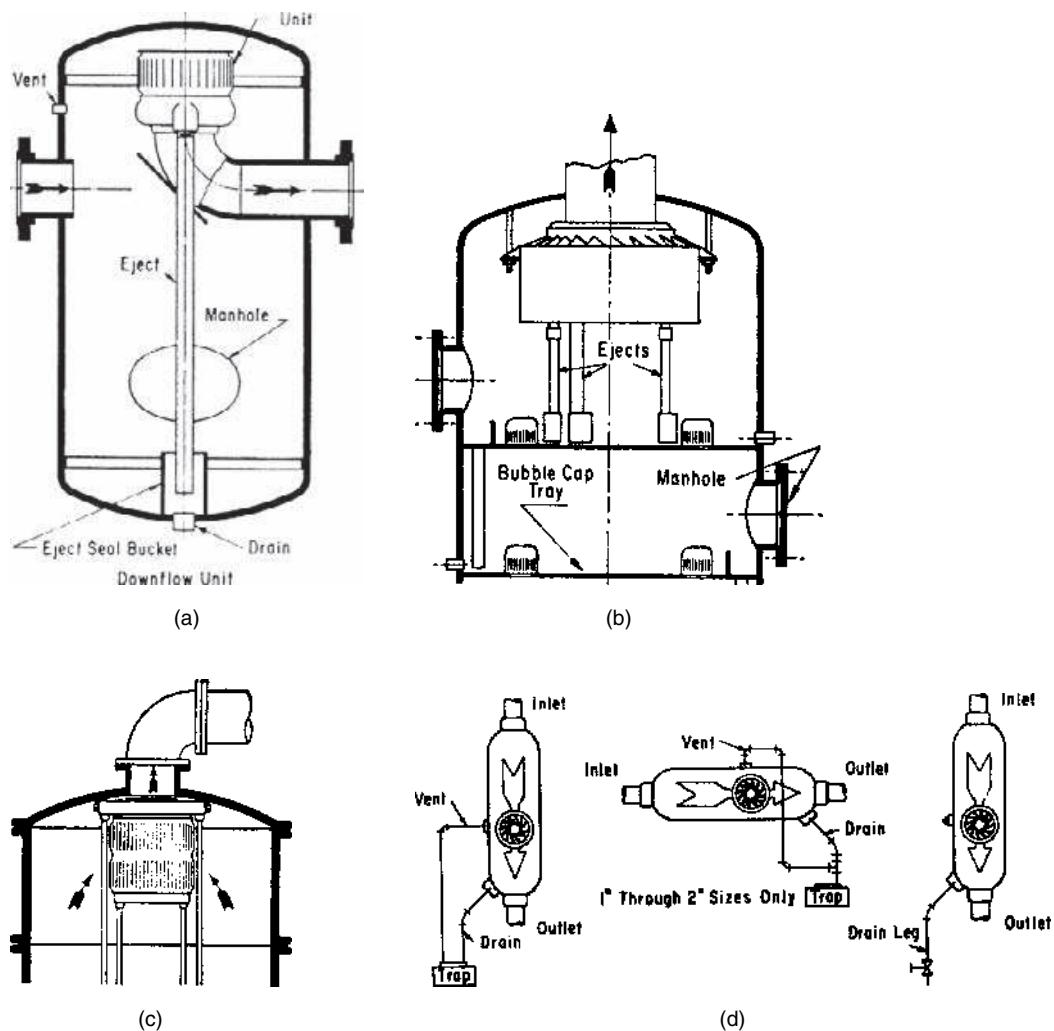
6. When the vessel is full, the separator vapor area can be assumed to occupy only 15–25% of the total cross-sectional area. Here, a value of 20% is used and the total cross-sectional area is expressed as

$$(A_{\text{Total}})_{\min} = \frac{(A_v)_{\min}}{0.2} \quad (6-68)$$

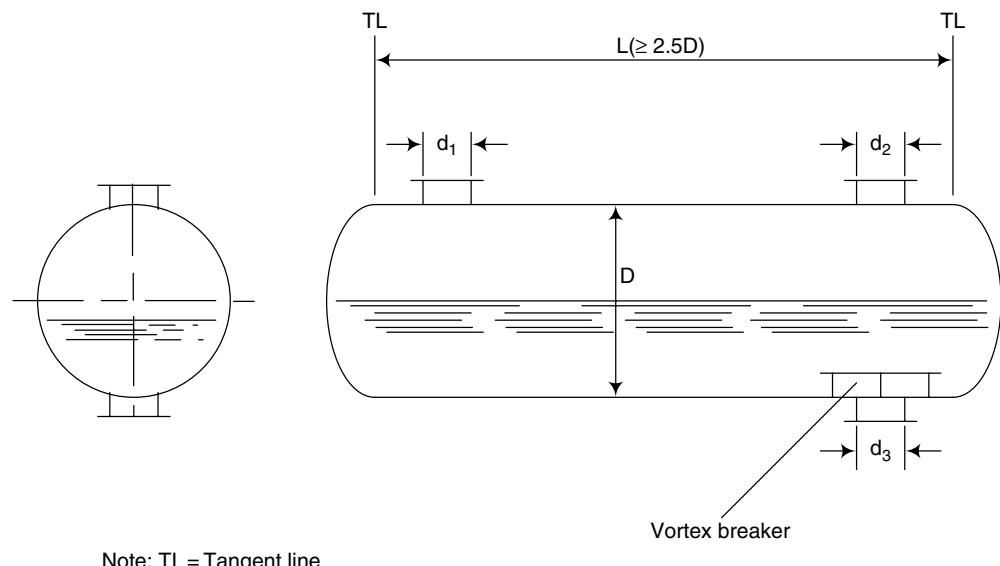
$$D_{\min} = \left( \frac{4(A_{\text{total}})_{\min}}{\pi} \right)^{0.5} \quad (6-69)$$

7. Assume the length-to-diameter ratio of 3 (i.e.,  $L/D = 3$ ). Calculate the vessel length.

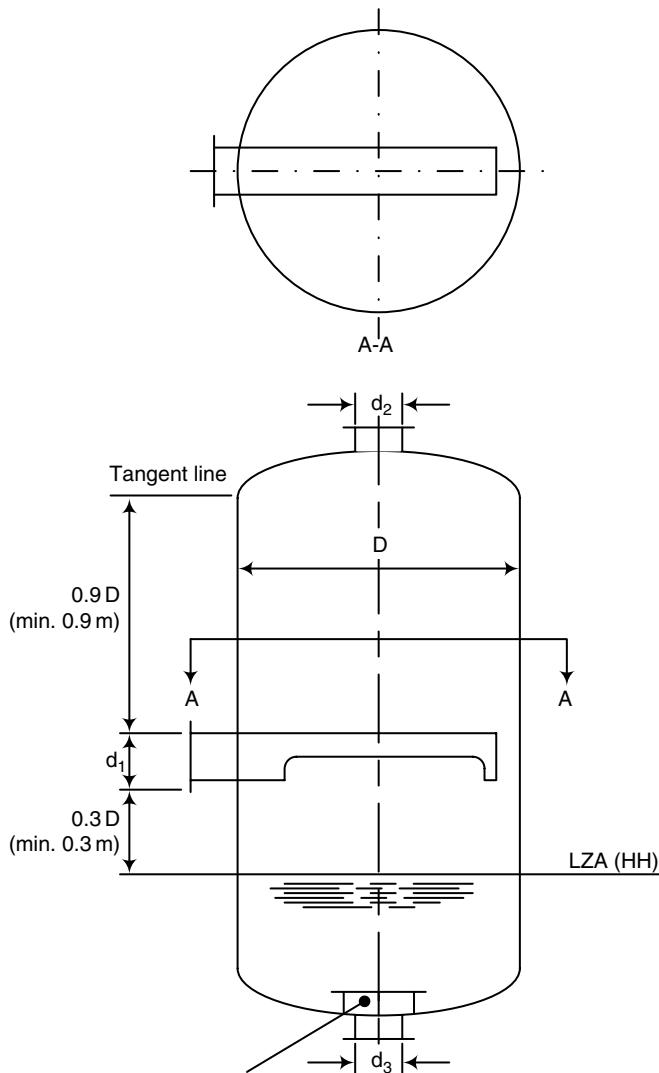
$$L = (3)(D_{\min}) \quad (6-70)$$



**Figure 6-37** Centrifugal separator applications. (By permission from Centrifix Corp.)



**Figure 6-38** Horizontal knock-out drum.



**Figure 6-39** Vertical knockout drum.

8. Because the vapor area is assumed to occupy 20% of the total cross-sectional area, the liquid area will occupy 80% of the total cross-sectional area.

$$A_L = (0.8)(A_{\text{total}})_{\text{min}} \text{ ft}^2 \quad (6-71)$$

9. Calculate the vessel volume.

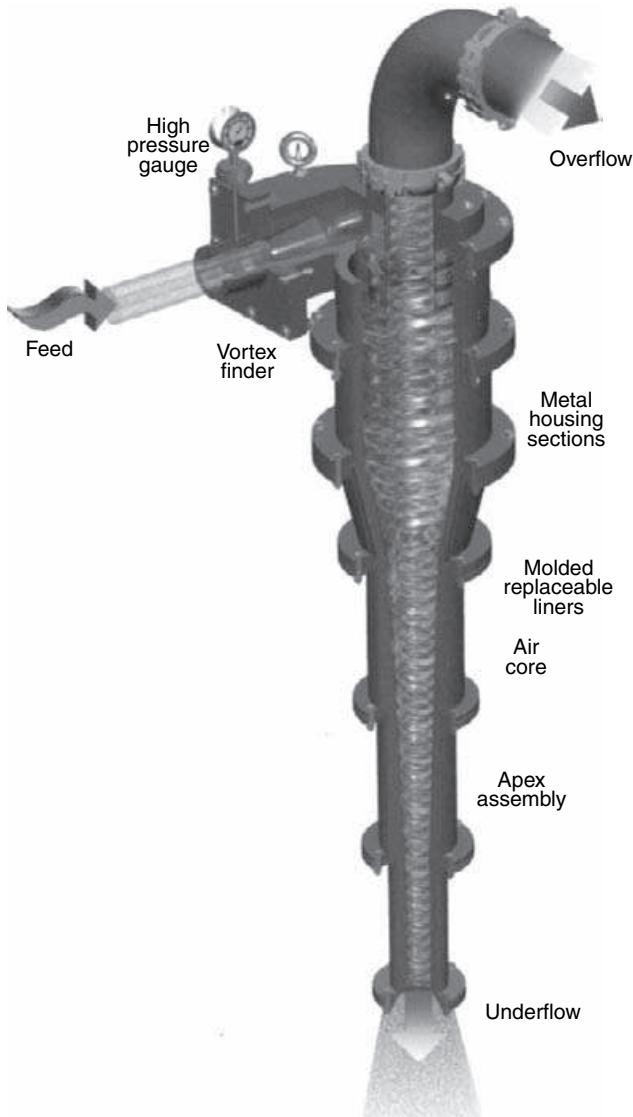
$$V_{\text{VES}} = (A_{\text{total}})_{\text{min}} (L) \text{ ft}^3 \quad (6-72)$$

10. Calculate liquid surge time.

$$T = \frac{(60)(A_L)(L)(\rho_L)}{W_L} \text{ min} \quad (6-73)$$

11. If  $5 < L/D < 3$ , resize.

If there is water to be settled and withdrawn from hydrocarbon, the water's settling time requirement needs to be checked. The



**Figure 6-40** A cyclone separator. (By permission from Krebs Engineers.)

water settling requirement rather than other process considerations might set the liquid surge capacity. Therefore, the liquid surge capacity that has been previously estimated from tables might have to be increased.

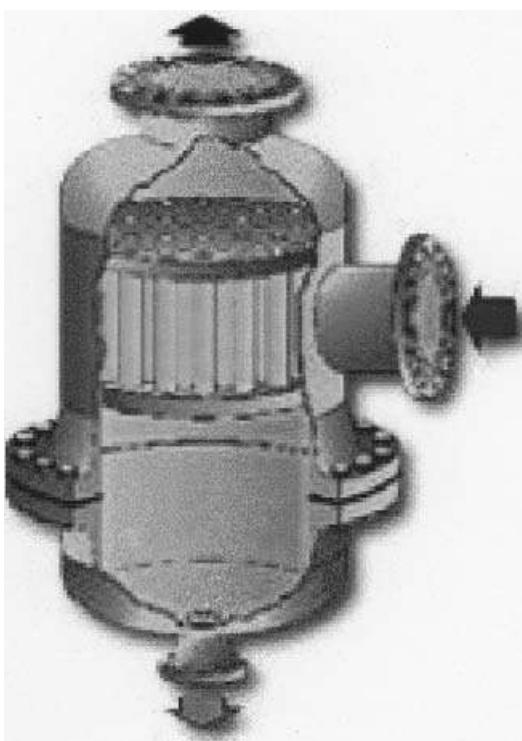
*The following is a quick check for water settling [35]:*

1. Estimate the water terminal settling velocity using

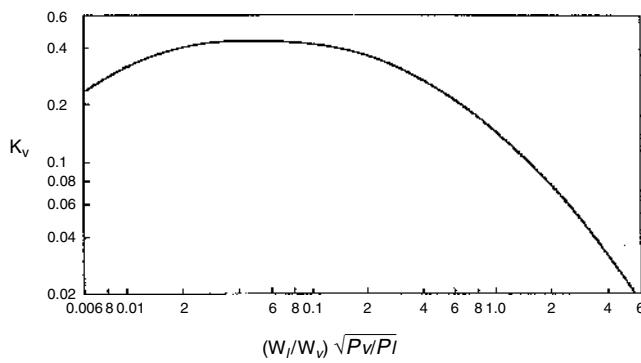
$$V_t = 44.7 \times 10^{-8} \frac{(\rho_w - \rho_o) F_s}{\mu_o} \quad (6-74)$$

where

- $V_t$  = terminal settling velocity, ft/s
- $F_s$  = correction factor for hindered settling
- $\rho_w$  = density of water, lb/ft<sup>3</sup>
- $\rho_o$  = density of oil, lb/ft<sup>3</sup>
- $\mu_o$  = absolute viscosity of oil, lb/ft.s.



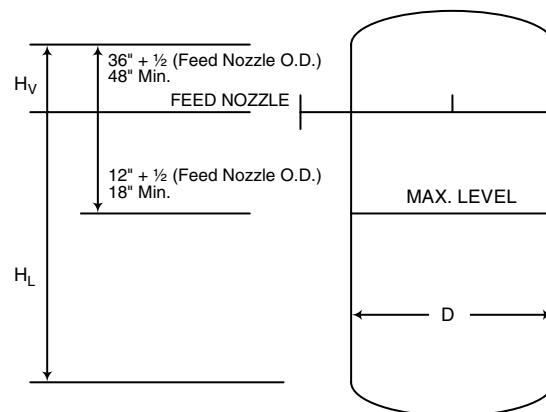
**Figure 6-41** A Multicyclone separator. (By permission from Krebs Engineers.)



**Figure 6-42** Design vapor velocity factor for vertical vapor-liquid separator at 85% of flooding.  
(Source: Watkins [34].)

**TABLE 6-13 Operation Factors for External Units**

Operating Characteristics	Factor $F_3$
Under good control	2
Under fair control	3
Under poor control	4
Feed to or from storage	1.25
Factor $F_4$	
Board mounted level recorder	1.0
Level indicator on board	1.5
Gauge glass at equipment only	2.0



**Figure 6-43** Dimensions of a vertical separator.

This assumes a droplet diameter of 0.0005 ft. (152 µm)  $F_s$  is determined from

$$F_s = \frac{X^2}{10^{1.82(1-X)}} \quad (6-75)$$

where  $X$  = volume fraction of oil.

2. Calculate the modified Reynolds number  $Re$  from

$$Re = 5 \times 10^{-4} \frac{\rho_o V_t}{\mu_o} \text{ (usually } < 1.0 \text{)} \quad (6-76)$$

This assumes a droplet diameter of 0.0005 ft. (152 µm)

3. Calculate  $V_s/V_t$  from

$$V_s/V_t = A + B(\ln Re) + C(\ln Re^2) + D(\ln Re^3) + E(\ln Re^4) \quad (6-77)$$

where

$V_s$  = actual settling velocity, ft/s

$A = 0.919832$

$B = -0.091353$

$C = -0.017157$

$D = 0.0029258$

$E = -0.00011591$ .

4. Calculate the length of the settling section (Figure 6-44a) as

$$L = \frac{hQ}{AV_s} \quad (6-78)$$

**TABLE 6-12 Design Criteria for Reflux Distillate Accumulators**

Operation	Minutes				
	Instrument Factor $F_1$		Labor Factor $F_2$		
	w/Alarm	w/o Alarm	Good	Fair	Poor
FRC	1/2	1	1	1.5	2
LRC	1	1 1/2	1	1.5	2
TRC	1 1/2	2	1	1.5	2

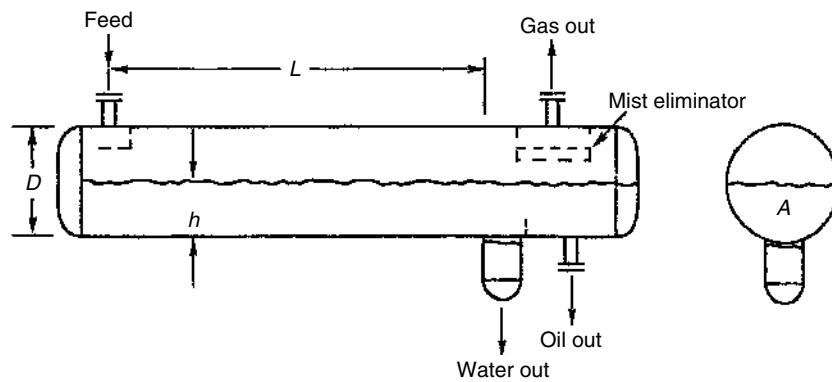
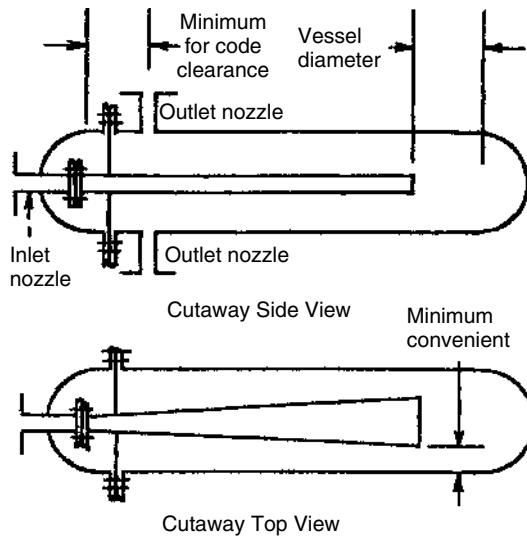


Figure 6-44a Typical vapor-liquid separator [35].



*Note:*

1. For large vessels, a manway and internally dismantled cone may be used where more economical.
2. Inlet and outlet nozzles should be sized for pump discharge.
3. Gauge glass and level instruments should be located at inlet-outlet end.
4. Mechanical design should suit economy under operating conditions.

Figure 6-44b Recommended design for a liquid-liquid separator [36].

where

$$L = \text{length of settling zone, ft}$$

$$h = \text{height of oil, ft}$$

$$Q = \text{flow rate, ft}^3/\text{s}$$

$$A = \text{cross sectional area of the oil settling zone, ft}^2$$

This allows the water to fall out and be drawn off at the bootleg before leaving the settling section.

#### LIQUID-LIQUID SEPARATION

The separation of immiscible liquids is an essential chemical/process engineering operation. Barton [36] provides the following empirical method, which has proven satisfactory for sizing liquid-liquid phase separators.

**1. Calculate the holdup time as**

$$T = 0.1 \left[ \frac{\mu}{(\text{sp gr}_b - \text{sp gr}_t)} \right] \quad (6-79)$$

where

$$T = \text{holdup time, h}$$

$$\mu = \text{viscosity of the continuous phase, cP}$$

$$\text{sp gr}_b = \text{specific gravity of the bottom phase}$$

$$\text{sp gr}_t = \text{specific gravity of the top phase.}$$

2. Assign a length-to-diameter ratio of 5, and size a tank to accommodate the required holdup time.
3. Provide inlet and outlet nozzles at one end, and an internal flat cone (Figure 6-44b).

The relation between viscosity and specific gravity difference of the phases corresponds to those of the equations for terminal settling velocity in the Stokes' law region and free-settling velocity of isomeric particles. Also, the dimensions of the tank and cone recognize that the shape of turbulence created by nozzles discharging into liquids spreads at an angle whose slope is about 1–5. This method is not applicable for emulsions.

#### LIQUID HOLDUP AND VAPOR SPACE DISENGAGEMENT

The dimensions of both vertical and horizontal separators are based on rules designed to provide adequate liquid holdup and vapor disengaging space. For instance, the desired vapor space in a vertical separator is at least 1½ times the diameter, with 6 in. as the minimum above the top of the inlet nozzle. In addition, a 6 in. minimum is required between the maximum liquid level and the bottom of the inlet nozzle. For a horizontal separator, the minimum vapor space is equal to 20% of the diameter of 12 in.

#### WIRE MESH PAD

Pads of fine wire mesh indicate coalescence of impinging droplets into larger ones, which then separate freely from the gas phase. No standard equations have been developed for the pressure drop across wire mesh because there are no standardized mesh pads. However, as a rule of thumb, the pressure drop of a wire mesh is  $\Delta P = 1.0 \text{ in. H}_2\text{O}$ . Every manufacturer makes a standard high

**EXAMPLE 6-7**

Size a horizontal separator for the following conditions

$$W_L = 56150 \text{ lb/h} \quad \rho_L = 60.0 \text{ lb/ft}^3$$

$$W_v = 40000 \text{ lb/h} \quad \rho_v = 1.47 \text{ lb/ft}^3$$

The liquid surge time is six minutes. Use  $L/D = 3.0$

*Solution*

Calculate the vapor–liquid separation factor by Eq. (6-51) and  $K_v$  by Eqs (6-52 and 6-53).

$$\text{S. Fac} = \frac{56150}{40000} \left( \frac{1.47}{60.0} \right)^{0.5} = 0.21972$$

From correlation, determine the design vapor velocity factor  $K_v$  and the maximum design vapor velocity [35].

From Eq. (6-52),

$$X = \ln(0.21972) = -1.51539$$

From Eq. (6-53),

$$K_v = \exp[-1.942936 + (-0.814894)(-1.51539)]$$

$$+ (-0.17939)(-1.51539)^2$$

$$+ (-0.012379)(-1.51539)^3$$

$$+ (0.000386235)(-1.51539)^4$$

$$+ (0.00025955)(-1.51539)^5]$$

$$K_v = [-1.942936 + 1.234882 - 0.411952 + 0.043078]$$

$$+ 0.002037 - 0.000207]$$

$$K_v = \exp[-1.075098] = 0.34126$$

From Eq. (6-65), for horizontal vessels,  $K_H$  is

$$K_H = 1.25(0.341) = 0.426$$

Calculate the maximum design vapor velocity.

From Eq. (6-66),

$$(U_v)_{\max} = 0.426 \left( \frac{60 - 1.47}{1.47} \right)^{0.5} = 2.69 \text{ ft/s}$$

efficiency, very high efficiency, or high throughput mesh under various trade names, each for a specific requirement.

## STANDARDS FOR HORIZONTAL SEPARATORS

The following specifications are generally standard in the design of horizontal separators [37].

1. The maximum liquid level shall provide a minimum vapor space height of 15 in., but should not be below the centerline of the separator.
2. The volume of dished heads is not considered in vessel sizing calculations.

Calculate the required vapor flow area using Eq. (6-67).

$$Q_v = \frac{W_v}{(3600)(\rho_v)} = \frac{40000}{(3600)(1.47)} = 7.56 \text{ ft}^3/\text{s}$$

$$(A_v)_{\min} = \frac{7.56}{2.69} = 2.81 \text{ ft}^2$$

When the vessel is full, the separator vapor area can be assumed to occupy only 15–25% of the total cross-sectional area. Here, a value of 20% is used and the total cross-sectional area is expressed as follows (using Eqs. 6-68 and 6-69).

$$(A_{\text{Total}})_{\min} = \frac{2.81}{0.2} = 14.05 \text{ ft}^2$$

$$D_{\min} = \left[ \frac{4 \times 14.05}{\pi} \right]^{0.5} = 4.23 \text{ ft}$$

Assume the length-to-diameter ratio of 3 (i.e.,  $L/D = 3$ ). Calculate the vessel length.

From Eq. (6-70),

$$L = (3)(4.23) = 12.7 \text{ ft}$$

Because the vapor area is assumed to occupy 20% of the total cross-sectional area, the liquid area will occupy 80% of the total cross-sectional area.

From Eq. (6-71),

$$A_L = (0.8)(14.05) = 11.24 \text{ ft}^2$$

Calculate the vessel volume from Eq. (6-72).

$$V_{\text{VES}} = (14.05)(12.7) = 178.435 \text{ ft}^3$$

Calculate liquid surge time using Eq. (6-73).

$$T = \frac{(60)(11.24)(12.7)(60)}{56150} = 9.15 \text{ min.}$$

The Excel spreadsheet Example 6-7.xls has been developed to size horizontal gas–liquid separator.

3. The inlet and outlet nozzles shall be located as closely as practical to the vessel tangent lines. Liquid outlets shall have anti-vortex baffles.

## PIPING REQUIREMENTS

Pipes that are connected to and from the process vessels must not interfere with the good working of the vessels. Therefore, the following guidelines should be observed:

- There should be no valves, pipe expansions, or contractions within 10 pipe diameters of the inlet nozzle.
- There should be no bends within 10 pipe diameters of the inlet nozzle except the following.

- For knockout drums and demisters, a bend in the feed pipe is permitted if this is in a vertical plane through the axis of the feed nozzle.
- For cyclones, a bend in the feed pipe is allowed if this is in a horizontal plane and the curvature is the same direction as the cyclone vortex.
- A pipe reducer may be used in the vapor line leading from the separator, but it should be no nearer to the top of the vessel than twice the outlet pipe diameter.
- A gate or ball type valve that is fully opened in normal operation should be used, where a valve in the feed line near the separator cannot be avoided.
- High pressure drops that cause flashing and atomization should be avoided in the feed pipe.
- If a pressure reducing valve in the feed pipe cannot be avoided, it should be located as far upstream of the vessel as practicable.

### CYCLONE SEPARATORS

Cyclones are widely used for the separation and recovery of industrial dusts from air or process gases. Cyclones are the principal type of gas-solid separators using centrifugal force. They are simple to construct, of low cost, and are made from a wide range of materials with an ability to operate at high temperatures and pressures. Cyclones are suitable for separating particles where agglomeration occurs. Pollution and emission regulations have compelled designers to study the efficiency of cyclones.

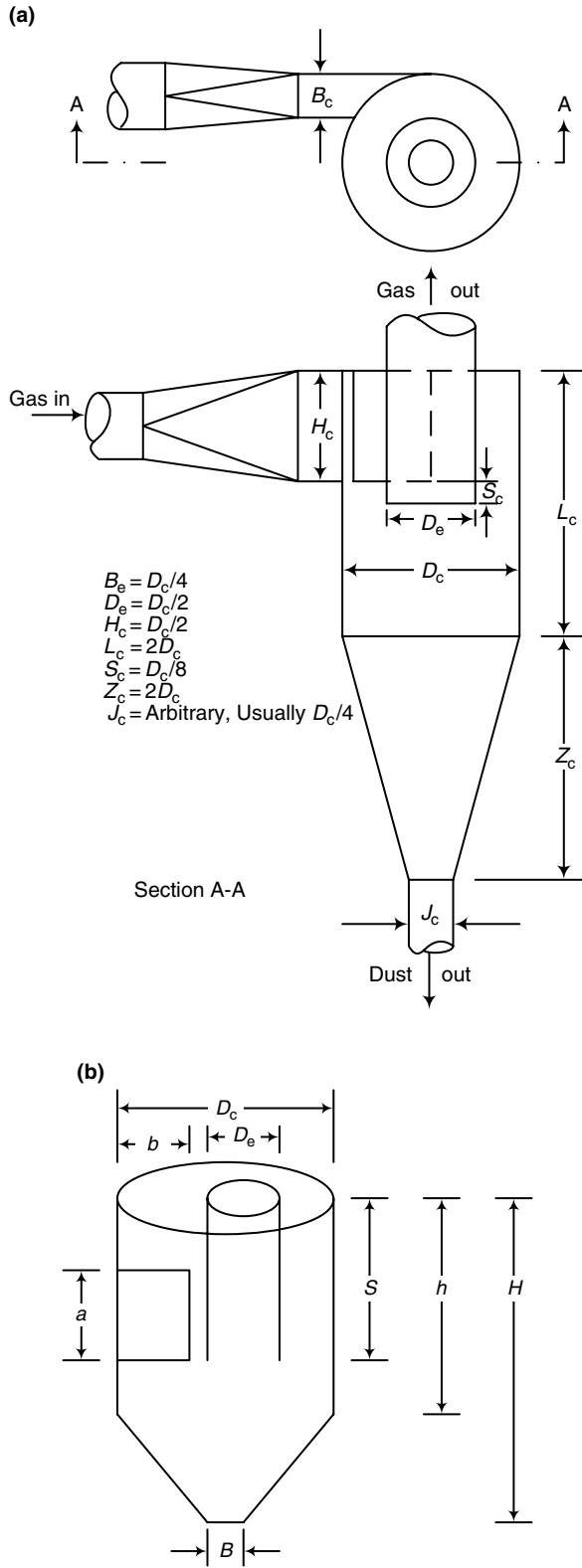
Cyclones offer the least expensive means of dust collection. They give low efficiency for collection of particles smaller than  $5\text{ }\mu\text{m}$ . A high efficiency of 98% can be achieved on dusts with particle sizes of  $0.1\text{--}0.2\text{ }\mu\text{m}$  that are highly flocculated.

Cyclone reactor types permit study of flow pattern and residence time distribution [38, 39]. For example, see the studies by Coker [40, 41] of synthetic detergent production with fast reaction. Reactor type cyclones are widely used in separating a cracking catalyst from vaporized reaction products.

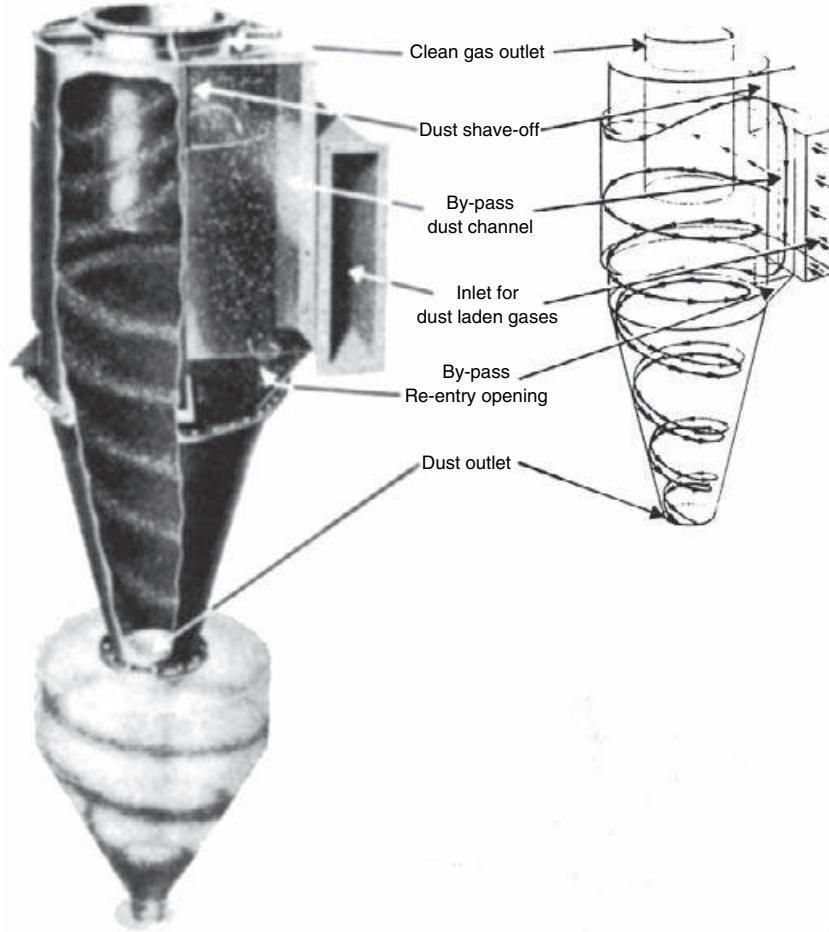
Reversed flow cyclones are the most common design, in which the dust-laden gas stream enters the top section of the cylindrical body either tangentially or via an involute entry. The cylindrical body induces a spinning vortexed flow pattern to the gas-dust mixture. Centrifugal force separates the dust from gas stream; the dust travels to the walls of the cylinder, and down the conical section to the dust outlet. The spinning gas also travels down the wall toward the apex of the cone, but reverses direction in an air core and leaves the cyclone through the gas outlet tube at the top. This consists of a cylindrical sleeve, the vortex finder, whose lower end extends below the level of the feed port. Separation depends on particle settling velocities, governed by size, density, and shape.

Figure 6-45 gives a good typical cyclone arrangement, but this is by no means the highest efficiency or best design. Some researchers [42–44] have provided good design and performance analysis. Stairmand [45] and Strauss [46] have given guidelines for designing cyclones. The effects of feed and cyclone parameters on the efficiency are complex, as many parameters are interdependent. Figures 6-46–6-49 show photographs of some commercial units and Table 6-14 gives the effects of important operating and design parameters on cyclone performance.

The zone of most efficient separation is in the conical region designated by dimension  $Z_c$  in Figure 6-45. The larger particles are thrown against the wall before the outlet is reached, the finer particles are thrown out in the inner vortex as the direction of motion is reversed. Here, the relative velocity difference between the particle and the carrier is the greatest for any point in the separator. Although the tangential velocity component predominates



**Figure 6-45** (a) Cyclone-separator proportions – dust systems. (By permission from Perry, J.H., *Chemical Engineers Handbook*, 3rd ed., McGraw-Hill Company, 1950.) (b) A cyclone: Design configurations.



**Figure 6-46** Van Tongeran dust shave-off design. (Courtesy of Buell Engineering Co.)

throughout the cyclone, the axial velocity prevails in the turbulent center. Van Dongen and ter Linden [47] measured pressure patterns in a typical cyclone and found the lowest total pressure at the extreme bottom point of the cone, even lower than at the gas exhaust. Their pressure profiles indicate considerable eddy or secondary gas movement in the unit near the vertical axis.

**Solid Particle Cyclone Design.** Following the general dimensional relations of the typical cyclone as shown in Figure 6-45, the following general guides apply. This cyclone is better suited to solid particles removal than liquid droplets. To avoid re-entrainment it is important to keep the separated material from entering the center vortex of the unit. Solid particles generally slide down the walls with sufficient vertical velocity to avoid re-entrainment.

- Select outlet diameter to give gas velocity out not exceeding 600 ft/min. Bear in mind that higher velocity can be used in special designs.
- Due to the usual conditions of limiting pressure drop, entrance velocities range from 1000 to 4000 ft/min. A velocity of 3000 ft/min is good average, although velocities up to 6000 ft/min are used in some applications.
- Select cylindrical shell diameter  $D_c$  with two considerations in mind:

- Large diameter reduces pressure drop
- Small diameter has higher collection efficiency for the same entrance conditions and pressure drop.

- The length of the inverted cone section,  $Z_c$ , is critical, although there is no uniformity in actual practice. The dimensions suggested in Figure 6-45 are average.

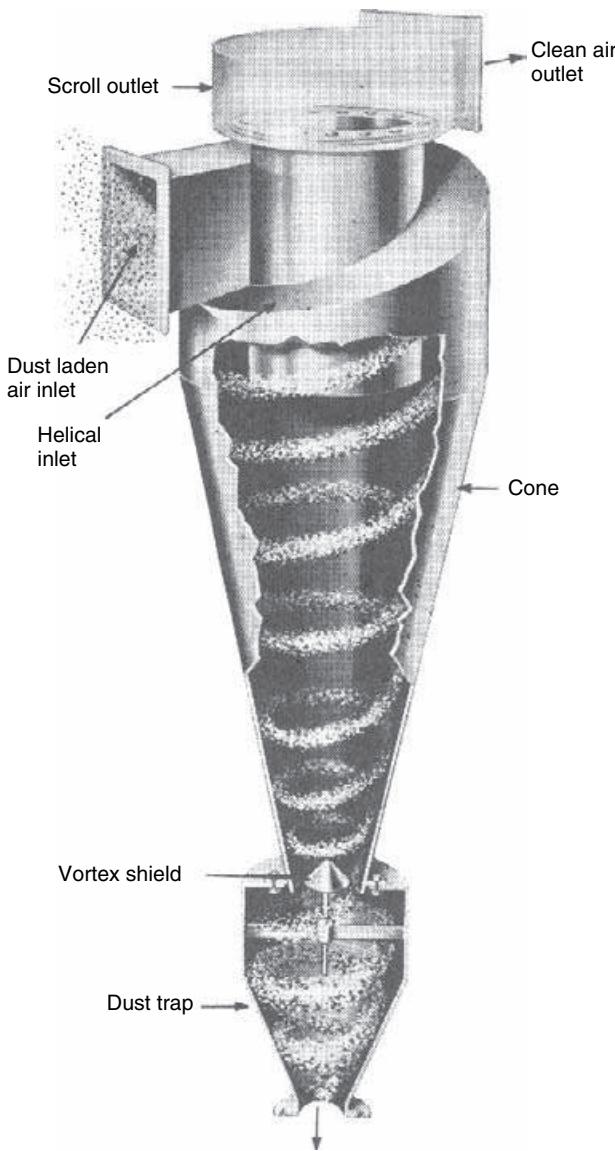
#### CYCLONE DESIGN PROCEDURE

The computation of cyclone fractional grade efficiency depends on cyclone parameters and flow characteristics of particle-laden gases. The procedure involves a series of equations containing exponential and logarithmic functions. Koch and Licht [42] describe a cyclone using seven geometric ratios in terms of its diameter as

$$\frac{a}{D_c}, \frac{b}{D_c}, \frac{D_e}{D_c}, \frac{S}{D_c}, \frac{h}{D_c}, \frac{H}{D_c}, \frac{B}{D_c}$$

They further stated that certain constraints are observed in achieving a sound design. They are as follows.

1.  $a < S$  to prevent short-circuiting
2.  $b < \frac{1}{2}(D_c - D_e)$  to avoid sudden contraction



**Figure 6-47** Helical entry cyclone. (Courtesy of The Ducon Co. Inc.)

3.  $S + 1 \leq H$  to keep the vortex inside the cyclone
4.  $S < h$
5.  $h < H$
6.  $\Delta P < 10 \text{ in. H}_2\text{O}$
7.  $v_i/v_s \leq 1.35$  to prevent re-entrainment  
where

$v_i$  = inlet linear velocity, ft/s

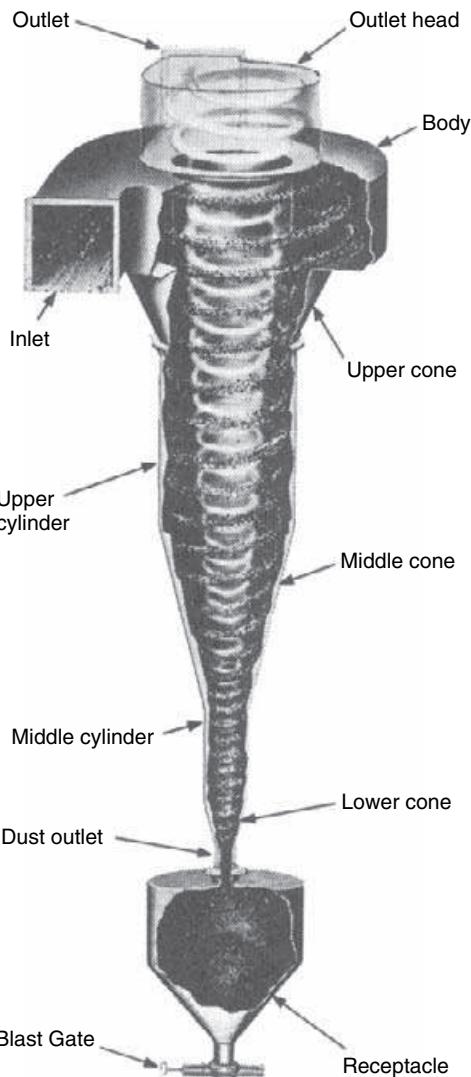
$v_s$  = Saltation velocity, ft/s.

8.  $v_i/v_s \simeq 1.25$  for optimum efficiency.

The Equations (Figure 6-45b)

Natural length  $l$  is the distance below the gas outlet where the vortex turns.

$$l = 2.3D_e \left( \frac{D_c^2}{ab} \right)^{1/3} \quad (6-80)$$



**Figure 6-48** Involute entry cyclone. (By permission from American Blower Div. American Radiator and Standard Sanitary Corp.)

For  $l < (H - S)$ , the cyclone volume at the natural length (excluding the core) is  $V_{nl}$

$$V_{nl} = \frac{\pi D_e^2}{4} (h - S) + \left( \frac{\pi D_c^2}{4} \right) \left( \frac{1+S-h}{3} \right) \left( 1 + \frac{d}{D_c} + \frac{d^2}{D_c^2} \right) - \left( \frac{\pi D_c^2 l}{4} \right) \quad (6-81)$$

The diameter of a central core at a point of vortex turns is

$$d = D_c - (D_c - B) \left( \frac{S+l-h}{H-h} \right) \quad (6-82)$$

For  $l > (H - S)$ , the cyclone volume below the exit duct (excluding the core) is  $V_H$

$$V_H = \frac{\pi D_e^2}{4} (h - S) + \left( \frac{\pi D_c^2}{4} \right) \left( \frac{H-h}{3} \right) \left( 1 + \frac{B}{D_c} + \frac{B^2}{D_c^2} \right) - \frac{\pi D_c^2}{4} (H - S) \quad (6-83)$$

**TABLE 6-14 Effects of Variables on Cyclone Performance**

Variable	Effect
Pressure drop increases	Cut size (diameter on particles of which 50% are collected) decreases; flow rate increases; sharpness increases.
Solids content of feed increases	Cut size increases (large effect above 15–20% v/v).
$(\rho_p - \rho_f)$ increases	Cut size decreases.
Viscosity of liquid phase increases	Little effect below 10 mPas.
Cyclone diameter ( $D_c$ ) increases	Cut size increases; pressure drop usually decreases.
Cyclone inlet (a) diameter increases	Gravitational force in cyclone decreases; cut size increases; capacity falls; pressure drop decreases.
Overflow diameter increases	Cut size increases; risk of coarse size appearing.
Underflow diameter increases	Bring excess fines from liquid phase into underflow.
Cyclone shape becomes longer	Decreases cut size; sharpens separation.

For a vortex exponent  $n$ :

$$n = 1 - \left[ 1 - \frac{(12D_c)^{0.14}}{2.5} \right] \left[ \frac{T + 460}{530} \right]^{0.3} \quad (6-84)$$

The cyclone volume constant  $K_c$  using  $V_{nl}$  or  $V_H$ :

$$K_c = \frac{(2V_s + V_{nl,H})}{2D_c^3} \quad (6-85)$$

where

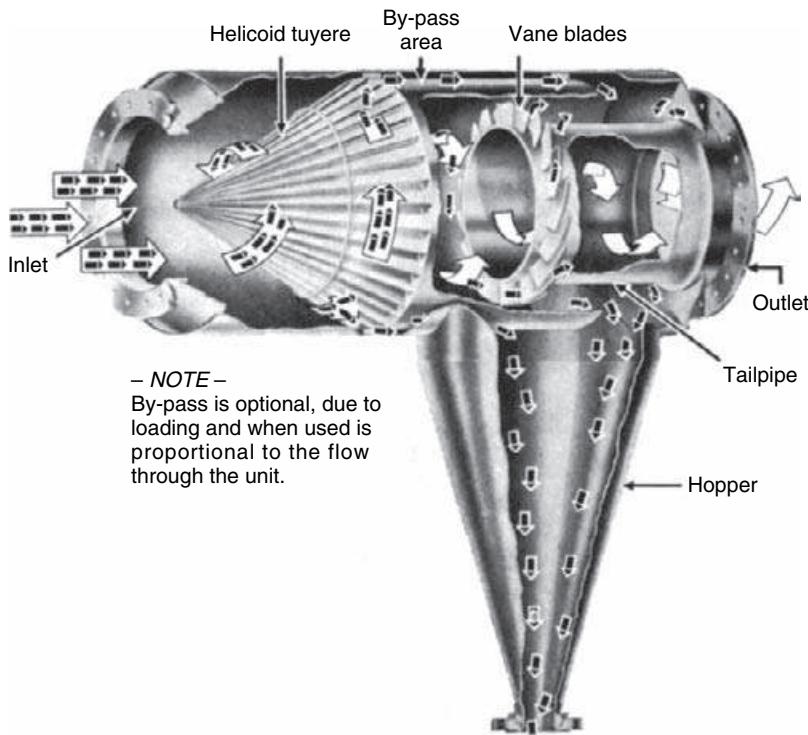
$$V_s = \left[ \frac{\pi (S - a/2) (D_c^2 - D_e^2)}{4} \right] \quad (6-86)$$

The relaxation time  $\tau_i$  for particle species  $i$  of diameter  $d_{pi}$  is

$$\tau_i = \frac{\rho_p (d_{pi})^2}{18 \mu}, \text{ s} \quad (6-87)$$

Cyclone configuration factor  $G$  is specified by the geometric ratios that describe the cyclone's shape. The cyclone configuration factor  $G$  is a function only of the configuration and is specified by the seven geometrical ratios that describe its shape.  $G$  is expressed as

$$G = \frac{8K_c}{K_a^2 K_b^2} \quad (6-88)$$



**Figure 6-49** Stationary vane centrifugal separator. (Courtesy of Centrifix Corp.)

where

$$K_a = \frac{a}{D_c}; K_b = \frac{b}{D_c} \quad (6-89)$$

Substituting the values of  $K_a$ ,  $K_b$ , and  $K_c$  in Eq. (6-88) gives

$$G = \left\{ 2[\pi(S - a/2)(D_c^2 - D_e^2)] + 4V_{H,nl} \right\} \frac{D_c}{a^2 b^2} \quad (6-90)$$

The fractional or grade efficiency  $\eta_i$  can be expressed as

$$\eta_i = 1 - \exp \left\{ -2 \left[ \frac{G \tau_i Q}{D_c^3} (n+1) \right]^{0.5/(n+l)} \right\} \quad (6-91)$$

where

$a$	= inlet height, ft
$b$	= inlet width, ft
$B$	= cyclone dust outlet diameter, ft
$d$	= diameter of central core at a point where vortex turns, ft
$d_{pi}$	= diameter of particle in size range $i$ , ft
$D_e$	= cyclone gas exit duct diameter, ft
$D_c$	= cyclone diameter, ft
$(D_p)_{crit}$	= critical particle diameter, ft
$g$	= acceleration due to gravity, 32.2 ft/s <sup>2</sup>
$G$	= cyclone configuration factor
$h$	= cylindrical configuration factor
$H$	= height of a segment of a circle, ft
$i$	= subscript denotes interval in particle size range
$K_a$	= $a/D_c$
$K_b$	= $b/D_c$
$K_c$	= cyclone volume constant
$l$	= natural length (distance below gas outlet where vortex turns), ft
$n$	= vortex component
$Q$	= total gas flow rate, actual ft <sup>3</sup> /s
$S$	= gas outlet length, ft
$T$	= temperature, °F
$V_H$	= volume below exit dust (excluding core), ft <sup>3</sup>
$V_{nl}$	= volume at natural length (excluding core), ft <sup>3</sup>
$V_s$	= annular volume above exit duct to middle of entrance duct, ft <sup>3</sup>
$\eta_i$	= grade efficiency for particle size at midpoint of interval, $i$ %
$\tau$	= relaxation time, s.

**Saltation Velocity.** Koch and Licht [42] expressed the saltation velocity as the minimum fluid velocity necessary to prevent the settling out of solid particles carried in the stream and as the necessary velocity that picks up deposited particles and transports them without settling.

Zenz [48] has shown that the velocity given by the latter differs from the former by a factor of 2–2.5. Kalen and Zenz [49] have applied the saltation concept to cyclone design by assuming the following.

1. There is no slippage between fluid and particles. The cyclone inlet width is the effective pipe diameter for calculating saltation effects.
2. Grain loading (dust concentration) is less than 10 gr/ft<sup>3</sup>.
3. The diameter effect on the saltation velocity is proportional to the 0.4 power of the inlet width.

The saltation velocity  $v_s$  is dependent on cyclone dimensions, as well as particle and fluid properties;  $v_s$  is expressed as

$$v_s = 2.055\omega \left[ \frac{b/D_c}{(1-b/D_c)^{1/3}} \right] D_c^{0.067} v_i^{2/3}, \text{ ft/s} \quad (6-92)$$

where

$$\omega = \left[ \frac{4g\mu(\rho_p - \rho_f)}{3\rho_f^2} \right]^{1/3} \quad (6-93)$$

Inlet velocity,  $v_i$

$$v_i = \frac{Q}{(ab)}, \text{ ft/s} \quad (6-94)$$

Kalen and Zenz have shown that maximum cyclone collection efficiency occurs at  $v_i/v_s = 1.25$ , and Zenz has found experimentally that fluid re-entrainment occurs at  $v_i/v_s = 1.36$

**Pressure Drop ( $\Delta P$ ).** Several attempts have been made to calculate the frictional loss or  $\Delta P$  of a cyclone, although none has been very satisfactory. Assumptions made have not considered entrance compression, wall friction, and exit contraction, all of which have a major effect. Consequently, no general correlation of cyclone  $\Delta P$  has been adopted. Pressure drop in a cyclone with collection efficiency is important in evaluating its cost. Correlations for the pressure drop have empirical values, and are acceptable up to  $\Delta P = 10$  in. H<sub>2</sub>O. The pressure drop  $\Delta P$ , or the frictional loss is expressed in terms of the velocity head based on the cyclone inlet area. The frictional loss through cyclones is from 1 to 20 inlet velocity heads, and depends on the geometric ratios;  $\Delta P$  through a cyclone is given by

$$\Delta P = 0.003\rho_f v_i^2 N_H \quad (6-95)$$

where

$$N_H = K \left( \frac{ab}{D_e^2} \right) \quad (6-96)$$

and

$N_H$  = number of inlet velocity heads

$\Delta P$  = pressure drop, in. H<sub>2</sub>O

$v_i$  = inlet velocity, ft/s

$\rho_f$  = fluid density, lb/ft<sup>3</sup>.

$K = 16$  for no inlet vane;  $K = 7.5$  with a neutral inlet vane.  $\Delta P$  depends strongly on the inlet velocity, and high velocities can cause both re-entrainment and high pressure drop. However, entrainment can be reduced to a minimum, if the cyclone has a small base angle.

**Critical Particle Diameter.** Many theories have been proposed to predict the performance of a cyclone, although no fundamental relationship has been accepted. Capacity and efficiency depend on the inlet velocity and the dimensions of the vessel. Attempts have been made to predict the critical particle diameter  $(D_p)_{crit}$ . This is the size of the smallest particle that is theoretically separated from the gas stream with 50% efficiency. The critical particle diameter is defined by Walas [50] as

$$(D_p)_{crit} = \left[ \frac{9\mu D_c}{4\pi N_t v_i (\rho_p - \rho_g)} \right]^{0.5} \quad (6-97)$$

where

$$N_t = (v_i) [0.1079 - 0.00077v_i + 1.924(10^{-6})v_i^2] \quad (6-98)$$

and

$D_c$  = cyclone diameter, ft  
 $v_i$  = inlet linear velocity, ft/s  
 $\mu$  = fluid viscosity, lb/ft/s  
 $\rho_f$  = fluid density, lb/ft<sup>3</sup>  
 $\rho_p$  = particle density, lb/ft<sup>3</sup>  
 $N_t$  = effective number of turns made by the gas stream in the cyclone.

$N_t$  has been found to be about 5 turns of the gas stream in the unit, and is considered somewhat conservative. When re-entrainment takes place,  $N_t$  may drop to 1.0 or 2.0. The API study presents an excellent survey of cyclone dust collectors [51].

With a height of opening equal to 2.5 times the width, the volumetric rate is

$$Q = Av_i = 2.5D_c v_i / 16 \quad (6-99)$$

To obtain a high efficiency, the vessel diameter must be small, but in order to accommodate a required volumetric rate, many units in parallel may be needed. These units, called "multi-cyclones," may be incorporated in a single shell at a cost that may be justifiable in view of greater efficiency and lower pressure drop. Capacity and efficiency depend on the inlet velocity and the dimensions of the vessel. An important parameter is the critical particle diameter which is one that is removed to the extent of 50%. Figure 6-50 shows the percentage removal of particles in a cyclone as a function of the ratio of the particle diameter to the critical diameter.

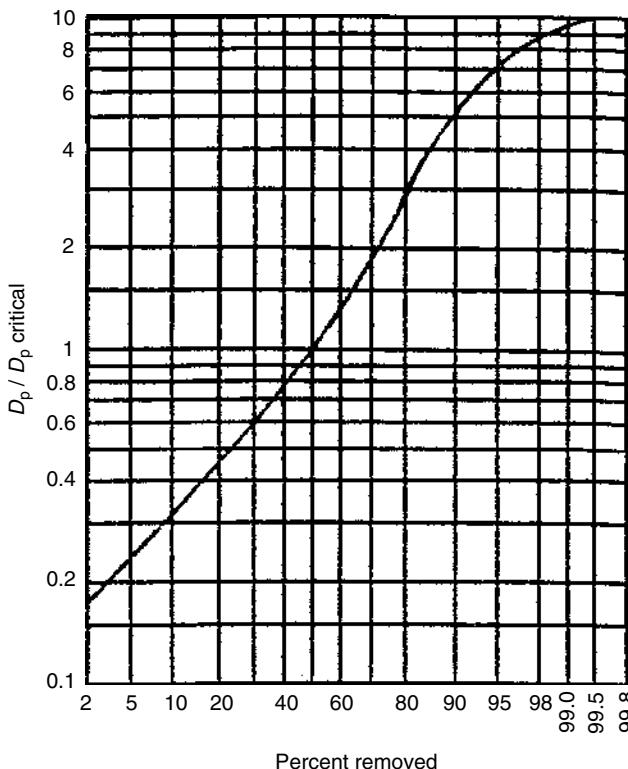


Figure 6-50 Percentage removal of particles as a function of particle diameter relative to the critical diameter.

**Cyclone Design Factors.** Cyclones are designed to meet specified  $\Delta P$  limitations. The factor that controls the collection efficiency is the cyclone diameter, and a smaller diameter cyclone at a fixed  $\Delta P$  will have a higher efficiency. Therefore, small diameter cyclones require multiple units in parallel for a given capacity. Reducing the gas outlet diameter results in an increased collection efficiency and  $\Delta P$ . High efficiency cyclones have cone lengths in the range of 1.6–3.0 times the cyclone diameters.

Collection efficiency increases as the gas throughput increases. Kalen and Zenz [49] reported that collection efficiency increases with increasing gas inlet velocity to a minimum tangential velocity. This reaches the point where the dust is re-entrained or not deposited because of saltation. Koch and Licht [43] showed that saltation velocity is consistent with cyclone inlet velocities in the range of 50–90 ft/s.

**Troubleshooting Cyclone Maloperations.** In general, cyclones are used to separate particles from the gas stream, but recent developments have enabled cyclones to function as reactors. Some cyclone reactors can separate cracking catalyst from vaporized reaction products in the range of 950°F and 1000°F, or can function as regenerators for flue gases between 1250°F and 1500°F. In both cases, the high particle velocities can cause rapid erosion of the cyclone material. This often results in poor performance of the cyclone. Other causes of poor cyclone performance are

- a hole in a cyclone body
- a cyclone volute plugged
- a dip leg unsealed
- a dip leg plugged and
- dip leg failure.

Lieberman [52] has reviewed the causes of these maloperations, which often result in loss of catalyst and reduced efficiency. A deficient cyclone reactor is identified by bottom sediment and water levels in the slurry oil products. For a regenerator cyclone, problems are visibly identified by the increased opacity of the regenerator flue gas or by reduced rates of spent catalyst withdrawal.

**Cyclone Collection Efficiency.** Typical estimating efficiencies are given in Figures 6-51 and 6-52. Note the curves indicate how much dust of each particle size will be collected. The efficiency increases as the pressure drop increases; that is, a smaller separator might have a higher efficiency due to the higher gas velocities and increased resistance than a larger unit for the same gas flow. For example, there are several curves of the typical shape of Figure 6-51, with each curve for a definite resistance to flow through the unit.

The pressure drop in a typical cyclone is usually between 0.5 and 8 in. of water. It can be larger, but rarely exceeds 10 in. water for single units. The API study [51] summarizes the various factors. Lapple [5, 53] gives calculation equations, but in general the most reliable pressure drop information is obtained from the manufacturer. Here is how the pressure drop may be estimated.

For the typical cyclone of Figure 6-45 [5],

(a) Inlet velocity head based on inlet area:

$$h_{vi} = 0.003\rho V_c^2 \quad (6-100)$$

**EXAMPLE 6-8**

Determine the efficiency and pressure drop ( $\Delta P$ ) based on cyclone dimensions and the gas flow rate at 516.7 ft<sup>3</sup>/s and density of 0.075 lb/ft<sup>3</sup> containing particles with density of 62.43 lb/ft<sup>3</sup>. The dimensions and data required are shown in the following table.

Data	Dimension
Cyclone gas inlet height, ft	4.5
Cyclone gas inlet width, ft	1.896
Cyclone gas outlet length, ft	3.448
Cyclone gas outlet diameter, ft	3.792
Cyclone cylindrical height, ft	8.552
Cyclone diameter, ft	6.333
Gas rate, ft <sup>3</sup> /s	516.7
Gas density, lb/ft <sup>3</sup>	0.075
Particle density, lb/ft <sup>3</sup>	62.43
Gas viscosity, lb/ft s	$1.28 \times 10^{-5}$
Temperature, °F	110
Particle size, ft	$3.281 \times 10^{-5}$

*Solution*

Natural length  $l$  is the distance below the gas outlet where the vortex turns.

From Eq. (6-80),

$$l = 2.3 (3.792) \left( \frac{6.33^2}{4.5 \times 1.896} \right)^{1/3} = 14.597 \text{ ft}$$

For  $l < (H - S)$ , the cyclone volume at the natural length (excluding the core) is  $V_{nl}$ :

Check if  $l < (H - S)$

$$14.597 < (26.333 - 3.448)$$

$$14.597 < 22.885$$

Cylindrical volume at the natural length  $V_{nl}$  is calculated using Eq. (6-81).

$$\begin{aligned} V_{nl} &= \frac{\pi (6.333)^2}{4} (8.552 - 3.448) + \left( \frac{\pi 6.333^2}{4} \right) \\ &\quad \times \left( \frac{14.597 + 3.448 - 8.552}{3} \right) \left( 1 + \frac{4.304}{6.333} + \frac{4.304^2}{6.333^2} \right) \\ &\quad - \left[ \frac{\pi (3.792)^2 (14.597)}{4} \right] \\ &= 209.37 \text{ ft}^3 \end{aligned}$$

The diameter  $d$  of a central core at a point of turn is calculated from Eq. (6-82).

$$d = 6.333 - (6.333 - 2.533) \left[ \frac{(3.448 + 14.597 - 8.552)}{(26.333 - 8.552)} \right] = 4.304 \text{ ft}$$

From Eq. (6-84), a vortex exponent  $n$  is

$$n = 1 - \left[ 1 - \frac{(12 \times 6.333)^{0.14}}{2.5} \right] \left[ \frac{(110 + 460)}{530} \right]^{0.3} = 0.767$$

From Eq. (6-86),

$$V_s = \left[ \frac{\pi (3.448 - 4.5/2) (6.333^2 - 3.792^2)}{4} \right] = 24.207 \text{ ft}^3$$

Then, from Eq. (6-85),

$$K_c = \frac{(2 \times 24.207 + 209.37)}{(2 \times 6.333^3)} = 0.507$$

The relaxation time  $\tau_i$  for particle species  $i$  of diameter  $d_{pi}$  (from Eq. (6-87)) is

$$\tau_i = \frac{(62.43) (0.00003281)}{(18) (0.0000128)} = 0.0002916 \text{ s}$$

Cyclone configuration factor  $G$  (from Eq. (6-90)) is

$$\begin{aligned} G &= \{2 [\pi (3.448 - 4.5/2) (6.333^2 - 3.792^2)] \\ &\quad + 4 (209.37)\} \frac{6.333}{(4.5^2) (1.896^2)} = 89.71 \end{aligned}$$

Inlet velocity  $v_i$ , ft/s (from Eq. (6-94)) is

$$v_i = \frac{516.7}{(4.5) (1.896)} = 60.56 \text{ ft/s.}$$

Saltation velocity  $v_s$  (from Eq. (6-93)) is

$$v_s = 2.055 \omega \left[ \frac{b/D_c}{(1 - b/D_c)^{1/3}} \right] D_c^{0.067} v_i^{2/3} \text{ ft/s}$$

where

$$\omega = \left\{ \frac{4g\mu(\rho_p - \rho_f)}{3\rho_f^2} \right\}^{1/3}$$

$$\omega = \left[ \frac{4 \times 32.2 \times 0.0000128 (62.43 - 0.075)}{(3) (0.075)^2} \right]^{1/3} = 1.826$$

Then

$$\begin{aligned} v_s &= (2.055) (1.826) \left[ \frac{1.896/6.333}{(1 - 1.896/6.333)^{1/3}} \right] (6.333)^{0.067} \\ &= 22.0775 \text{ ft/s} \end{aligned}$$

The critical particle diameter (from Eq. (6-97)) is

$$(D_p)_{crit} = \left[ \frac{9 \mu D_c}{4 \pi N_t v_i (\rho_p - \rho_g)} \right]^{0.5} \text{ ft}$$

(continued)

**EXAMPLE 6-8—(continued)**

where the effective number of turns  $N_t$  (from Eq. (6-98)) is

$$\begin{aligned} N_t &= (v_i) [0.1079 - 0.00077 v_i + 1.924 (10^{-6}) v_i^2] \\ &= 60.56 [0.1079 - 0.00077 (60.56)] \\ &\quad + 1.924 (10^{-6}) (60.56)^2 = 4.137 \end{aligned}$$

Then

$$\begin{aligned} (D_p)_{\text{crit}} &= \left[ \frac{9(0.0000128)(6.333)}{4(\pi)(4.137)(60.56)(62.43 - 0.075)} \right]^{0.5} \\ &= 6.096 \times 10^{-5} \text{ ft (19 } \mu\text{m)} \end{aligned}$$

The ratio of particle diameter to critical particle diameter is

$$\frac{D_p}{(D_p)_{\text{crit}}} = \frac{0.00003281}{0.00006096} = 0.538$$

From Figure 6-50, 28% of the other sized particles are removed. The cyclone pressure drop  $\Delta P$  (from (6-95)) is

$$\Delta P = 0.003 \rho_f v_i^2 N_H \text{ in. H}_2\text{O}$$

where (according to Eq. (6-96))

$$N_H = K \left( \frac{ab}{D_e^2} \right)$$

$K = 16$  for no inlet vane;  $K = 7.5$  with a neutral inlet vane.

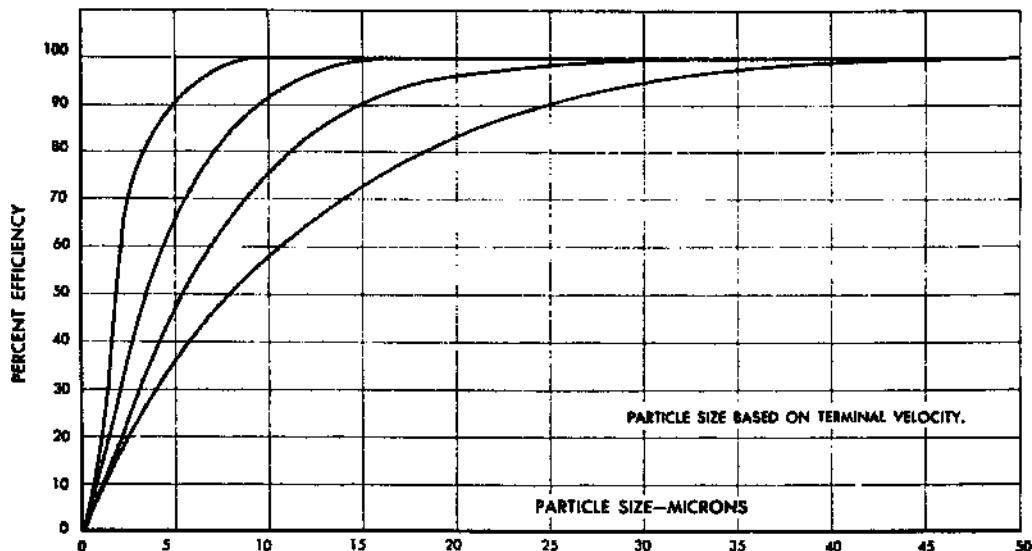
$$N_H = 16 \left( \frac{(4.5)(1.896)}{(3.792)^2} \right) = 9.4936$$

$$\Delta P = 0.003 (0.075) (60.56)^2 (9.4936) = 7.83 \text{ in. H}_2\text{O}$$

The fractional or grade efficiency  $\eta_i$  (from Eq. (6-91)) is

$$\eta_i = 1 - \exp \left\{ -2 \left[ \frac{(89.71)(0.0002916)(516.7)}{(6.333)^3} \right. \right. \\ \left. \left. (1 + 0.767) \right]^{0.5/(1.767)} \right\} = 0.641$$

The Excel spreadsheet Example 6-8.xls has been developed for Example 6-8.



**Figure 6-51** General efficiency curves, applied specifically to helical entry cyclone dust separators. (Courtesy of the Ducon Co.)

**(b)** Internal cyclone friction loss:

$$F_{cv} = \Delta P_{cv} + 1 - \left( \frac{4A_c}{\pi D_e^2} \right)^2 \quad (6-101)$$

where  $A_c$  = cyclone inlet area.

$$\Delta P_{cv} = k \left( \frac{D_c}{D_e} \right)^2 \quad (6-102)$$

where

$D_c$  = cyclone diameter, ft

$D_e$  = cyclone gas exit duct diameter, ft.

$K$  has been found to be constant at 3.2 for cyclones with an involute entrance

$$B_e/D_c = 1/8 - 3/8$$

$$H_c/D_c = 1.0$$

$$D_e/D_c = 1/4 - 3/4$$

For the typical cyclone of Figure 6-45 with tangential inlet:

$$F_{cv} = \frac{KB_e H_c}{D_e^2} \quad (6-103)$$

where  $B_e H_c$  = cyclone inlet area,  $\text{ft}^2$ .

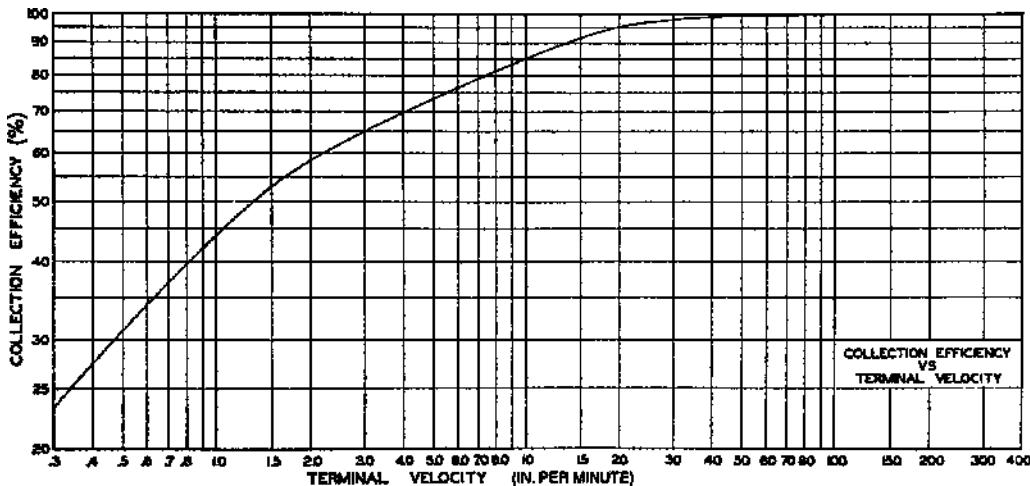


Figure 6-52 General efficiency curve applies specifically to involute entry cyclone dust separators. (Courtesy of American Blower Div., American Radiator and Standard Sanitary Corp.)

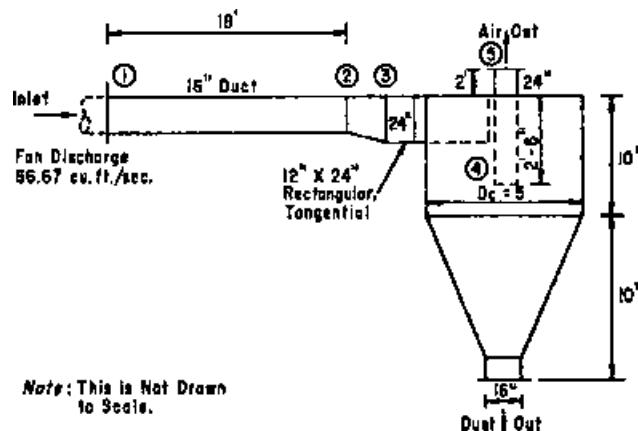


Figure 6-53 Pressure drop for cyclone separator system. (Adapted by permission from Lapple, C.E., *Fluid and Particle Dynamics*, 1st ed., Univ. of Delaware, 1954.)

$$K = 16$$

$$F_{cv} = 8.0$$

If inlet vane is formed with inlet connection,  $K = 7.5$

#### LIQUID CYCLONE-TYPE SEPARATOR

The unit shown in Figure 6-54 has been used in many process applications with a variety of modifications[47, 54, 55]. It is effective in liquid entrainment separation, but is not recommended for solid particles due to the arrangement of the bottom and outlet. The flat bottom plate serves as a protection to the developing liquid surface below. This prevents re-entrainment. In place of the plate a vortex breaker type using vertical cross plates of 4–12 in. depth also is used (also see [56]). The inlet gas connection is placed above the outlet dip pipe by maintaining dimension of only a few inches at point 4. In this type of unit some liquid will creep up the walls as the inlet velocity increases.

In order to handle higher loads, the liquid baffle is placed at the top to collect liquid and cause it to drop back down through the gas body. If the baffle is omitted, the liquid will run down the outlet pipe and be swept into the outlet nozzle by the outgoing

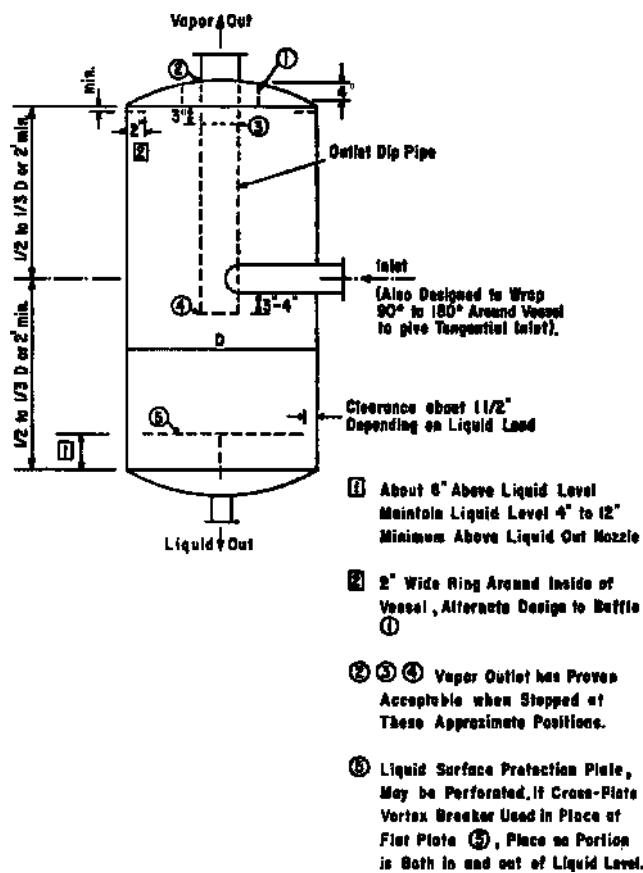


Figure 6-54 Centrifugal liquid separator.

gas as shown in Figure 6-55b. Figures 6-55 and 6-56 show several alternate inlet and outlet details. The unit with a tangential entry is 30–60% more efficient than one with only a turned-down 90° elbow in the center.

If the design of Figure 6-45 is used for liquid–vapor separation at moderately high liquid loads, the liquid sliding down the walls in sheets and ripples has somewhat of a tendency to be torn off

**EXAMPLE 6-9****Cyclone System Pressure Drop**

A cyclone system is to be installed as a part of a bagging operation. The unit is shown in Figure 6-53. Determine the head required for purchase of the fan. The conditions are as follows:

Air volume: 4000 ft<sup>3</sup> per min of air at 70°F

Air density: 0.075 lb/ft<sup>3</sup>

Areas	(ft <sup>2</sup> )	Gas Velocity (ft/s)	Velocity Head* (in. H <sub>2</sub> O)
Inlet duct	1.398	47.7	0.50
Cyclone inlet	2.0	33.33	0.25
Cyclone exit duct	3.14	21.2	0.10

\* Velocity head, in. H<sub>2</sub>O =  $V_D^2/(16)(106)$ ,  $V_D$  = ft/min

Friction Loss ① to ②:

$$N_{Re} = 398,000$$

$$f = 0.0038$$

$$\text{No. Vel. hd} = \frac{4fL}{D} = \frac{4(0.0038)(18)}{(16/12)} = 0.204 \text{ Vel. hd}$$

$$\text{Friction loss} = (0.204)(0.5) = 0.102 \text{ in. H}_2\text{O}$$

Friction Loss ② to ③:

Assume as 1 Vel. hd (conservative).

$$\text{Friction loss} = (1)(0.50) = 0.50 \text{ in. H}_2\text{O}$$

Friction Loss ③ to ④:

$$F_{cv} = \frac{KW_i H_c}{D_e^2} = \frac{(16)(1)(2)}{(2)^2} = 8.0 \text{ Vel. hd}$$

Where  $W_i$  = width of rectangular cone inlet duct, ft.

$$\text{Friction loss} = 8.0(0.25) = 2.0 \text{ in. H}_2\text{O}$$

Friction Loss ④ to ⑤:

$$N_{Re} = 280,000$$

$$f = 0.004$$

$$\text{No. Vel. hd} = \frac{4fL}{D} = \frac{4(0.004)(4.5)}{(2)} = 0.036 \text{ Vel. hd}$$

$$\text{Friction loss} = 0.036(0.1) = 0.0036 \text{ in. H}_2\text{O}$$

Since the unit exhausts to atmosphere with no additional restrictions, the total pressure drop is

$$\Delta P_{(\text{total})} = \text{Friction loss} + \text{downstream Vel. Hd at } ⑤$$

— upstream Vel. Hd

$$\Delta P_{(\text{total})} = (0.102 + 0.50 + 2.0 + 0.0036) + 0^* - 0.50$$

$$\Delta P_{(\text{total})} = 2.6056 + 0 - 0.50 = 2.10 \text{ in. H}_2\text{O}$$

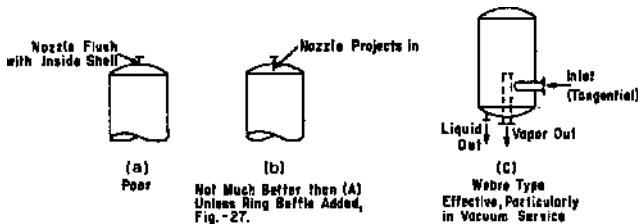


Figure 6-55 Separator outlets for liquid–vapor service.

from the rotating liquid and become re-entrained in the upward gas movement.

**Liquid Cyclone Design (Based on Air–water at Atmospheric Pressure).** For maximum liquid in outlet vapor of 4% wt based on incoming liquid to separator (see Figure 6-54),

- Select inlet pipe size to give vapor velocity at inlet of 100–400 ft/s for tangential pipe inlet.
- Select separator diameter to give velocity of 0.02–0.2 (max) times the inlet velocity. At 400 ft/s pipe velocity the separator velocity should be 0.018–0.03 times the pipe velocity. At 130 ft/s pipe velocity the separator velocity should be 0.15–0.2 times the pipe velocity.
- Establish dimensions from typical unit of Figure 6-54. Always evaluate the expected performance in terms of the final design, adjusting vertical dimensions to avoid gas whipping on liquid films or droplets. Do not direct inlet gas toward an outlet. Place manway on same side of vessel as tangential inlet.
- Pressure drop is essentially negligible for the average conditions of use. Some estimate of entrance and exit losses can be made by fluid flow techniques, and an internal loss of 0.25–2.0 psi assumed, depending upon system pressure and general unit dimensions.

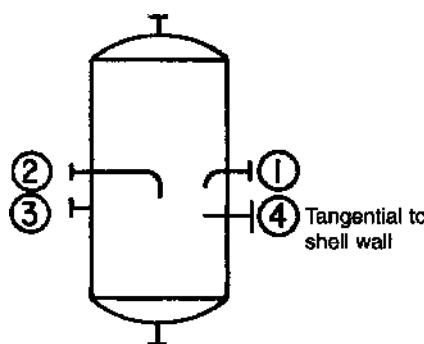
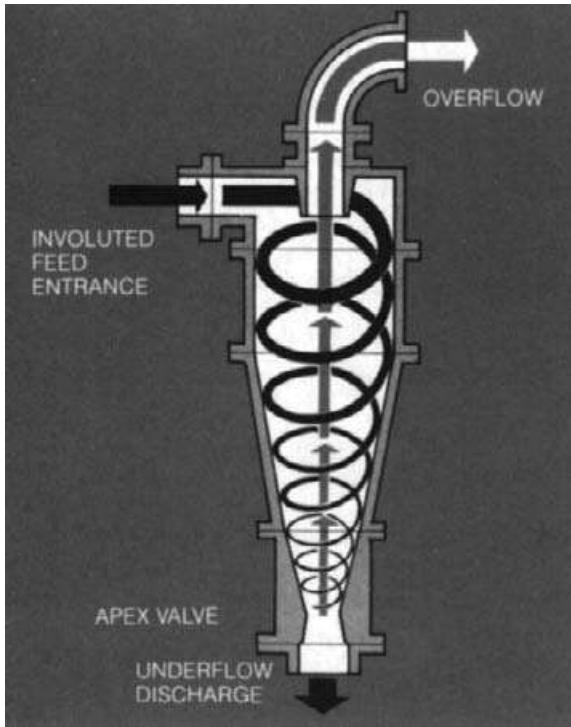


Figure 6-56 Separator inlets for liquid–vapor service.

\*Note that point ⑤ is at atmospheric pressure and the velocity head is zero; however, if there had been a back pressure or resistance at this point before discharging, it would have to be added in.



**Figure 6-57a** Liquid–solids removal cyclones. Feed enters tangentially along sidewall. (By permission from Krebs Engineers.)

e. For liquids and vapors other than air–water the separator velocity

$$V_{(\text{separator})} = 0.1885 V_{\text{sa}} \left( \frac{\rho_L - \rho_v}{\rho_v} \right)^{0.25} \quad (6-104)$$

where  $V_{\text{sa}}$  is the selected separator velocity when using an air–water system, ft/s.

The Webre design as tested by Pollak and Work [57] showed (Figure 6-55c) that internal action in the separator was responsible for some of the entrainment, particularly liquid creep up the vessel walls.

The performance of this unit correlated for several different types of particle distribution by [57]

$$\log[L_v(V'/L_1)^a] = b + cV' \quad (6-105)$$

where

$a = 2$  for Webre unit

$L_v$  = entrainment, lb liquid/min/ft<sup>2</sup> of inlet

$V'$  = vapor velocity entering, lb/min/ft<sup>2</sup> of inlet

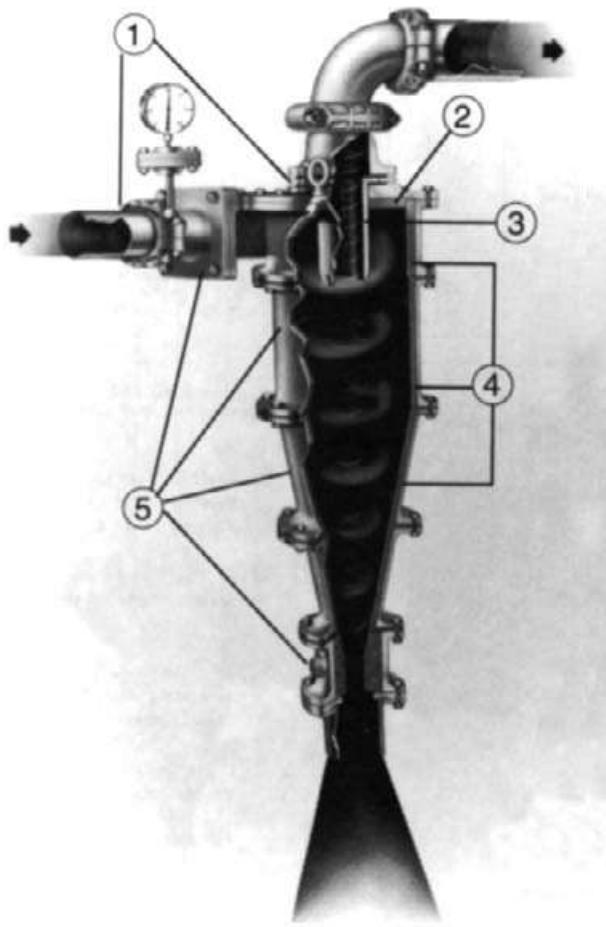
$L_1$  = liquid entering, lb/min/ft<sup>2</sup> of inlet.

$a, b, c$  are constants associated with the type and physical conditions of the system. For the unit of Pollak and Work [57],

$$b = 1.85 \text{ and } c = 0.00643$$

#### LIQUID-SOLIDS CYCLONE (HYDROCYCLONES) SEPARATORS

This type of solids removal device (Figures 6-57a and b) is a relatively low-cost approach to remove/separate solids from solid/liquid suspensions. The incoming feed to such a unit is injected along



1. Feed inlet and overflow connections are elastomer lined spool-piece adapters.

2. The top cover plate has all wetted surfaces lined, including the area mating with the overflow adapter.

3. The vortex finder is completely elastomer covered.

4. The molded liners for the inlet head, cylinder section, and conical sections have integral molded gaskets for sealing at the flanged joints. Molded liners and vulcanized linings are offered in gum rubber, polyurethane, nitrile rubber, butyl, Neoprene®, Viton®, Hypalon®, and other liner materials can be supplied.

Many of the molded elastomer liners are interchangeable with ceramic liners of silicon carbide or high purity alumina.

5. All metal housings are of cast or fabricated mild steel. Standard housings are for system pressures up to 25 psi, and special designs are available for higher system pressures.

**Figure 6-57b** Liquid–solids cyclone fabricated to resist corrosion and abrasion. (By permission from Krebs Engineers.)

the inner wall where the centrifugal force causes rotation at high angular velocity. The kinetic energy of this feed is converted to centrifugal force. The coarse/heavier particles will be concentrated at the bottom as underflow. Most of the feed liquid and part of the fine solids will discharge through the vortex and overflow.

This unit is good to pre-thicken feed to centrifugal filters and similar applications. One cyclone may satisfy a requirement, or the units can be arranged in parallel for large capacities or in series for

removal of extreme fines. Countercurrent wash system is given in Figure 6-58. Solids as small as  $10\text{ }\mu$  can be separated.

These units are made of abrasion-resistant metals, solid plastics, or with corrosion-wear-resistant plastic liners, such as molded rubber and elastomers; for example, butyl, Hycar,<sup>®</sup> Hypalon,<sup>®</sup> urethane, and metal alloys, silicon carbide, alumina ceramics. These units require little or no maintenance.

The manufacturer requires complete solids and/or liquids data, feed size analysis, and requirements for separation. In some instances, it may be best to have a sample tested by the manufacturer in their laboratory.

Zanker, and Reid and Voller [58, 59] give good performance analysis of these designs.

### SOLID PARTICLES IN GAS/VAPOR OR LIQUID STREAMS

The removal of solid particles from gas/vapor or liquid streams can be accomplished by several techniques, some handling the flow "dry," others wetting the stream to settle/agglomerate the solids (or even dissolve) and remove the liquid phase from the system with the solid particles. Some techniques are more adaptable to certain industries than others. Figure 6-59 illustrates typical ranges of particle size removal of various types of common equipment or technique. All of these will not be covered in this chapter. Attention will be directed to the usual equipment associated with the chemical/petrochemical industries.

### INERTIAL CENTRIFUGAL SEPARATORS

Specification Sheet (Figure 6-60) can be used as a guide in summarizing and specifying conditions for this type of equipment.

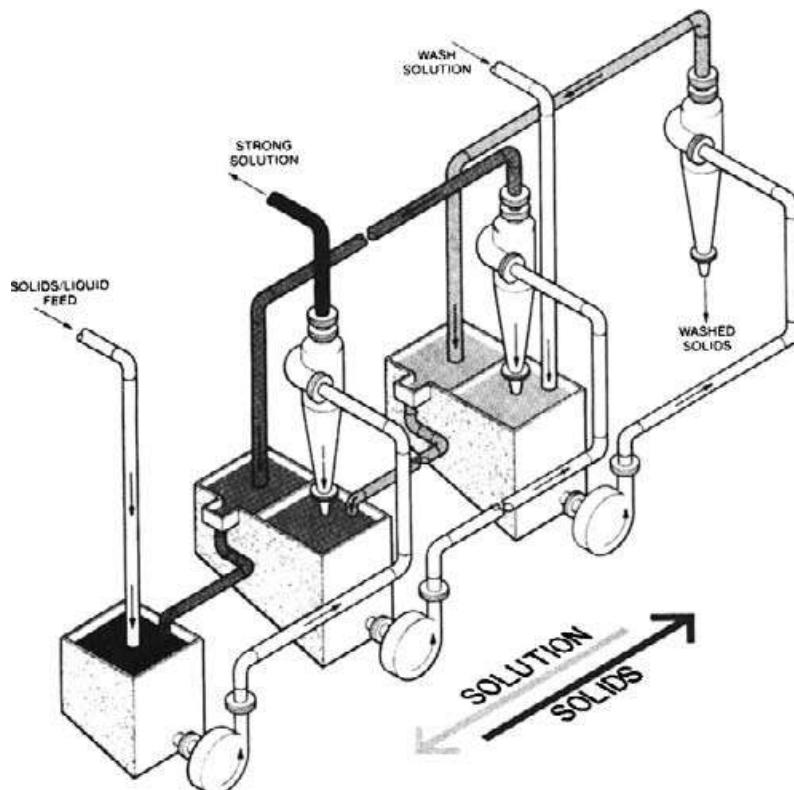
There are a few mechanical arrangements that use external power to exert centrifugal force on the gas particle stream. The fan type blades direct the separating particles to the collection outlet. Figures 6-61 and 6-62 show such a unit. These units are compact and have been used in various dust applications. However, caution should be given to avoid installations involving sticky or tacky materials which might adhere to the walls and blades of the unit. The efficiency of these is about 90–99% similar to a small, high pressure drop cyclone. The air-handling performance can be predicted using the fan laws.

### SCRUBBERS

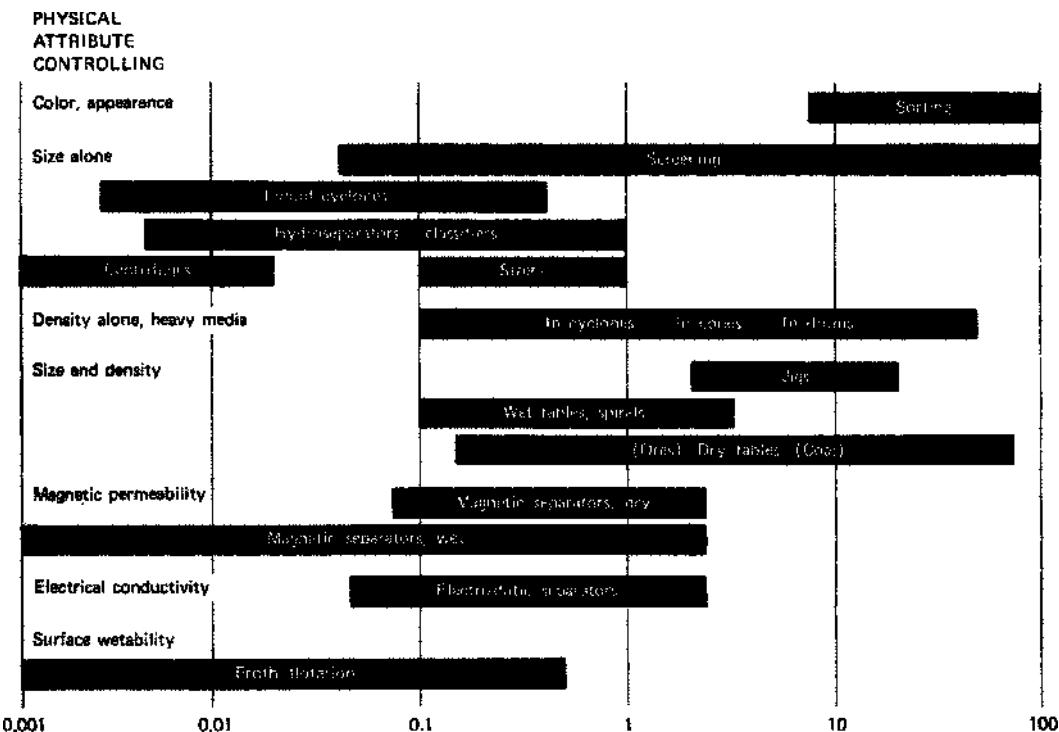
Scrubber separators use a liquid to form some type of liquid surface (spray droplets, film, etc.) to assist the internal arrangements of the separator in the separating action. Essentially the incoming dust or liquid particles are wet by the action of the liquid (usually water or oil) and are made larger and/or heavier and thus can be separated from the moving stream. There are many types and styles of units falling under this classification (Figures 6-63–6-69a). Toy and Bonady [60] provide a good summary of manufacturers and their products for wet scrubbing.

One or more of the following mechanisms are employed in the separating action of the wet scrubbers.

1. Impingement on internal parts.
2. Wetting of particle to help agglomerate and prevent re-entrainment.
3. Diffusion – dust particles deposited on the liquid droplets. Predominant for the submicron and particles up to about  $5\text{ }\mu$ .
4. Humidification aids in flocculation and agglomeration of particles.



**Figure 6-58** Cyclones used for countercurrent wash system. (By permission from Krebs Engineers.)



**Figure 6-59** Size ranges where particular solid–solid/solid–liquid separation techniques can be applied. (By permission from Roberts, E.J., et al. *Chemical Engineering*, 29 Jun, 1970 [31].)

5. Condensation will cause particle size to grow if gas cooled below its dew-point.
6. Dust Disposal – running film action of liquid washes dust and collected liquid out of scrubber.
7. Gas Partition – segregates gas into small streams and segments when flowing through a liquid or foam.
8. Electrostatic Precipitation – the electrical charging of the liquid droplets may come about by the interaction of the gas and liquid streams. Not much known of this action.

The separating ability of most units is limited to  $5\mu$  particles. However, some will take out 1 to  $5\mu$  particles at a sacrifice in collection efficiency. Due to the peculiarities of each system as well as the equipment available to perform the separation, it is good to consult manufacturers regarding expected performance. Quite often they will want to run test units, particularly on difficult separations. Montrose [12] and Perry [5] give good descriptions of the various types of equipment illustrated in Figures 6-63–6-69a.

Figures 6-69a and b use a floating valve variable orifice opening as used in distillation contacting on one or more trays included in the manufacturer's design. This provides for good contact to wet down the solid particles as well as scrub many water soluble gas/vapors in the incoming stream (such as chloride, sulfur, and nitrogen compounds). Heat and mass transfer can take place under these conditions. The pressure drop through this type of unit typically ranges from 1 in. water to 2 in. of water for a fivefold increase in gas flow rates. Particle removal can go as low as  $0.5\mu$  to greater than  $30\mu$ . Usually a wire mesh entrainment pad is mounted in the outgoing "clean" vapors to knock out liquid entrained particles, not solids.

#### CLOTH OR FABRIC BAG SEPARATORS OR FILTERS

Beddow [61] provides additional details beyond the bag filter applications, and Boothroyd [62] provides a technical and analytical review of flowing gas–solids suspensions.

Filters of this type or class may be of the large bag filter type for large volumes of low pressure dust-laden gases or vapor, or of the generally smaller cartridge or pack types for gas/vapors or liquids containing suspended solid materials.

Figures 6-70–6-72 show several units of the bag. The bags may be of cotton, wool, synthetic fiber, and glass or asbestos with temperature limits on such use as  $180^\circ\text{F}$ ,  $200^\circ\text{F}$ ,  $275^\circ\text{F}$ , and  $650^\circ\text{F}$  respectively, except for unusual materials (Tables 6-15a and b). These units are used exclusively on dry solid particles in a gas stream, not being suitable for wet or moist applications. The gases pass through the woven filter cloth, depositing the dust on the surface. At intervals the unit is subject to a de-dusting action such as mechanical scraping, shaking or back flow of clean air or gas to remove the dust from the cloth. The dust settles on the lower section of the unit and is removed. The separation efficiency may be  $99\%+$ , but is dependent upon the system and nature of the particles. For extremely fine particles a pre-coat of dry dust similar to that used in some wet filtrations may be required before re-establishing the process gas–dust flow.

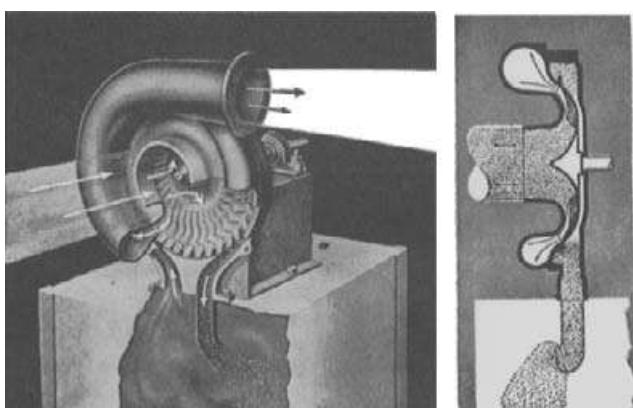
For heavy dust loads these units are often preceded by a dry cyclone or other separator to reduce the total load on the bags. Suggested air-to-cloth ratios are given in Table 6-16.

**Specifications.** The details of specifications for bag filter dust collectors are important to a proper and operable design selection. There are many variables which must be furnished by the manufacturer so that the user can understand how the unit operates mechanically and the unit's dust-loading capabilities. The larger the air/cloth ratio for the unit, the smaller will be its physical

**Specification Sheet**  
**Gas Phase Centrifugal Entrainment Separator**  
**(Liquid or Solid Particles)**

1. Application : (Describe service application of unit when possible) \_\_\_\_\_
2. Fluid Stream: \_\_\_\_\_ Composition: (Vol. %) \_\_\_\_\_
3. Entrained Particles : (Liquid or solid) \_\_\_\_\_  
Composition  
 a. Size range \_\_\_\_\_ microns (or Mesh) \_\_\_\_\_  
 b. Size percentage distribution: \_\_\_\_\_  
 c. True Specific Gravity \_\_\_\_\_ (of particle), referred to water = 1.0  
 d. Bulk density \_\_\_\_\_ of particle  
 e. Source of entrainment: (Boiling liquid, kiln dust, etc.) \_\_\_\_\_
4. Operating Conditions:      Maximum      Minimum      Normal  
 Gas Flow rate      \_\_\_\_\_      \_\_\_\_\_      \_\_\_\_\_  
 Entrained Flow Rate      \_\_\_\_\_      \_\_\_\_\_      \_\_\_\_\_  
 Temperature, °F      \_\_\_\_\_      \_\_\_\_\_      \_\_\_\_\_  
 Pressure, psi      \_\_\_\_\_      \_\_\_\_\_      \_\_\_\_\_  
 Moisture Content      \_\_\_\_\_      \_\_\_\_\_      \_\_\_\_\_  
 Dew Point, °F      \_\_\_\_\_      \_\_\_\_\_      \_\_\_\_\_
5. Installation Altitude: \_\_\_\_\_  
 a. Normal barometer \_\_\_\_\_ mmHg
6. Nature of entrained material:  
 Solids: (a) Describe (dry, moist, sticky, at operating conditions) \_\_\_\_\_  
 (b) Hygroscopic: \_\_\_\_\_  
 (c) Angle of repose \_\_\_\_\_  
 Liquid: (a) Describe: (Corrosive, oily) \_\_\_\_\_  
 (b) Surface tension at operating conditions: \_\_\_\_\_  
 (c) Viscosity at operating conditions: \_\_\_\_\_
7. Insulation required: \_\_\_\_\_ Reason \_\_\_\_\_
8. Construction Features:  
 (a) Describe separator location in system \_\_\_\_\_  
 (b) Indoors, outdoors, inside another vessel (Provide sketch if possible) \_\_\_\_\_  
 (c) Storage required for collected dust or liquid \_\_\_\_\_ (h)  
 (d) Preliminary size inlet connection: \_\_\_\_\_ in.  
 (diam., rect., sq.) \_\_\_\_\_  
 (e) Type of dust removal required \_\_\_\_\_  
 (f) Suggested Materials of Construction  
 shell: \_\_\_\_\_ internals \_\_\_\_\_
9. Special conditions: \_\_\_\_\_

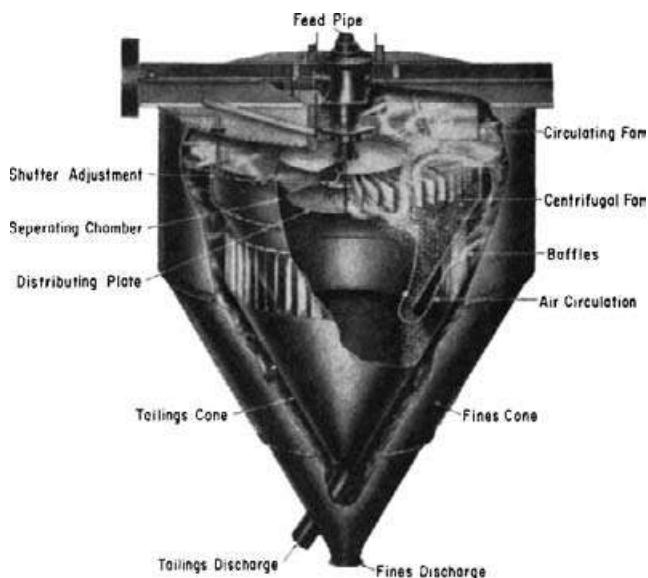
**Figure 6-60** Specification Sheet, gas phase centrifugal entrainment separator. (liquid or solid particles)



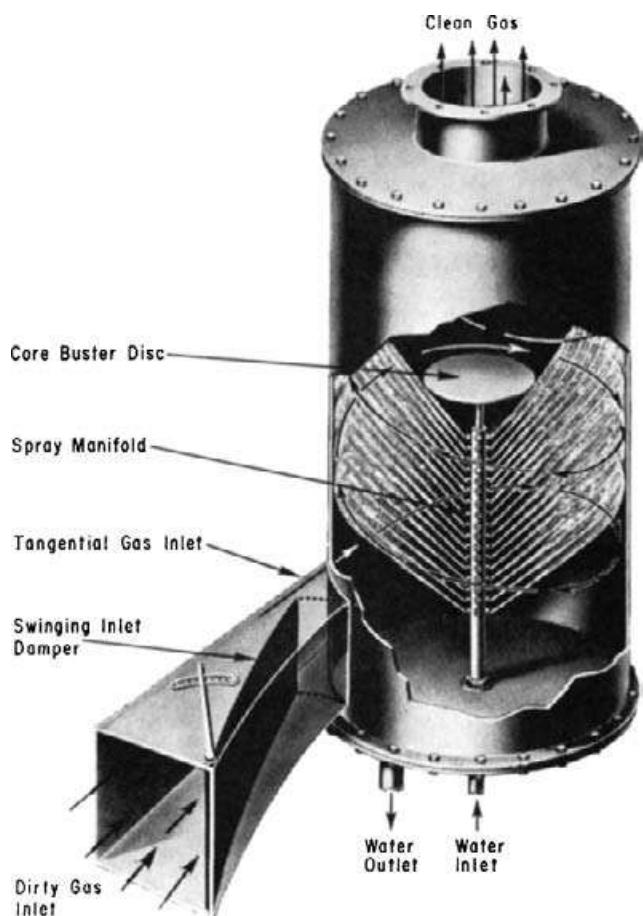
**Figure 6-61** Inertial centrifugal dust separator. (Courtesy of American Air Filter Co.)

dimensions and, generally, cost; however, the higher will be the frequency of cleaning. This can be quite troublesome; therefore low values of this ratio are preferable, consistent with the analysis of overall performance.

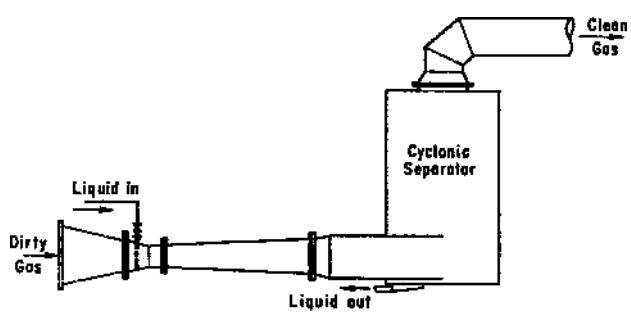
The removal or filtration of the entrained dust from the gas stream is accomplished by passing the mixture through a sufficiently porous fabric filter bag(s) (Table 6-17). These bags allow some air to flow through and are either cylindrical tubes or oblong tubes/bags. The dust is retained on the outside or inside (depending on unit design) of the bag surface and the small spaces between the fibers of the cloth (or felt). This dry cake builds up and acts as a pre-coat and then as the actual filtering medium as the dust particles build up. After a period of time, unique to the filter system of dust-laden air plus bag type, the pressure drop will build up. (These are low pressure and low pressure drop systems.) Therefore, the dust or "cake" is removed (cleaned) from the outside of the bag by internal arrangements in the "bag house" or housing by such techniques as (1) shaking or vibrating the bag or bag assembly to drop the dust into an integral hopper while there is no flow of air-dust feed into the unit or compartment, or (2) back pulse with jets of air into each bag (Figure 6-72). The criterion



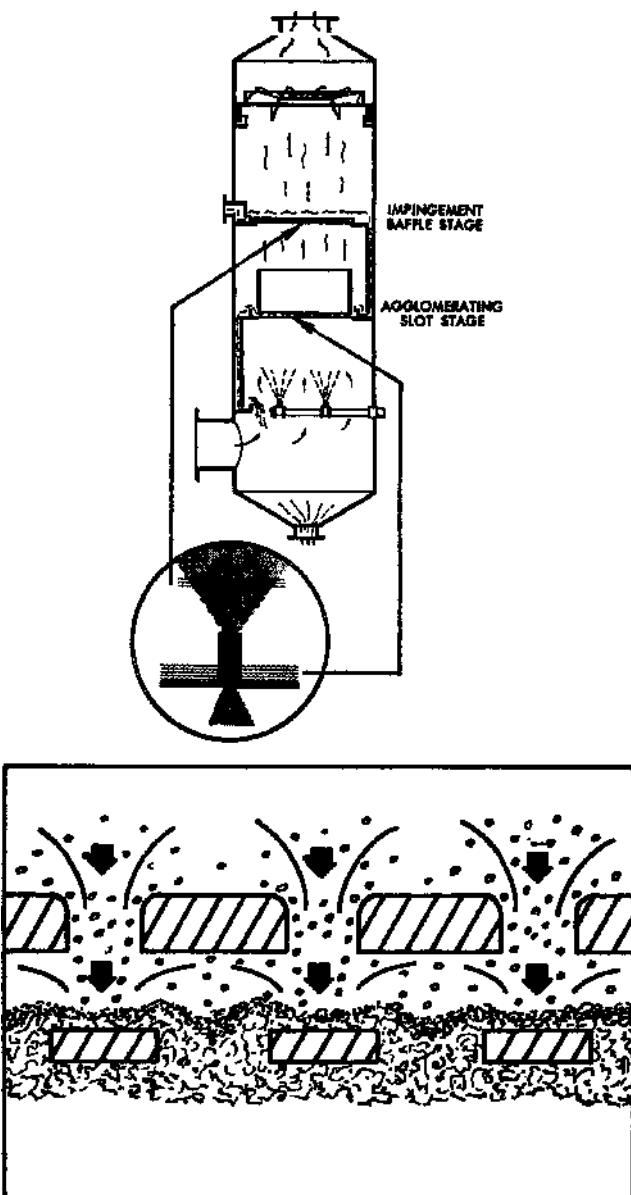
**Figure 6-62** Inertial centrifugal dust separator. (Courtesy of Universal Road Machinery Co.)



**Figure 6-63** Cyclonic scrubber. (Courtesy of Chemical Construction Corp.)



**Figure 6-64** Venturi scrubber. (Courtesy of Chemical Construction Corp.)



**Figure 6-65** Impingement scrubber. (Courtesy of Peabody Engineering Corp.)

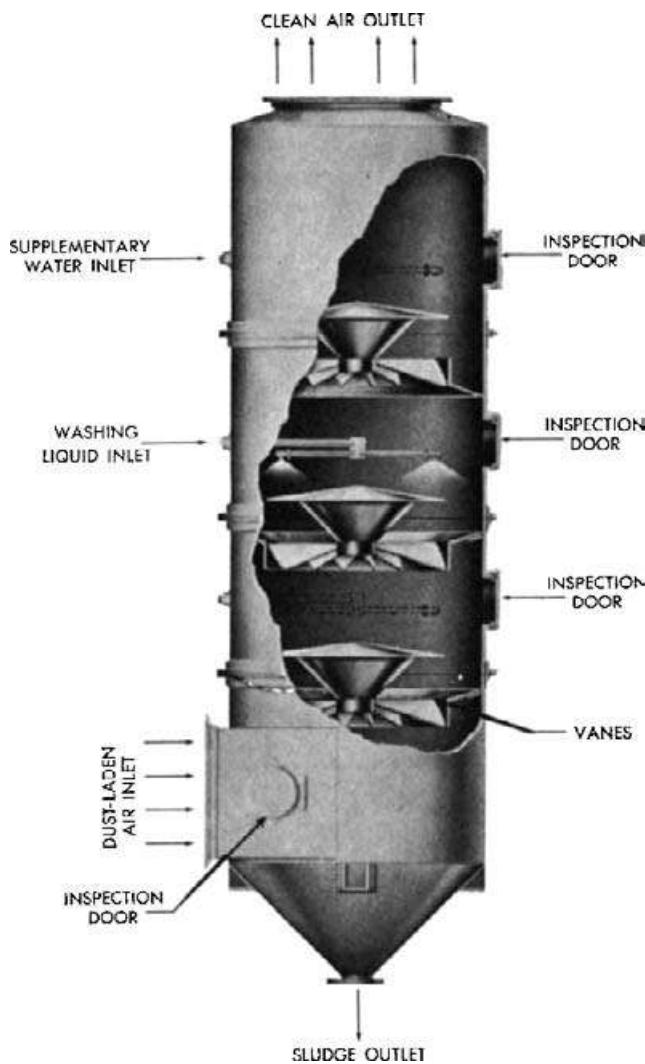


Figure 6-66 Spray scrubber. (Courtesy of The Ducon Co., Inc.)

should be a constant pressure drop across the fabric for a fixed air flow and a specified dust loading [63]. The air-to-cloth ratio so often used is only useful when comparing a particular manufacturer's equipment for handling different materials, and not for comparing manufacturers. Kraus [64] is an excellent summary of many details associated with specifying and selecting bag filters.

The following are suggested filter specifications:

1. Performance: define air/gas and dust rates, particle size distribution, and percent of particle sizes.
  - a. temperature at inlet to baghouse
  - b. moisture concentration, dewpoint
  - c. chemical composition of vapor and dust, including any abrasive, hydroscopic, or other characteristics.
2. Define dust recovery, as percentage below a certain particle size.
3. Indicate, if known, preferred bad material that will withstand environment, for example, fibers of glass, polyester, Teflon®, Nomex®, polypropylene, polyethylene, cotton, wool, nylon,

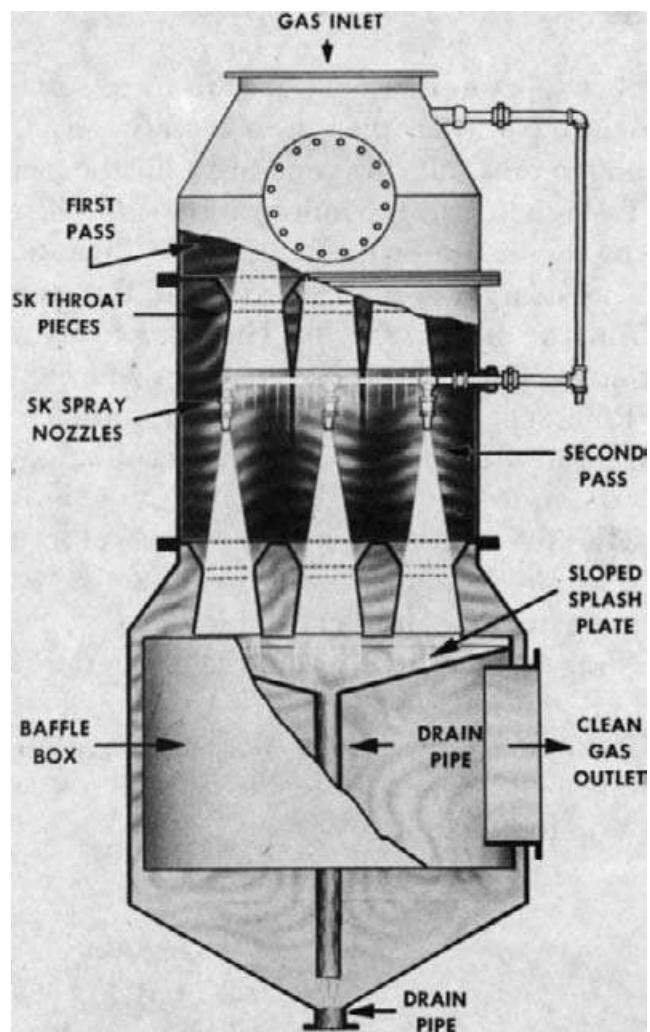


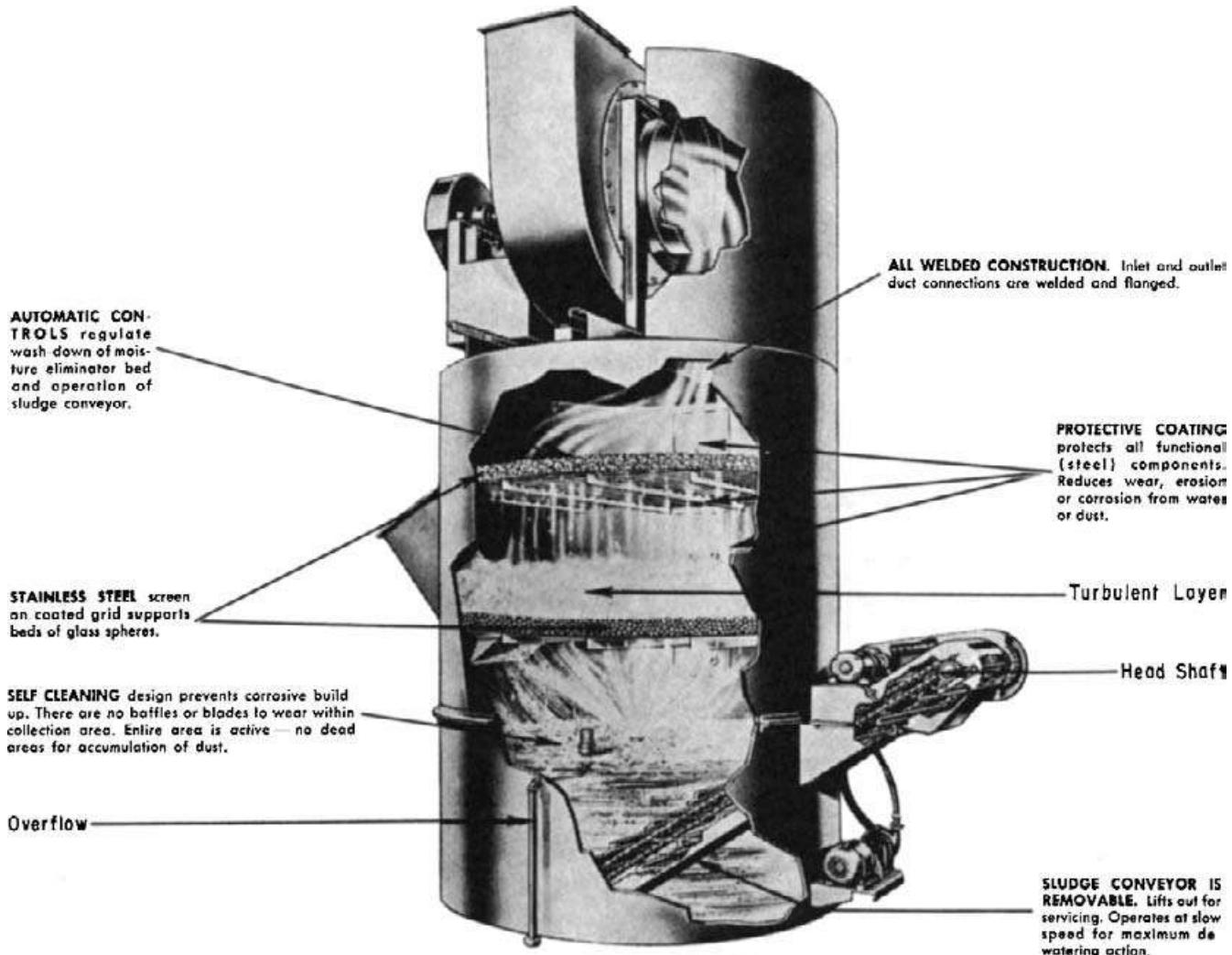
Figure 6-67 Spray scrubber-fume scrubber arranged for vertical down-flow. (Courtesy of Schutte & Koerting Co.)

Orlon®, Dacron®, and Dynel®. The type of weave of fibers should be recommended by the manufacturer. The fabrics may be felted or woven [63, 65] in weaves of plain, satin, or twill, and should be resistant to any corrosive material in the solid particles or the gas stream.

**4. Manufacturer should recommend**

- a. bag size (diameter, length);
- b. bag holding hardware: anti-collapse spreader rings, snap rings, and so on;
- c. number of baghouse compartments, number of bags per compartment;
- d. air/gas flow cycle to compartments;
- e. complete description with mechanical details of bag cleaning system (shaking, air-jet, etc.);
- f. dust removal arrangement.

The cleaning system set up for a particular bag house will determine whether the filtering system operates continuously or batch-wise/intermittently. Some systems operate as a continuous batch, with sections of the entering chambers being isolated by valves



**Figure 6-68** Tray-type scrubber with continuous sludge removal. (Courtesy of National Dust Collector Corp.)

to automatically switch from one section of one bag to another. Thereby, one or more bag groups/sections filter while another one or more are not operating, but are in the dust removal cycle. The permeability of the fabric is generally stated as the clean airflow in cubic feet/minute/square feet of fabric at a pressure differential of 0.5 in. water as determined by the ASTM standard D-737 (Frazier Test) [63]. Whereas this test is useful, several fabrics may have the same permeability yet have different fiber surfaces, and thereby do not perform the same for a specific application.

The felted fabrics are generally used for maximum recovery of product and are used at high face velocity for airflow-to-cloth-area ratio. The felt promotes the greatest dust collection surface. Monofilament fibers require special attention to ensure a uniform open space between the filaments.

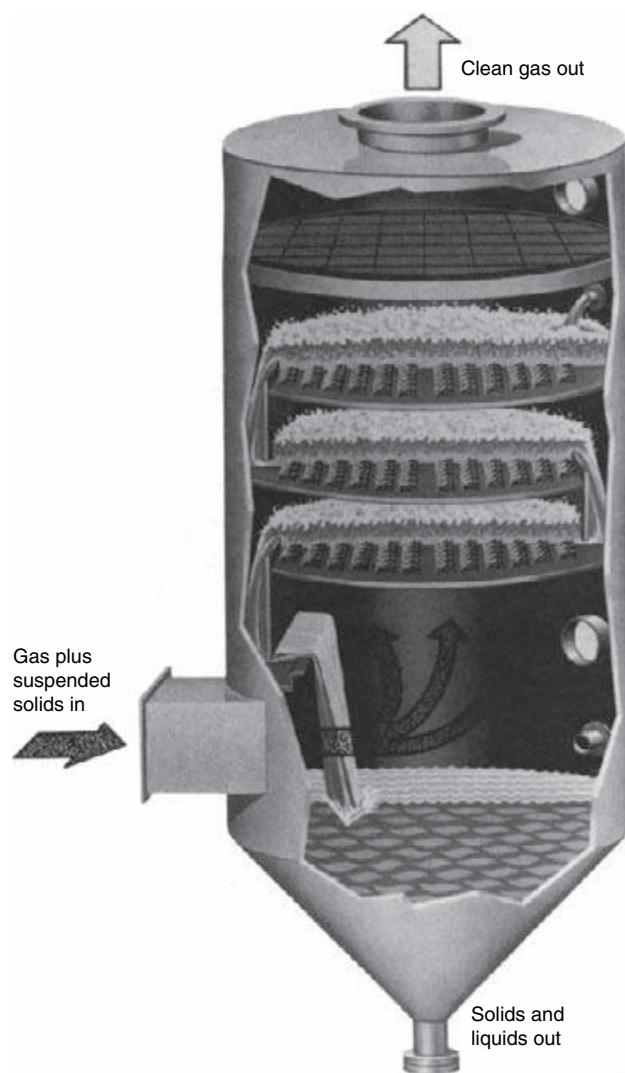
The woven fabrics have various yarn patterns for different spacings between the yarn fabrics (Table 6-17). There is a wide variety of choices for not only the materials of construction but the tightness of weave and the size of the yarn. All of these factors along with the others noted earlier make the selection of bag fabric an art that requires manufacturer's and plant's actual field tests. Woven fabrics have a low ratio of weave openings for yarn area

and generally have a limited face velocity for air flow of about 1.5 – 3.0 ft<sup>3</sup>/min/ft<sup>2</sup> [63].

Newer fabrics, not in common use but in development, test, and field trials, are described for higher temperature applications by Parkinson [66]. Application to 400 – 2100° F are potentially available using ceramic fibers Nextel 312®, laminated membrane of expanded PTFE on a substrate, polyimid fiber P-84, Ryton® polyphenylene sulfide, and woven fiberglass. The heat and acid resistance of these new materials varies; however, they can be handled at higher temperatures.

New cartridge designs for bag houses will allow improved servicing and cleaning techniques. It is important to keep bolts, nuts, and other potentially loose items to a minimum inside the unit, as vibration from air/gas flow and bag cleaning can loosen nuts, break small welds, and ultimately tear holes or rip bags. The bag construction is likewise extremely important, since loose edges and “unlocked” seams will fray and tear, allowing fibers into the product dust. The bag construction must have straight seams in order for them to bend and flex properly on cleaning and/or loading.

Cartridge filters may be single units or clusters in a single container or canister. Figures 6-73–6-79 illustrate typical units.

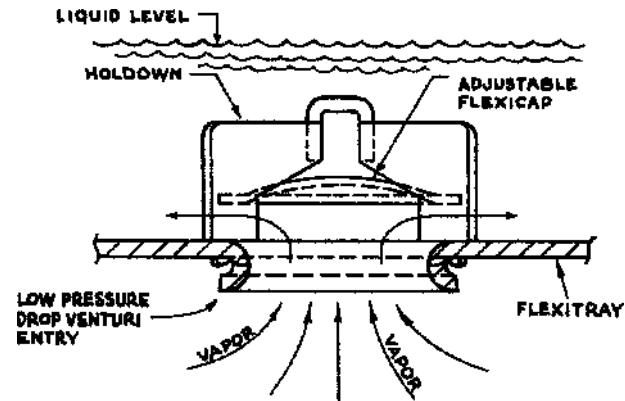


The scrubber is comprised of one or more trays. Each tray contains numerous venturi openings. Each of the MultiVenturi openings is surmounted by a spider cage holding a floating Flexicap (see insert). In addition, each tray is equipped with one or more "downcomers" and weir flow baffles that control the scrubbing liquid as it flows across the tray and then to the tray below.

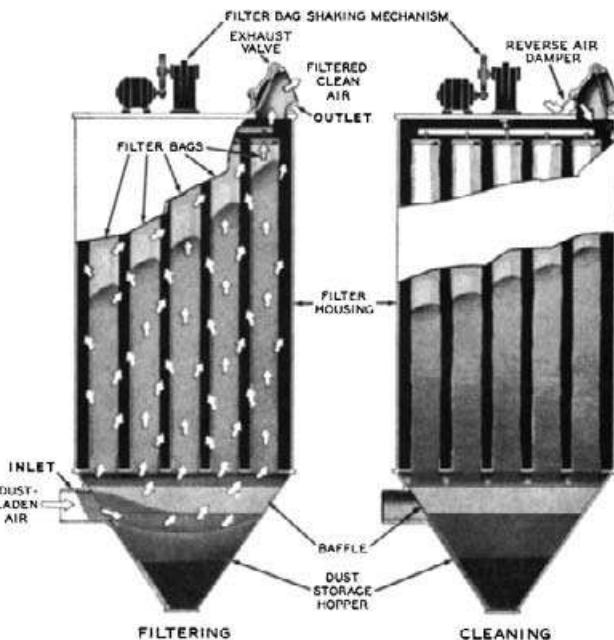
**Figure 6-69a** Variable orifice MultiVenturi Flexitray® scrubber at essentially constant pressure drop maintains good efficiency over wide flow rates. (By permission from Koch Engineering Co., Inc.)

These may be designed to filter suspended solids from gases or liquids. Table 6-17 presents representative physical property and application data for the more commonly used filter media. These media may be in filament, fiber, or "felt" form and arranged by weaving techniques to control the pore or free spaces to specific size for removal of various sizes of particles. The particle size retention listed in the table ranges from  $0.006\text{ }\mu$  to over  $100\text{ }\mu$ . A micron is often termed "micrometer" or a millionth of a meter, using symbol  $\mu\text{ m}$ .

Cartridges as illustrated are considered "throw-away" and are removed from service when the pressure drop builds up to a predetermined value, or when the effluent changes color or becomes opaque with suspended material breaking through. The flow in most applications is from outside cartridge to inside and



**Figure 6-69b** Adjustable "floating" caps for vapor flow. (By permission from Koch Engineering Co., Inc.)



**Figure 6-70** Bag filtration with mechanical shaking for bag cleaning. (Courtesy of Dracco Div. Fuller Co.)

into the hollow metal or plastic collection tubes. It then flows into the outlet pipe to the process. Materials for these cartridges are most commonly selected from cellulose, glass fibers, polypropylene (woven and non-woven) fibers, or monofilaments, molded resins, ceramics, or resin-impregnated fiberglass. The last three are termed "depth" filters, as they can hold a large amount of solids before the pressure drop builds up excessively. "Surface" filters are usually made of paper, non-woven fabrics, or cast membranes and are usually pleated to provide more working surface area. This type is fabricated from sheets of porous non-woven fabric often used for the absolute capture of sub-micron particles and has a sharp cutoff in particle size retention [67]. Yarn wound filters often have a graded density or decreasing pore size structure.

To aid in selection of the most probable successful filter media for the service, the summaries of Tables 6-15a and b can be a useful guide [68]; however, for some applications, actual testing in the plant using plant fluid streams can be the most conclusive. This plant testing is not necessary for every situation because the

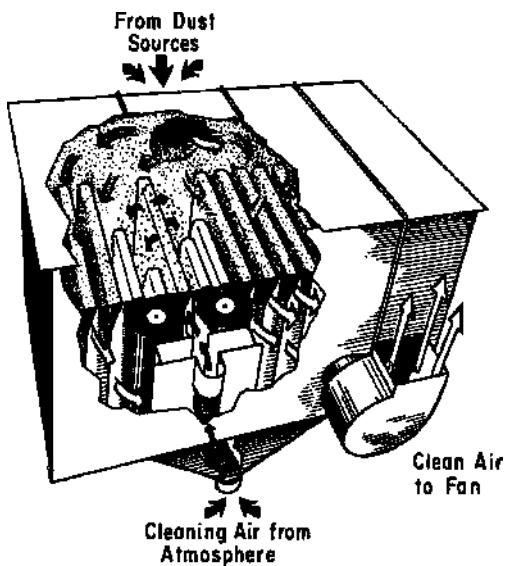


Figure 6-71 Bag filtration with continuous reverse air cleaning. (Courtesy of W.W. Sly Mfg. Co.)

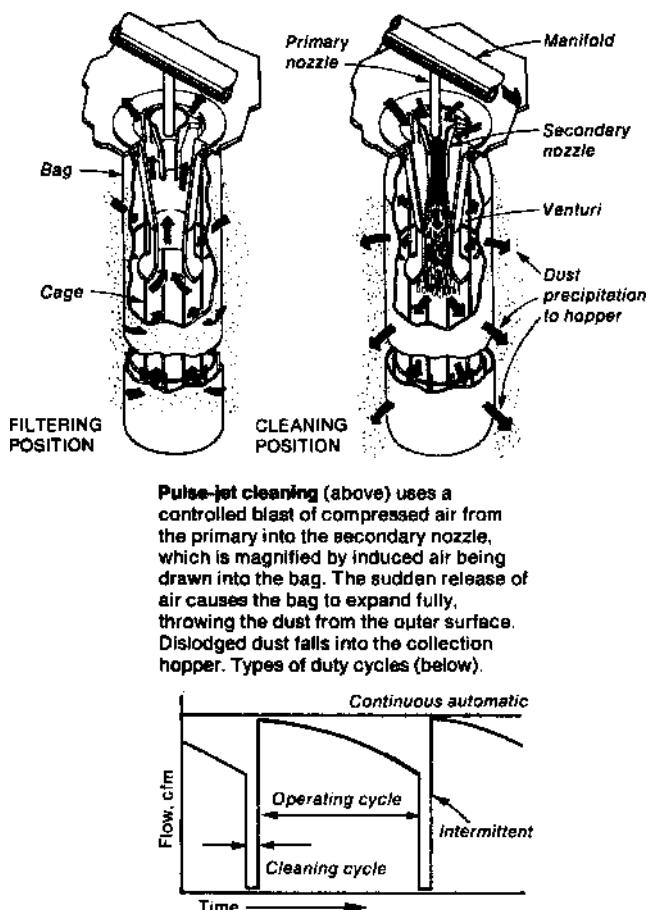


Figure 6-72 Pulse-jet air cleaning of fabric bags. (By permission from Power, Nov 1975, McGraw-Hill Co., Inc., New York, p. 41.)

manufacturer has large data files to often aid in a good selection. Generally the ability to collect solids at low flow rates is greater for the wound filter. Because the suspended particles are "captured" by different physical mechanisms depending on the particle size, shape, density, and concentration, all cartridges do not perform the same. The "capture" may be by (1) direct interception (2) sieving, and/or (3) bridging [69] (Figure 6-80). The cartridges from one manufacturer are generally consistent in performance; however, all cartridges from just any manufacturer may not be interchangeable in performance.

The micron ratings of a cartridge are intended to indicate the smallest particle that will be retained by the pores of the filter element. Often a "rough-cut" pre-filter is installed ahead of a final or "polishing" filter in order to increase the life of the final unit. Unfortunately, the method for determining the micron rating is not a universal standard between manufacturers. Thus, one manufacturer's "50 $\mu$ " filter may not perform the same as that of another manufacturer with the same rating number. The only reliable approach is to send the manufacturer an actual sample of the fluid and let it be tested in order to select the filter for the job. Alternatively, test the unit in your plant's field application [68].

An important feature of these cartridge units is the mechanism for assembling one or more in the housing. The top-bottom-sealing mechanisms determine what style of cartridge is required (open both ends, open one end) and the method of pressure loading/sealing each cartridge into its bracket in the housing. The housing may hold 1 or 40 cartridges, and the assembly inside to prevent leakage and cross-contamination is essential to good performance as a filtering device. The housings can be made of various metals (carbon steel, stainless, alloy) or plastic-lined steel using corrosion-resistant polymers, or elastomers, or solid plastic.

The cartridges can be selected to be useful over the range of low to high viscosities, that is, 100 000 cP with temperature ranges to 750°F at higher pressure of up to 3000 psi [68]. Usually for the average application, the concentration of the suspended solids is not over 100 ppm, but can be higher. These units do not perform well with pressure pulsations or surges in the system. Note the differences in expected performance of Figure 6-81 between a pleated cartridge. This does not necessarily mean that all cartridges perform in this manner, but these are typical of expected performance curves. When examining particle retention ratings, examine Shucosky [69].

*Note:* (a) Designations for both nominal and absolute ratings are based on the measure of a particle size, not a pore size. (b) Ratings are based on arbitrary laboratory tests by the filter manufacturer and can vary in actual plant conditions as previously discussed.

For some critical applications (such as polymer melt, beverage, or pharmaceutical filtration), it may be important to avoid cartridges that have a "nap" or "fuzz" on the fiber used, because these extremely fine fibers tend to break off and drift through the cartridge and go out with the finished product, thereby creating a visual acceptance problem, if not outright contamination.

In actual practice some companies have cartridges that will remove particles up to 0.25  $\mu$ m in size. The smaller the particle size that is specified to be removed from the vapor or liquid, the higher will normally be the ultimate pressure drop or lower the holding capacity. In normal operation, the pressure drop initially is quite low, perhaps 1–3 psig depending on flow rate, but as the solids build up the pressure drop will rise from 10 to 35 psig, in which range most companies recommend replacement.

These replaceable cartridges or packs are the most commonly used; however, there are cartridges of wire mesh, sintered or porous metal which can be removed, cleaned, and replaced. Usually, the fine pores of the metal become progressively plugged and the

**TABLE 6-15a Partial List of Commercial Crossflow Microfilter Media in Materials and Geometries**

Material	Geometries					
	Pleated Sheet	Tubular	Spiral Wound	Tubular MC*	Hollow Fiber	Flat Sheet
<i>Polymers</i>						
Cellulosics	◆◆◆		◆◆◆		◆◆◆	◆◆◆
Polysulfone		◆◆◆	◆◆◆		◆◆◆	◆◆◆
Polyvinylidene fluoride		◆◆◆	◆◆◆			
Acrylic					◆◆◆	
Polytetrafluoroethylene		◆◆◆				◆◆◆
Polybenzimidazole			◆◆◆			◆◆◆
Polypropylene					◆◆◆	◆◆◆
Nylon	◆◆◆				◆◆◆	◆◆◆
<i>Ceramics</i>						
Alumina				◆◆◆		◆◆◆
Zirconia/alumina				◆◆◆		
Zirconia/sintered metal		◆◆◆				
Zirconia/carbon		◆◆◆				
Silica		◆◆◆				
Silicon carbide				◆◆◆		
<i>Sintered metal</i>						
Type 316 stainless steel		◆◆◆				
Other alloys		◆◆◆				

\*MC = Multi-channel monolithic elements

(Source: By permission from Michaels [68].)

cartridges lose capacity. They are often used for filtering hot fluids, or polymers with suspended particles, pharmaceuticals, and foods (liquids). In the case of polymers and other applications a special solvent and blow-back cleaning system may be employed.

The small cartridge units can be conveniently placed ahead of instruments, close-clearance pumps, or a process to remove last indications of impurities in suspension.

Other useful cartridges are as follows.

1. woven stainless steel (or other wire) wire screen mesh (Figures 6-82 and 6-83)
2. wire wound (Figure 6-82)
3. sintered metal (Figure 6-84).

The woven wire mesh types are formed to control the open space between the wires, thereby limiting the maximum size particle that can pass through. The cartridge is installed in cases or small vessels to facilitate quick replacement, or they can be arranged for backwash by use of proper piping connections. The wire-wound units have consistent spaces for uniform particle size filtering.

The sintered metal units have uniform permeability with void spaces approximately 50% by volume for some metals and manufacturing techniques. The pore sizes can be graded to remove particles from  $1\ \mu$  to  $20\ \mu$  for liquids and smaller sizes when used in gaseous systems (see Figure 6-85b). Metals usually used are stainless steel, nickel, monel, inconel, high nickel alloys, and special designs for unique services.

The pressure drops for these units are typically low, ranging from 0.2 to 10–15 psi. The woven wire mesh runs even lower

in pressure drops for the same or larger flow rates. Consult the manufacturers for specific application data.

With some types of particles the porous metal tends to plug, but they can usually be backwashed or washed with a solvent or acid/alkali to remove the particles from within the metal pores. This is one reason why manufacturer's testing or plant testing can be important to the proper selection. Once the internal plug-gauge has reached a point of reduction in flow-through capacity, it must be discarded. The actual cost of this type of cartridge is several times that of the non-metallic unit; therefore, the economics involving the life span of each unit should be examined.

## ELECTRICAL PRECIPITATORS

The electrical precipitator is a dry dust or liquid mist removal unit which utilizes the ionization of the process gas (usually air) to impart electrical charges on the suspended entrained particles and effect particle collection by attraction to an oppositely charged plate or pipe. This type of unit is in use in services which are difficult for other types of entrainment removal equipment. Figures 6-85–6-87 illustrate the usual fundamental action of these units.

For these units the usual particle size for removal is greater than  $2\ \mu$  with a loading rate greater than  $0.1\text{ gr}/\text{ft}^3$ , with a collection efficiency of  $99\%\pm$ . The pressure drop is very low for a range of gas velocity through the unit of 100–600 ft/min [70]. Operating temperatures can be as high as  $1000^\circ\text{ F}$  and above [71].

To improve the efficiency of collection, several units can be installed in series. The plate type unit is the most common design for dry dust removal, while pipe design is mainly for

**TABLE 6-15b Partial List of Crossflow Microfilter Media in Chemical Service Applications**

Chemicals Table II	Compatible media												
	Ceramics	Acrylics	Carton	Poly benzimidazole	Polysulfone	Polytetra fluoroethylene	Polypropylene	Polyvinylidene Fluoride	Other Polymers	Sintered Halide resistant Alloys	Sintered 316 Stainless Steel	Other sintered Media	Sintered Chloride-resistant Alloys
Alkanes, alkenes and aromatic hydrocarbon, below 100°C	█		█			█			█	█	█	█	█
Oxygen-containing organics, below 100°C	█	█		█	█	█	█		█	█	█	█	
Chlorinated organics, below 100°C	█	█		█	█	█			█	█	█	█	
Esters, below 100°C	█	█		█	█	█	█		█	█	█	█	
Organics at 100–200°C	█	█		█	█	█			█	█	█	█	
Organics at 200–600°C	█	█							█	█	█	█	
Organics at 600–900°C	█	█							█	█	█	█	
<i>Gases</i>													
Inert, or low reactivity	█		█	█	█	█	█	█	█	█	█	█	
Oxygen	█												
Chlorine	█		█			█					█		
Other reactive gases	█				█								
<i>Aqueous solutions</i>													
pH = 3–7, no chlorides	█		█		█	█	█	█	█				
pH = 7–10, no chlorides	█		█		█	█	█	█	█				
pH = 0–3 (except HF)	█		█	█	█	█	█	█					
pH = 3–10, chloride present	█		█		█	█	█	█	█				
HF, with pH < 3	█				█	█	█	█	█				
pH = 10–13		█	█	█	█	█	█	█	█	█	█	█	
pH > 13		█	█	█	█	█	█	█	█	█	█	█	
Concentrated acids	█		█										
Steam (>100°C)	█		█		█	█	█	█	█	█	█	█	
Oxidants (e.g., bleach)	█		█		█	█	█	█	█	█	█	█	

(Source: By permission from Michaels [68].)

**TABLE 6-16 Suggested Air-to-Cloth Ratios for Dust Removal from Air**

Type of Dust	Ratio
Abrasives	2–2.5
Asbestos	2.5–3
Blast cleaning	3–3.5
Carbon	2–2.5
Cement (mills)	1.5–2
Cement (conveying and packing)	2–2.5
Clay	2–2.5
Coal	2–2.5
Feed	2.5–3
Graphite	1.5–2
Grinders	3–3.5
Gypsum	2–2.5
Lamblack	1.5–2
Limestone	2–2.5
Rubber	2–2.5
Salt	2.5–3
Silica flour	2–2.5
Soap	2–2.5
Soapstone	2–2.5
Talc	2–2.5
Wood flour	2–2.5

*Note:* Ratio is the volume in cubic feet per minute of dust-laden air to each square foot of active cloth area. If gain loading is above normal, ratios must be reduced.

(Source: By permission from Bulletin 104. The W.W. Sly Mfg. Co., Cleveland, OH.)

removal of liquid or sludge particles and volatilized fumes. The plates/pipes are the collecting electrodes, with the discharge electrodes suspended between the plates or suspended in the pipes [72–74].

In operation, the voltage difference between the discharge and the collecting electrodes sets up a strong electrical field between them [74]. The “dirty” gas with particles passes through this field, and the gas ions from the discharge electrode attach to the suspended “dirty” particles, giving them a negative charge. The charged particles are then attracted to the positively charged collecting electrode, discharging their charge on contact, becoming electrically inert.

Collected liquids flow down the pipes and drain to a collection sump. Collected solids are washed off the plates with water or other liquid. Sometimes the dust/solids can be removed by mechanically vibrating or knocking on the plates while the particles are dry. The electrical power of the precipitator is applied only to the particles collected, thereby allowing for large volumes of gas to be handled with very low pressure drop.

For corrosive gases/liquid particles, corrosion-resistant metals can be used for construction.

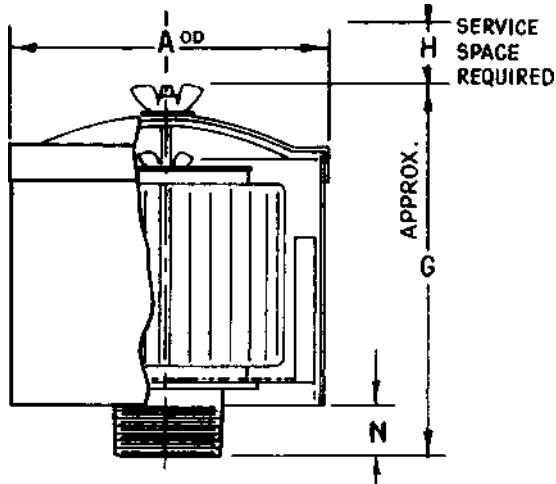
The performance of the unit involves the gas characteristics, analysis, velocity, flow rate, dust or liquid particle size and analysis, resistivity, and required final particle efficiency of removal. Some particle materials of high electrical resistivity prevent proper electrical operation.

Table 6-18 illustrates some industrial applications of electrostatic precipitators; however, it is not intended to be all-inclusive.

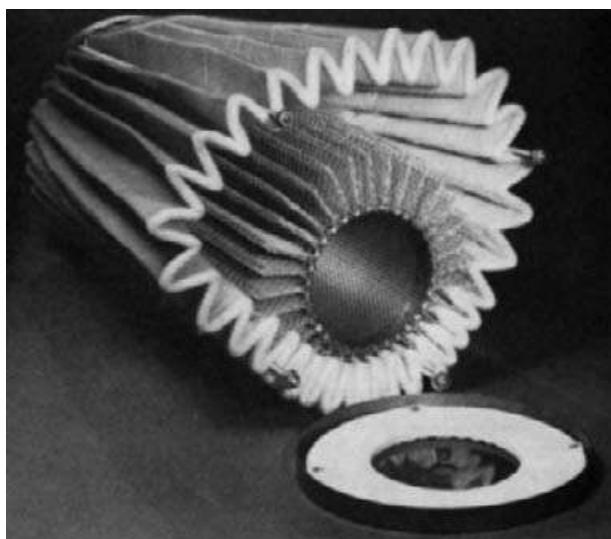
**TABLE 6-17 Physical Properties of Filter Media**

	Cotton	Wool	Glass	Polyester	Polypropylene	Nylon	Nomex (high temperature Nylon)	Rayon	Dynel	Teflon	Paper	Sintered Metal	Woven Wire Cloth	Porous Ceramics
Particle retention ( $\mu$ )	2 to >100	2 to >100	3 to >100	2 to >100	2 to >100	2 to >100	2 to >100	2 to >100	2 to >100	2 to >100	3 to >100	1 to >100	2 to >100	1 to >100
Contaminant-holding ability	G – E	E	F – G	G – E	F – E	F – E	F – E	F – E	F – E	G – E	G	F – G	F – G	F – G
Permeability	G	G	F	G	G	G	G	G	G	G	G	F	G	F
Chemical compatibility	F	F	G	G	E	G	G	F	E	E	P	E	E	E
Temperature limits ( $^{\circ}$ F)	200	200	700	300	200	250	450	200	200	450	200	1200	1200	2000
Strength	G	F	G	G	G	G	G	F	G	G	P	E	E	G
Abrasion resistance	G	G	P	E	E	E	E	G	G	E	F	E	E	G
Machineability (workability)	E	E	F	E	E	E	E	G	E	E	F	F	F	F
Cleanability	G	G	P	G	E	G	G	G	E	E	P	P	G	P
Cost	L	L	M	M	L	M	M	L	L	H	L	H	M	H

Note: P – poor; F – fair; G – good; L – low; M – medium; H – high.



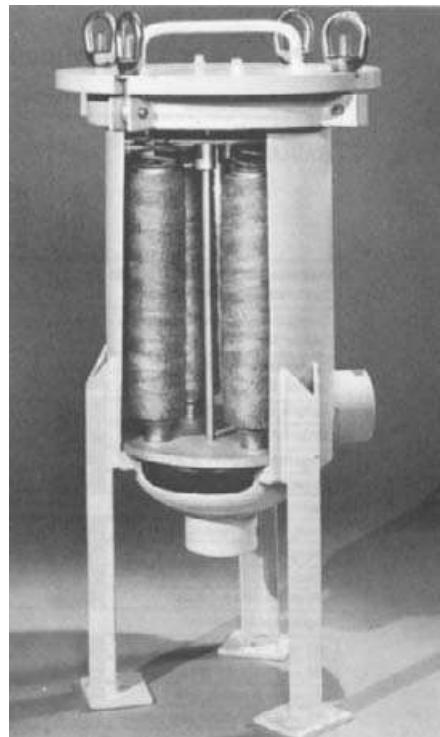
**Figure 6-73** Typical blower intake filter-silencer. Air to blower leaves through pipe connection, which may be screwed or flanged. (Courtesy of Dollinger Corp.)



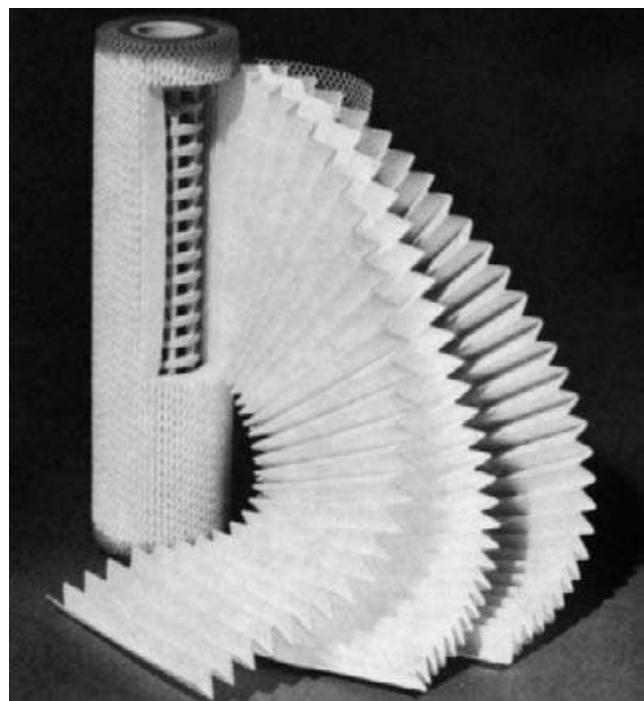
**Figure 6-74** Pleated radial-fan filter cartridge. Filtration is from outside to inside. (Courtesy of Dollinger Corp.)



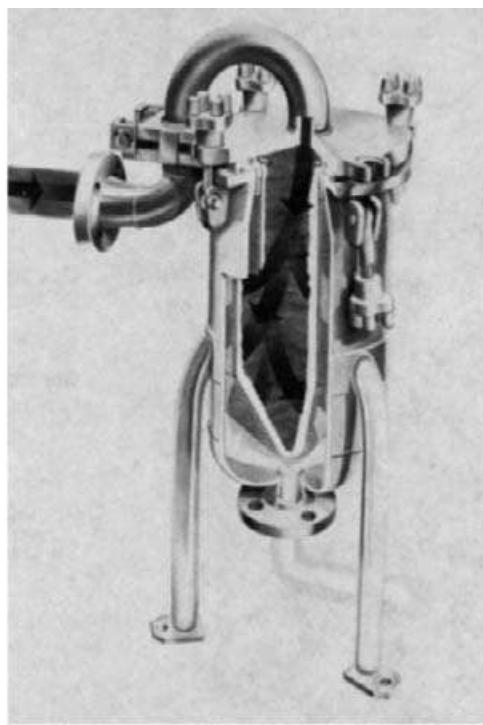
**Figure 6-75** Wound filter tube on stainless steel core. (Courtesy of Filterite Corp.)



**Figure 6-76** Cluster of filter cartridges in a single chamber. (Courtesy of Filterite Corp.)

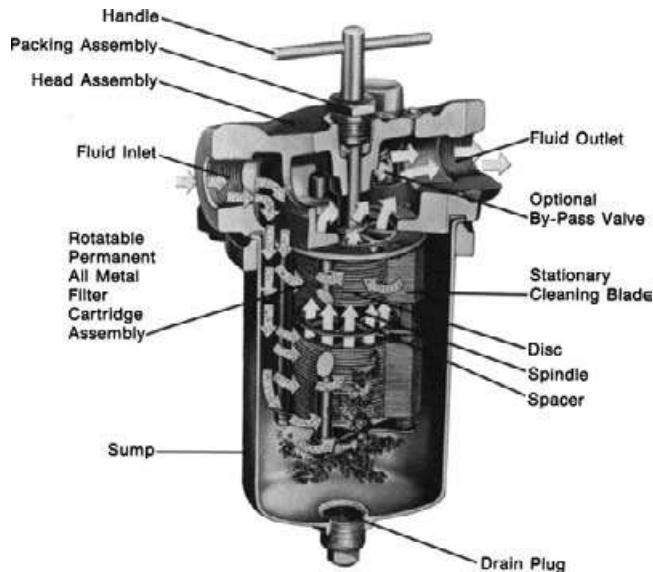


**Figure 6-77** Cartridge-type filter-pleated membrane. (Courtesy of Gelman Instrument Co.)

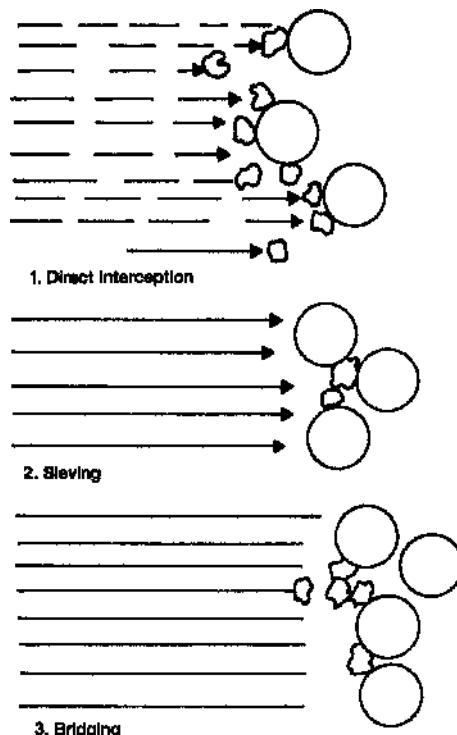


This three-dimensional cutaway drawing illustrates the filtering operation of the GAF® filter-bag pressure filter system, showing the flow patterns of unfiltered liquid through a preselected micronrated felt filter bag which renders the desired quality of filtered product.

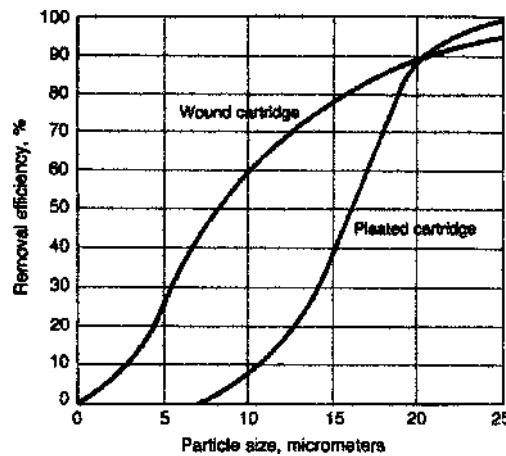
**Figure 6-78** Flow scheme for GAF filter-bag pressure filter system for liquids. (Courtesy of GAF Corporation, Chemical Group, Greenwich, C.)



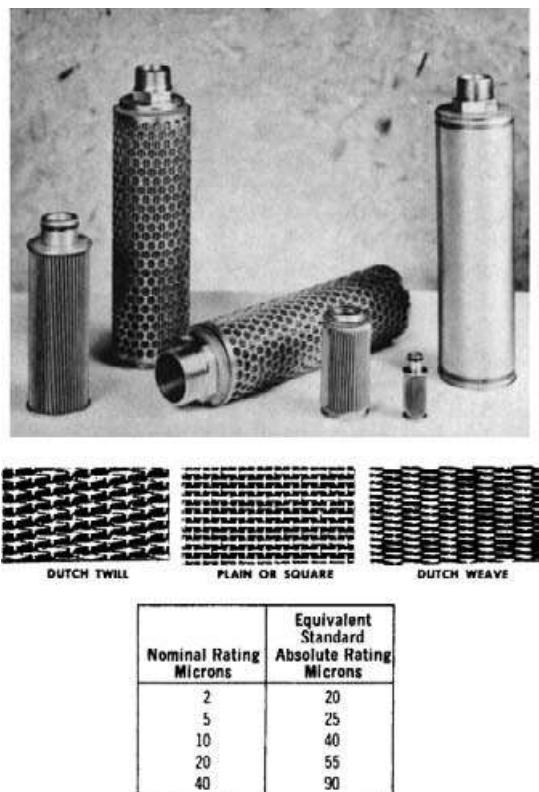
**Figure 6-79** Edge-type filter with the external online cleaning. (Courtesy of AMF Corp., Cuno Div.)



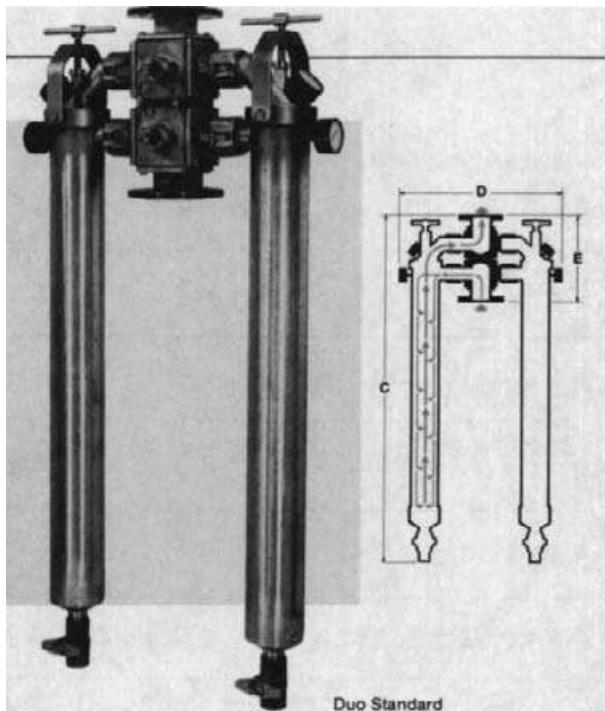
**Figure 6-80** "Capture" mechanism for cartridge filters. (Adapted by permission from Shucosky, A.C., *Chem. Eng.*, Vol. 95, No. 1, 1988, p. 72.)



**Figure 6-81** Pleated and wound cartridges differ in removal-efficiency profile. (By permission from Shucosky, A.C., *Chem. Eng.*, Vol. 95, No. 1, 1988, p. 72.)



**Figure 6-82** Woven wire mesh filter cartridges. (By permission from AMF Corp., Cuno Div., Catalog MP-20.1.)



**Figure 6-83** Tubular in-line pressure filter with reusable elements. The flow: unfiltered liquid enters the inlet port, flows upward, around, and through the media, which is a stainless steel or fabric screen reinforced by a perforated stainless steel backing. Filtered liquid discharges through the outlet (top) port. Because of outside-to-inside flow path, solids collect on the outside of the element so screens are easy to clean. (By permission from Ronningen-Petter® Engineered Filter Systems, Bulletin RP-2.)

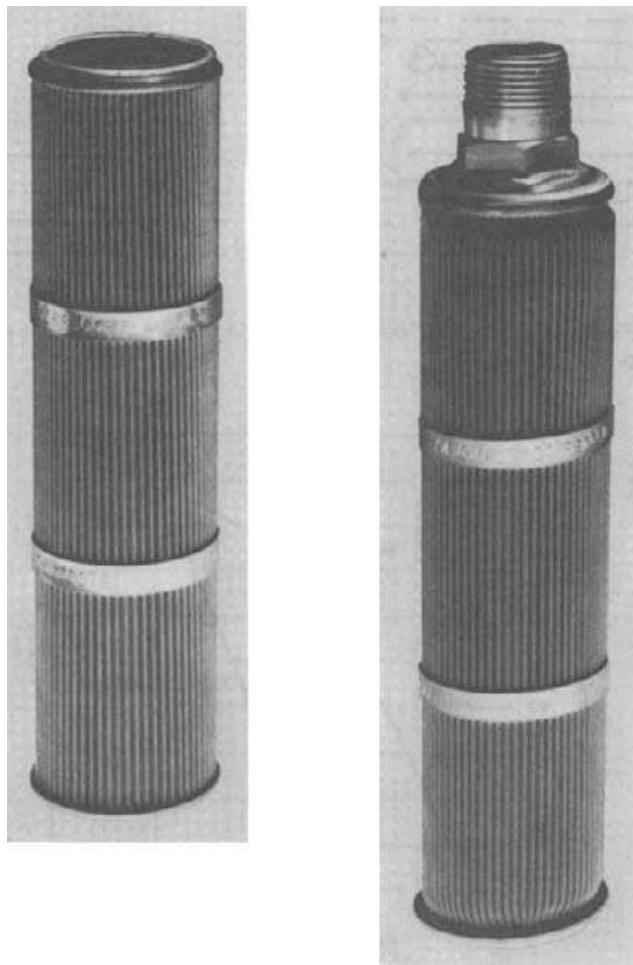


Figure 6-84 Porous sintered metal filter elements. (By permission from Pall Process Filtration Co.)

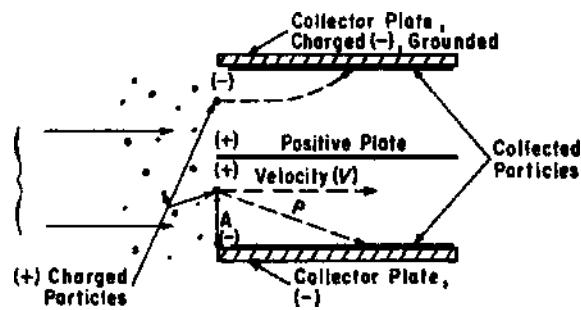


Figure 6-85 Charging particles in electrostatic precipitator. (By permission from adapted after A. Nutting, American Air Filter Co.)

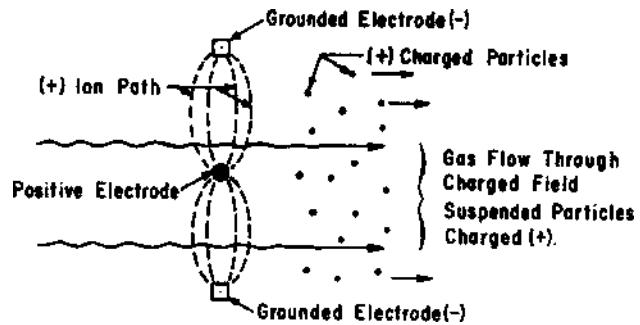


Figure 6-86 Particle collection. (By permission from Nutting, A., American Air Filter Co.)

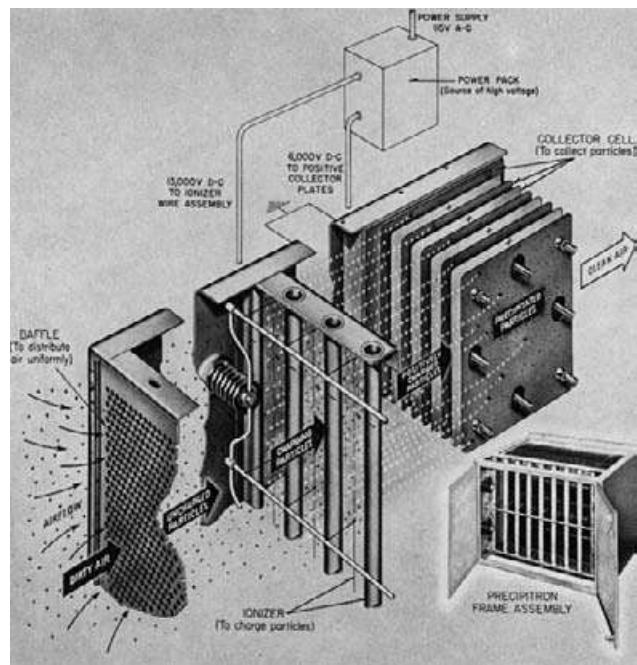


Figure 6-87 Electrical precipitator principle of operation. (Courtesy of Sturtevant Div. Westinghouse Electric Corp.)

**TABLE 6-18 Precipitator Operating Data for Common CPI Applications**

Precipitator Operating Data for Common CPI Applications	Ventilating				Grinding		Drying				Calcining				Miscellaneous													
	Asphalt Converter Saturator	Glass Melting	Pot Line	Carbon Plant	Cement Finish Grind	Jet Pulverized Catalyst	Cement Dryers	Bauxite Dryers	Gypsum Dryers	Copper Dryers	Other Drying Uses	Dry Process Cement	Wet Process Cement	Gypsum Kettles	Gypsum Calciners	Mg(OH) <sub>2</sub> Calciners	Nonferrous Metals Production	Sulfuric Acid	Phosphoric Acid	Shale Oil	Phosphorous Production	Acetylene	Waste Incinerator	Titanium Dioxide	Coal Pyrolysis	Pickle Liquor	Sulfur Cleaning	
Gas flow (1000 cfm)	5.0 to 24.8	7.0 to 10.0	94 to 120	15.5 to 27.0	10.0 to 13.5	10.0 to 13.5	40 to 80	100 to 150	11 to 150	39 to 45	20 to 187	62 to 305	157 to 346	3.0 to 42.9	7.6 to 82	25 to 135	27 to 115	3.6 to 600	15 to 102	13 to 43	12 to 15	10 to 37	42 to 46	32 to 270	37 to 45	83 to 90	32 to 36	15 to 20
Gas temperature (°F)	100 to 150	525 to 550	190 to 220	53 to 80	75 to 200	75 to 200	100 to 280	150 to 250	130 to 200	350 to 400	105 to 400	600 to 750	352 to 630	200 to 300	250 to 375	250 to 720	400 to 750	120 to 800	95 to 170	68 to 170	140	520 to 750	100	500 to 700	650	220	750	1292
Gas pressure (in. water)	-4 to 2	-6 to +6	Nega tive	Nega tive	-10 to +10	-10 to +10	-10 to +10	-10 to +10	Nega tive	-	-10 to +10	-4 to +4	-4 to +4	-6 to -2	-6 to -2	-10 to +10	Nega tive	-10 to +10	-36 to 0	-5 to +5	10	-0.3 to +0.3	50	-10 to +10	Nega tive	2 psig	-	
Gas moisture (volume %)	5	Ambi ent	2 to 3	Ambi ent	1 to 10	1 to 15	5 to 12	30 to 35	5 to 15	10 to 20	15 to 100	5.7 to 15.0	25 to 35	20 to 40	40 to 50	25 to 50	0.5 to 15	5 to 25	3 to 25	-	None	6	Various	27	-	26	21	
Inlet dust concentration (grs/ft <sup>3</sup> )	0.1 to 0.5	0.66 to 0.80	0.07 to 0.16	7.0 to 17.2	29 to 70	29 to 70	20 to 40	20 to 40	6.5 to 150	50 to 52	0.05 to 40	3.7 to 43.0	9.9 to 53.0	5.0 to 48.0	5.0 to 48.0	100 to 150	2 to 17	0.6 to 30	20 to 200	40 to 100	40	1.5 to 15.0	1.0	1.0 to 4.0	0.9 to 1.2	2.5	13	-
Outlet dust concentration (grs/ft <sup>3</sup> )	0.02 to 0.05	0.13 to 0.03	0.003 to 0.007	0.03 to 0.13	0.021 to 0.044	0.021 to 0.044	0.05 to 0.5	0.02 to 0.07	0.10 to 0.30	0.025 to 0.04	0.02 to 0.4	0.001 to 0.938	0.006 to 0.08	0.011 to 0.92	0.011 to 0.92	0.04 to 0.10	0.02 to 0.3	0.015 to 0.10	0.5 to 0.8	0.2 to 0.4	0.02 to 0.04	0.001 to 0.012	0.02 to 0.08	0.009 to 0.012	0.12	0.08	-	
Power input (kw)	3 to 7	7 to 12	10 to 14	3 to 4	4 to 6	4 to 6	4 to 12	15 to 20	7 to 12	21	7 to 24	2.4 to 51.0	20 to 94	5 to 10	14 to 36	13 to 30	7 to 30	8 to 40	8 to 16	14	15	14	14 to 50	14	20	12	14	
Collection efficiency (%)	90 to 98	98 to 98.5	86 to 97.6	98.5 to 99.8	95 to 99.96	95 to 99.96	90 to 99	99+	97 to 99.7	99.0 to 99.5	90 to 99	80 to 99.97	98.8 to 99.93	95 to 99.94	99 to 99.96	98 to 99.5	90 to 99.6	95 to 99.6	90 to 98	99.5	99	99.9	99	99.5	99	95	99	99

(Source: By permission from Sickels [71].)

## NOMENCLATURE

$a$	Specific surface area, $\text{ft}^2/\text{ft}^3$	$K'$	Constant for stationary vane separators, based on design
$a_e$	Acceleration due to gravity, $32.2 \text{ ft/s}^2$ or $9.8 \text{ m/s}^2$	$K_m$	Stokes-Cunningham correction factor, dimensionless
$A$	Area of segment of a circle, $\text{ft}^2$	$K_{me}$	Proportionality factor in Stokes-Cunningham correction factor, dimensionless
$A$	Cross-sectional flow area, $\text{ft}^2$	$k$	Constant for wire mesh separators
$A_b$	Cross-sectional area at the bottom of the vessel occupied by continuous aqueous phase, $\text{ft}^2$	$I$	Wire mesh thickness, $\text{ft}$
$A_c$	Cyclone inlet area = $B_e H_c$ for cyclone with rectangular inlet, $\text{ft}^2$	$L$	Length of the vessel from hydrocarbon inlet to hydrocarbon outlet, or length of decanter, $\text{ft}$
$A_I$	Area of interface, assumes flat horizontal, $\text{ft}^2$	$L_I$	Liquid entering Webre separator, lbs per minute per square foot of inlet pipe cross section
$A_H$	Cross-sectional area allocated to heavy phase, $\text{ft}^2$	$L_v$	Entrainment from Webre unit, lb liquid per minute per square foot of inlet pipe cross section
$A_L$	Cross-sectional area allocated to light phase, $\text{ft}^2$	$m$	Exponent given by equations
$A_p$	Area of particle projected on plane normal to direction of flow or motion, $\text{ft}^2$	$m_p$	Mass of particle, lb mass
$A_t$	Cross-sectional area at the top of the vessel occupied by continuous hydrocarbon phase, $\text{ft}^2$	$n$	Constant given in table
ACFS	Actual flow at conditions, $\text{ft}^3/\text{s}$	$N_{Re}$	Reynolds number, dimensionless (use $(Re)$ consistent units)
$b_1$	Constant given in table	$N_t$	Number of turns made by gas stream in a cyclone separator
$c$	Volume fraction solids	$\Delta P$	Pressure drop, $\text{lb/in.}^2$
$C$	Overall drag coefficient, dimensionless	$\Delta p$	Pressure drop, in. water
$D$	Diameter of vessel, $\text{ft}$	$\Delta p_D$	Pressure drop, no entrainment, in. water
$D_b$	See $D_p$ , min	$\Delta p_L$	Pressure drop due to liquid load, in. water
$D_c$	Cyclone diameter, $\text{ft}$	$\Delta p_T$	Pressure drop, total across wet pad, in. water
$D_e$	Cyclone gas exit duct diameter, $\text{ft}$	$Q_D$	Dispersed phase, volumetric flow rate, $\text{ft}^3/\text{s}$
$D_H$	Hydraulic diameter, $\text{ft} = 4$ (flow area for phase in question/wetted perimeter); also, $D_H$ in decanter design represents diameter for heavy phase, $\text{ft}$	$Q_H$	Volumetric flow rate, heavy phase, $\text{ft}^3/\text{s}$
$D_L$	Diameter for light phase, $\text{ft}$	$Q_L$	Volumetric flow rate, light phase, $\text{ft}^3/\text{s}$
$D_p$	Diameter of particle, $\text{ft}$ , or equivalent diameter of spherical particle, $\text{ft}$	$r$	Vessel radius, $\text{ft}$
$D_{p-\min}$	Minimum diameter of particle that is completely collected, $\text{ft}$	$\text{sp gr}$	Specific gravity of continuous phase at flow conditions
$D'_p$	Diameter of particle, in. or mm	$\text{sp gr}_p$	Specific gravity of settling particle at flow conditions
$d$	Droplet diameter, $\text{ft}$	$\Delta \text{sp gr}$	Difference in specific gravity of the particle and the surrounding fluid
$f$	Factor relating average velocity to maximum velocity	$t_{\text{avg}}$	Average residence time based on liquid flow rate and vessel volume, min
$f_c$	Friction factor, dimensionless	$t_{\min}$	Minimum residence time to allow particles to settle based on Stokes' Law, min
$F$	Flow rate of one phase, gpm	$u$	Relative velocity between particle and main body of fluid, $\text{ft/s}$
$F_{aq}$	Aqueous phase flow rate, gpm	$u_t$	Terminal settling velocity determined by Stokes' Law, of particle under action of gravity, $\text{ft/s}$
$F_{cv}$	Cyclone friction loss, expressed as number of Cyclone inlet velocity heads, based on $A_c$	$u_{ts}$	Terminal settling velocity as calculated from Stokes' Law, $\text{ft/s}$
$F_d$	Drag or resistance to motion of body in fluid, pounds	$v = V_t$	Terminal settling velocity, in./min
$F_{hc}$	Hydrocarbon phase flow rate, gpm	$v_a$	Average velocity of gas, $\text{ft/s}$
$F_t$	Total flow rate of both phases, gpm	$v_{aq}$	Terminal settling velocity of hydrocarbon droplets in aqueous phase in the bottom of the vessel, in./min
$g = g_c = g_L$	Acceleration due to gravity, $32.17 \text{ ft/s}^2$	$v_c$	Velocity down flow channel for continuous phase, $\text{ft/s}$
$h$	Distance from center to given chord of a vessel, $\text{ft}$	$v_d$	Terminal settling velocity of a droplet, $\text{ft/s}$
$h_b$	Height of continuous aqueous phase in the bottom of the vessel, in.	$v_{hc}$	Terminal settling velocity of aqueous droplets in hydrocarbon phase in the top of the vessel, in./min
$h_c$	Height of a segment of a circle, in.	$v_t$	Terminal settling velocity of particle under action of gravity, $\text{ft/s}$
$h_t$	Height of continuous hydrocarbon phase in the top of the vessel, in.	$v_{ts}$	Terminal settling velocity of particle as calculated from Stokes' Law, $\text{ft/s}$
$h_{vi}$	Cyclone inlet velocity head, in. water	$V$	Velocity of gas or vapor entering, $\text{ft/min}$
$H$	Height of a segment of a circle, $\text{ft}$		
$H_c$	Height of rectangular cyclone inlet duct, $\text{ft}$		
$H_D$	Height of dispersion band, $\text{ft}$		
$I$	Width of interface, $\text{ft}$		
$k = K$	Empirical proportionally constant for cyclone pressure drop or friction loss, dimensionless		

$V_{(\text{separator})}$	Separator vapor velocity evaluated for the gas or vapor at flowing conditions, ft/s
$V'$	Vapor velocity entering unit, lb per minute per square foot of inlet pipe cross section
$V_a$	Maximum allowable vapor velocity across inlet face of mesh calculated by relation, ft/s
$V_{\text{act}}$	Actual operating superficial gas velocity, ft/s or ft/min, for wire mesh pad
$V_D$	Design vapor velocity (or selected design value), ft/s
$V_c$	Cyclone inlet velocity, average, based on area $A'_c$ ft/s
$V_{\max}$	Calculated maximum allowable superficial gas velocity, ft/s, or ft/min wire mesh pad
$V_s$	Superficial gas velocity, ft/s
$V_{\text{sa}}$	Separator vapor velocity evaluated for air–water system, ft/s
$V_{\text{set}}$	Active volume of settler occupied by one of the phases, $\text{ft}^3$
$V_t$	Settling velocity for single spherical particle, ft/s or m/s
$V_{ts}$	Settling velocity for hindered uniform spherical particle, ft/s or m/s
$W_i$	Width of rectangular cone inlet duct, ft
$z_h$	Heavy phase outlet dimensions of decanter measured from horizontal bottom
$z_i$	Interface of decanter liquids measured from bottom
$z_l$	Light phase outlet measured from the bottom of the decanter

## SUBSCRIPTS

L or 1	Light phase
H or h	Heavy phase
C or c	Continuous phase
D or d	Dispersed phase
l	Liquid
v	vapor or gas

## GREEK SYMBOLS

$\varepsilon$	Void fraction of wire mesh, dimensionless
$\eta$	Fraction of dispersoid in swept volume collected on target
$\theta$	Factor for establishing type of flow for decanters [25]
$\mu$	Viscosity of surrounding fluid, cP, except where it is lb/ft s
$\mu_c$	Viscosity of continuous phase, lb/ft s
$\mu_H$	Viscosity of heavy phase, lb/ft s
$\mu_v$	Viscosity of fluid, cP
$\mu_L$	Viscosity of light phase, lb/ft s
$\mu$	Fluid viscosity, (lbmass)/ft s = cP/1488
$\mu_m$	Milli-micron = 0.001 mm
$\pi$	3.1416
$\rho$	Fluid density, lb mass/ft <sup>3</sup>
$\rho_c$	Density of fluid for continuous phase, lb/ft <sup>3</sup>
$\rho_f$	Density of fluid, lb/ft <sup>3</sup> or kg/m <sup>3</sup>
$\rho_L$	Liquid density, lb/ft <sup>3</sup>
$\rho_d$	Density of fluid in the droplet, lb/ft <sup>3</sup>
$\rho_L$	Density of light phase fluid, lb/ft <sup>3</sup>
$\rho_p$	Density of particle, lb/ft <sup>3</sup>
$\rho_s$	True density of particle, lb mass/ft <sup>3</sup>
$\rho_v$	Vapor density, lb/ft <sup>3</sup>

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## MIXING OF LIQUIDS

**M**any operations in the chemical process industries (CPI) are dependent to a great extent on effective mixing of fluids. While mixing refers to any operation used to change a non-uniform system into a uniform one (i.e., the random distribution into and through one another, of two or more initially separated phases), agitation implies forcing a fluid by mechanical means to flow in a circulatory or other pattern inside a vessel. Mixing is an integral part of a chemical or physical process such as blending, dissolving, dispersion, suspension, emulsification, heat transfer, and chemical reactions. Mixing of fluids is necessary in many chemical processes. It may include mixing of liquid with liquid, gas with liquid, or solids with liquid. Agitation of these fluid masses does not necessarily imply any significant amount of actual intimate and homogeneous distribution of the fluids or particles, and for this reason mixing requires a definition of degree and/or purpose to properly define the desired state of the system. Table 7-1 summarizes the principal purposes for agitating fluids.

In order for the mixing operation to accomplish the overall process requirement, it is necessary to establish the factors which are significant for a mixing device that provides the required end result for the industrial application. It is impractical for the design engineer to completely design a mixer, that is, define its type, diameter, operating speed, shape, and impeller type. Rather, it is reasonable for the engineer to understand the mechanical and processing essentials, and anticipated performance when dealing technically with a mixing equipment representative. For standard nomenclature, see [1, 2]. The technical performance and economics of various designs often need to be examined in order to make a good, cost-effective selection of the device in a process application. In some situations, particularly chemical reaction and/or mass transfer, it may be necessary to conduct test work to develop a sound basis for a larger-scale industrial unit. In other cases, the needed data may be drawn from the public technical literature or a manufacturer's application files (see [1-21]).

**TABLE 7-1 Characteristics for Agitating Fluids**

1. Blending of two miscible (e.g., NaCl and NaOH) or immiscible liquids (e.g., ethyl alcohol and water).
2. Dissolving solids in liquids (e.g., salt in water).
3. Dispersing a gas in a liquid as fine bubbles (e.g., oxygen from air in a suspension of microorganism for fermentation, or for the activated sludge process in waste treatment).
4. Agitation of the fluid to increase heat transfer between the fluid and a coil or jacket in the vessel wall.
5. Suspension of fine solid particles in a liquid, such as in the catalytic hydrogenation of a liquid where solid catalyst and hydrogen bubbles are dispersed in the liquid.

Mixer performance is often related in terms of the fluid velocity during agitation, total pumping capacity (flow of the fluid in the system) generated by one impeller, and the total flow in the tank (or sometimes as blending time or a solids-suspension criterion) [10]. Mixing applications often include one or more of the following [11]:

- bulk mixing
- chemical reaction
- heat transfer
- mass transfer
- phase interaction (suspending/dispersing).

Mixing is accomplished by the rotating action of an impeller in the continuous fluid. This action shears the fluid, setting up eddies which move through the body of the system. Generally, the fluid motion involves (a) the mass of the fluid over large distances and (b) the small-scale eddy motion or turbulence which moves the fluid over short distances [22, 23].

The size and shape of the vessel to be used for the mixing operation is important in achieving the desired mixing results; therefore, this aspect of the design must accompany the actual mechanical mixer design/size selection. Figure 7-1 shows typical mixing equipment.

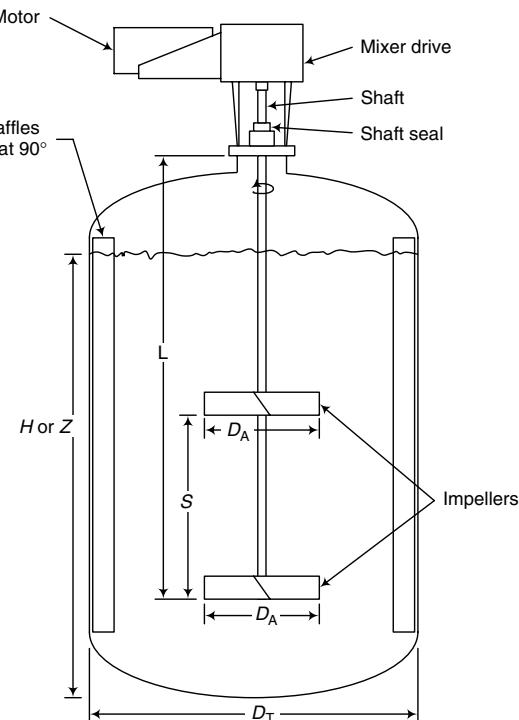


Figure 7-1 Impeller mixing equipment.

The performance of mixers involves high volume or flow operations, or high head or shear operations. Many mixing processes utilize a combination of these two; however, there are many which can have only high volume or only high head. Some operations listed in decreasing order of high volume requirements include: blending, heat transfer, solids suspension, solids dissolving, gas dispersion, liquid-liquid dispersion (immiscible), and solid dispersion (high viscosity).

Impeller types usually used with mixing and listed in decreasing order of high volume ability (hence in increasing order of high head ability or requirement) are: paddle, turbine, propeller, sawtooth impeller or cut-out impeller disc (no blades), and colloid mill. Recently, high performance impeller types with better flow patterns have been employed to enhance mixing effectiveness and operating below critical speed with the ability to handle demanding high fluid force mixing conditions. Some benefits are true improvements in pumping efficiency while other advantages involve better application of basic mixing technology. Added features are impeller's reduced weight, robust shaft design, and reduced operating cost.

Figures 7-2 and 7-3 are useful as guides in the general selection of mixing impellers and associated vessels. Note that the shape relationships of Figure 7-3 are applicable only to turbine-type impellers.

An example of the use of the chart occurs in the leaching of 50% water slurry of a 20-mesh, 3.8-gravity ore by dilute acid of equal volume and the heat of solution to be removed by cooling coils. The controlling factor is suspension of the solids to promote the reaction in which heat is developed. The criteria for solid suspension are circulation and liquid velocity sufficient to overcome the settling rate of the solids. The same criteria are also pertinent to good heat transfer and reaction. The large particle size and gravity difference between solids and solution suggest fast settlement. Best impeller position is therefore on the vessel bottom so that its radial discharge will sweep all solids up into the tank. In order to maintain maximum distribution of solids yet allow sufficient depth of liquid for the cooling coils, the maximum tank diameter-height ratio of 1:1 from the chart would be used. Impeller ratio is regulated by reaction and suspension, with the latter controlling because of particle size. Tank depth and particle size in this case suggest a large impeller diameter, or a ratio of about 2.5:1. As the circulation pattern now established is radially across the bottom and up the sides, the slurry will flow up across and through a helical coil for good transfer rate. This pattern will be assured by four full vertical baffles mounted inside the cooling coil [24]. With this size information and reference to the horsepower charts, the preliminary design is complete.

The three velocity components of flow created by the impeller discharge are radial, axial, and rotational flow. These velocity components relate directly to cylindrical coordinates and are appropriate because many agitated tanks are vertical cylinders with the mixer shaft on the central axis (Figure 7-1). Assuming the cylindrical axis is oriented vertically, these components are briefly described as [25] follows.

Mixing Device	Viscosity, Centipoises				Plastic State	Solid State
	$10^2$	$10^3$	$10^4$	$10^5$		
Air Agitation	—	—	—	—		
Liquid Jets	—	—	—	—		
Paddles	—	—	—	—		
Propellers	—	—	—	—		
Turbines	—	—	—	—		
Cones	—	—	—	—		
Disk	—	—	—	—		
Screws	—	—	—	—		
Barrels	—	—	—	—		
Ball Mills	—	—	—	—		
Ribbons	—	—	—	—		
Kneaders	—	—	—	—		
Colloid Mills	—	—	—	—		
Special Mills	—	—	—	—		
Mullers	—	—	—	—		
Pug Mills	—	—	—	—		
Internal Mixers	—	—	—	—		
Roll Mills	—	—	—	—		
Conical Mills	—	—	—	—		
Pen Mills	—	—	—	—		
Impact Wheels	—	—	—	—		

Key = — Batch Process  
— Continuous Process

Figure 7-2 Range of operation of mixers. (By permission from Quillen, C.S., Chem. Eng., Jun 1954, p. 177 [22].)

Selection Chart				Shape Relationships for Turbine Designs		
Service	Mixing Device	Range	Criteria	Tank Diameter In Impeller Diameter Ratio	Tank Height To Diameter Ratio	Impeller Position
			1. Volume Circulation	3:1 to 6:1	Unrestricted	Single or Multiple
Blending	Turbine	—	1. Drop Size Control 2. Re-Circulation	3.0:1 to 3.5:1	1:1 to 3:1	After Bottom Center Line of Liquid Charge
	Propeller	—		2.5:1 to 3.5:1	1:1 to 3:1	
	Paddle	—		2.5:1 to 3.5:1	1:1 to 3:1	
	Tank Vol.	→ 1,000,000 gal		2.5:1 to 3.5:1	1:1 to 3:1	
Dispersion (Immiscible Systems)	Turbine	—	1. Intensity 2. Volume Circulation	1.6:1 to 3.2:1	1:2 to 2:1	After Bottom Center Line of Liquid Charge
	Propeller	—		1.6:1 to 3.2:1	1:2 to 2:1	
	Paddle	—		1.6:1 to 3.2:1	1:2 to 2:1	
	Flow	→ 1,000 gal/Min.		1.6:1 to 3.2:1	1:2 to 2:1	
Reactions in Solution (Miscible Systems)	Turbine	—	1. Intensity 2. Volume Circulation	2.0:1 to 3.5:1	1:1 to 1:2	Single or Multiple
	Propeller	—		2.0:1 to 3.5:1	1:1 to 1:2	
	Paddle	—		2.0:1 to 3.5:1	1:1 to 1:2	
	Charge Vol.	→ 20,000 gal		2.0:1 to 3.5:1	1:1 to 1:2	
Dissolution	Turbine	—	1. Shear 2. Volume Circulation	2.0:1 to 3.5:1	1:1 to 1:2	Depending on Particle Size, Location Off Bottom, and Residence Length (see)
	Propeller	—		2.0:1 to 3.5:1	1:1 to 1:2	
	Paddle	—		2.0:1 to 3.5:1	1:1 to 1:2	
	Charge Vol.	→ 10,000 gal		2.0:1 to 3.5:1	1:1 to 1:2	
Solids Suspension	Turbine	—	1. Circulation 2. Velocity	2.0:1 to 3.5:1	1:1 to 1:2	Depending on Particle Size, Location Off Bottom, and Residence Length (see)
	Propeller	—		2.0:1 to 3.5:1	1:1 to 1:2	
	Paddle	—		2.0:1 to 3.5:1	1:1 to 1:2	
	% Solids	→ 100%		2.0:1 to 3.5:1	1:1 to 1:2	
Gas Applications	Turbine	—	1. Controlled Shear 2. Circulation 3. High Velocity	2.5:1 to 4.0:1	4:1 to 1:1	Multiple-Layered One Impeller Diameter Off Bottom, 2. Surface, and Reside Length (see)
	Propeller	—		2.5:1 to 4.0:1	4:1 to 1:1	
	Paddle	—		2.5:1 to 4.0:1	4:1 to 1:1	
	Gas Vol.	→ 5,000 ft <sup>3</sup> /min.		2.5:1 to 4.0:1	4:1 to 1:1	
High Viscosity Applications	Turbine	—	1. Volume Circulation 2. Low Velocity	1.5:1 to 2.5:1	1:2 to 2:1	Single or Multiple
	Propeller	—		1.5:1 to 2.5:1	1:2 to 2:1	
	Paddle	—		1.5:1 to 2.5:1	1:2 to 2:1	
	Vis.	→ 1,000,000 cP		1.5:1 to 2.5:1	1:2 to 2:1	
Heat Transfer	Turbine	—	1. Volume Circulation 2. High Veloc. 3. Axial Transf. Per Surface	Related to Other Services	Depends on Other Services Being Performed	Simple or Multiple Impeller, Operable Transf. for Surface when Cooling
	Propeller	—		Related to Other Services	Depends on Other Services Being Performed	
	Paddle	—		Related to Other Services	Depends on Other Services Being Performed	
	Charge Vol.	→ 20,000 gal		Related to Other Services	Depends on Other Services Being Performed	
Crystallization or Precipitation	Turbine	—	1. Circulation 2. Low Velocity 3. Shear Control	2.0:1 to 3.2:1	2:1 to 1:1	Single, After Bottom Center Line of Liquid Charge
	Propeller	—		2.0:1 to 3.2:1	2:1 to 1:1	
	Paddle	—		2.0:1 to 3.2:1	2:1 to 1:1	
	Charge Vol.	→ 20,000 gal		2.0:1 to 3.2:1	2:1 to 1:1	

Figure 7-3 General selection chart for mixing. (By permission from Lyons, E.J., and N.H. Parker, Chem. Eng. Prog., Vol. 50, 1954, p. 629 [24].)

**Radial flow** refers to the impeller discharge direction, which is horizontally (radially) outward from the vertical axis of the tank and impeller.

**Axial flow** is the vertical flow which is parallel to the axis of the cylinder. Most axial flow impellers pump downward, creating downward flow at the center of the tank and upward flow at the walls.

**Rotational flow** is the velocity component that follows the direction of rotation for the impeller. All rotational impellers create a rotational flow component.

All styles and designs of mixing impellers produce either an axial flow or a radial flow of the fluid during the impeller rotation. There are, of course, degrees of variation of each of these patterns, which then become a part of the selection and specifying process to achieve the mixing objective. Any single component alone will produce very poor mixing, especially in the case of rotational flow.

Axial flow impellers in an unbaffled tank will produce vortex swirling about the vertical shaft. This will be discussed later in detail.

## 7.1 MECHANICAL COMPONENTS

Figure 7-4 highlights the most commonly used radial and axial flow impeller styles for process applications. Figures 7-5 and 7-6 cover the styles/designs that are used for special specific applications to accomplish the mixing objectives, and Figure 7-4 covers most of the common applications. Fluid flow patterns for the axial and radial flows are shown in Figures 7-5a and b.

## 7.2 IMPELLERS

### General Types

#### A. Propeller, marine type

1. Circulates by axial flow parallel to the shaft and its flow pattern is modified by baffles, normally a downward flow
2. Operates over wide speed range
3. Can be pitched at various angles, most common is the three blades on square pitch (pitch equal to diameter)
4. Shearing action very good at high speed, but not generally used for this purpose
5. At low speed it is not easily destroyed
6. Economical on power
7. Generally self-cleaning
8. Relatively difficult to locate in vessels to obtain optimum performance
9. Not effective in viscous liquids, unless special design
10. Cost: moderate.

#### B. Open turbine: Radial

1. Circulates by radially directed centrifugal force using turbine blades. Circulation good for tank extremes; less danger of fluid short-circuiting in tank
2. Generally limited to a maximum speed, range may be narrow for some services
3. Used for fairly high shear and turbulence
4. Better than axial unit for tanks with cone bottom of greater than 15° angle, to lift material from bottom of cone and mix with bulk of liquid
5. Effective in high viscosity systems
6. Generally requires slower speeds and hence greater gear reduction than propeller, higher power per unit volume
7. Cost: moderate.

#### C. Open turbine: Axial four blades [10], most common applications

1. Four-bladed 45° pitched blade, blade width is function of diameter
2. Made in wide range of sizes for top-entering mixers from 1 to 500 motor hp, and in diameters of 18–120 in.

3. Primarily for flow-controlled requirements, such as solids suspensions, heat transfer, and other high pumping efficiency applications

4. Preferred for shallow tanks of low Z/T with low liquid cover over turbine of 1.20% of turbine diameter
5. Cost: moderate.

#### D. Open turbine: Bar turbine, radial flow

1. Produces highest shear rate of any basic impeller. Runs at high speed, uses lower torque
2. Blade width and height normally 1/20 of impeller diameter.

#### E. Open turbine: Axial, three blades [10]

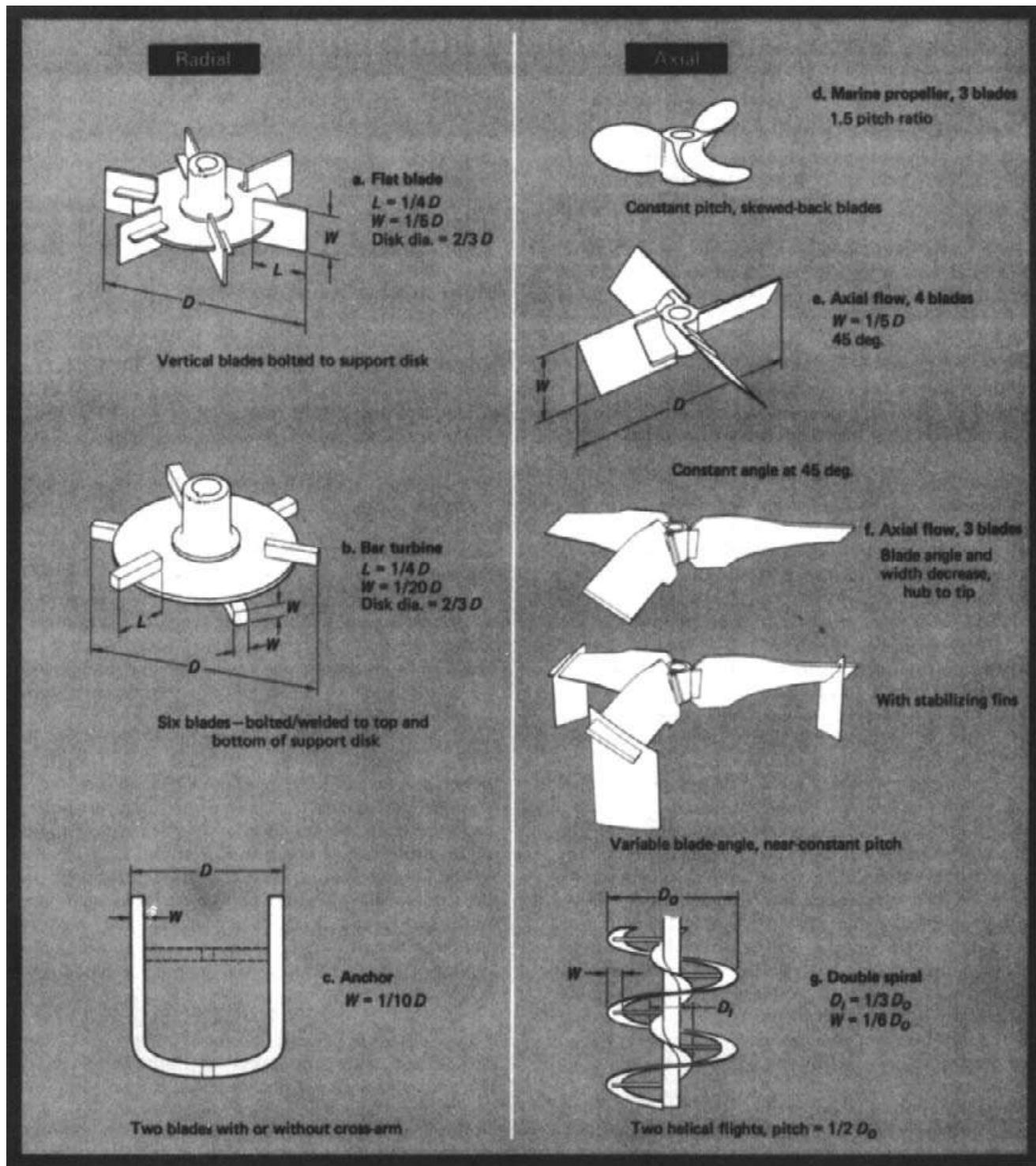
1. Provides more flow and less shear than the four-bladed design
2. Produces nearly constant, uniform velocity across entire discharge area; has nearly constant pitch ratio
3. Close to hub the blade angle is steeper, and blade is wider than at tip
4. Size ranges from 20 to 120 in. diameter for motors of 1–500 hp. Impeller speeds range from 56 to 125 rpm
5. Large flow-directing stabilizer fins improve pumping capacity for viscosity ranges from 500 to 1500 cP. Performs much like a marine propeller, but does not have some of its disadvantages
6. Cost: moderate, less than marine propeller.

#### F. Double-spiral impeller [10]

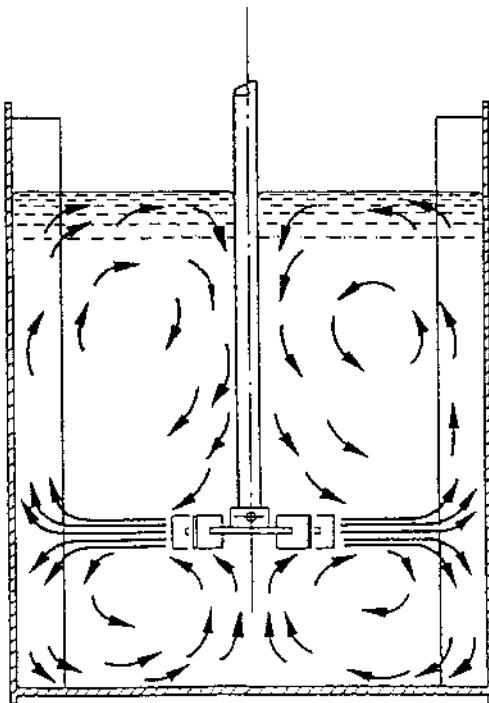
1. Used for viscous materials; has inner and outer flights
2. The inner flights pump down, and the outer flights pump upward
3. Diameter of inner flight is one-third the impeller diameter
4. Width of outer ribbon is one-sixth diameter of impeller
5. Impeller height is equal to its diameter
6. Available in 20–120 in. diameter for 1–250 motor hp, and speeds of 5.5–45 mph
7. Cost: moderate.

#### G. Shrouded turbine

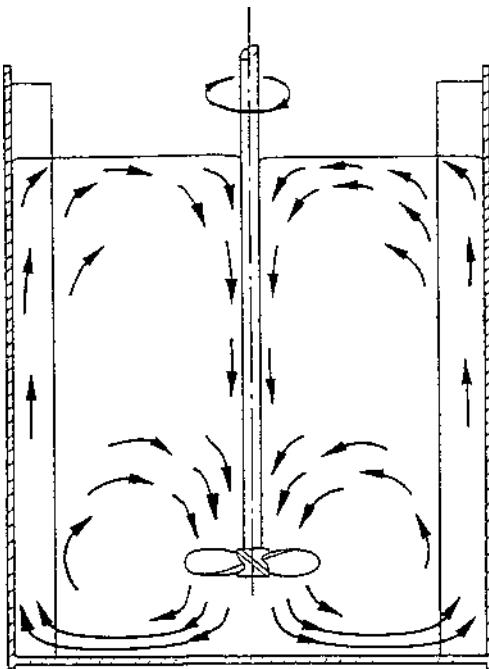
1. Circulates by radially directed centrifugal force using enclosed impeller stators. Circulation very good
2. Speed range may be limited
3. Not easily destroyed at reasonable speeds
4. Not self-cleaning; fouls and plugs relatively easily
5. Flow capacity limited, relatively low
6. Effective in high viscosity systems
7. Cost: relatively high.



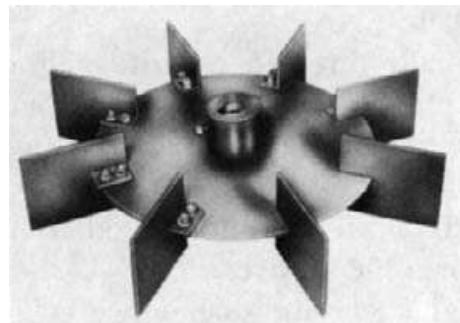
**Figure 7-4** Impeller styles and general sizes commonly in use in process industry plant. (By permission from Oldshue, J.Y., "Fluid Mixing Technology and Practice", *Chem. Eng.*, Jun 13, 1983, p. 84 [10].)



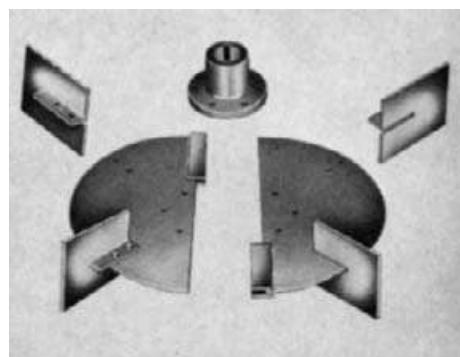
**Figure 7-5a** Radial flow pattern produced by a flat-blade turbine. (Source: Holland, F.A. and R. Bragg, *Fluid Flow for Chemical Engineers*, 2nd ed., Edward Arnold, 1995.)



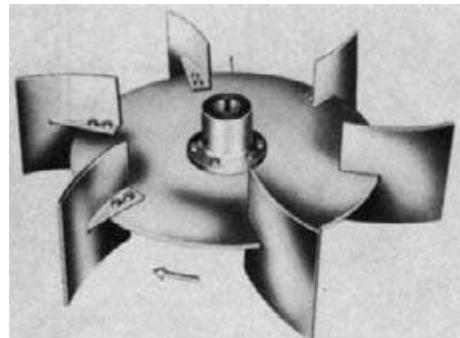
**Figure 7-5b** Axial flow pattern produced by a marine propeller. (Source: Holland, F.A. and R. Bragg, *Fluid Flow for Chemical Engineers*, 2nd ed., Edward Arnold, 1995.)



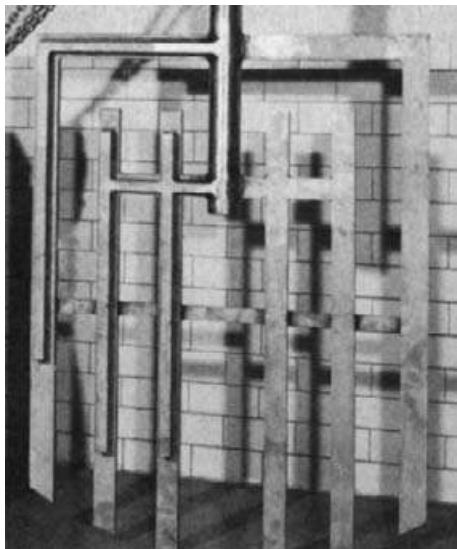
**Figure 7-6a** A flat-blade turbine can handle the majority of all fluid mixing applications when correctly applied. Its high pumping capacity makes it preferable for general mixing operations. It is well adapted to the applications of protective coverings, such as lead, rubber, and plastics. (Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



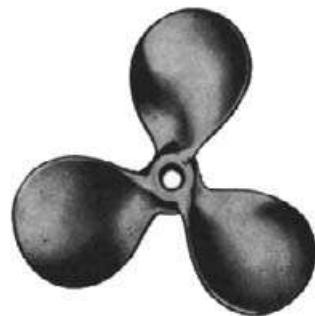
**Figure 7-6b** Small openings are no problem with turbines. With blades removed, a turbine can pass through openings about  $\frac{2}{3}$  as large as the assembled impeller or impeller can be split, as above, to pass through openings  $\frac{1}{3}$  of turbine diameter. (Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



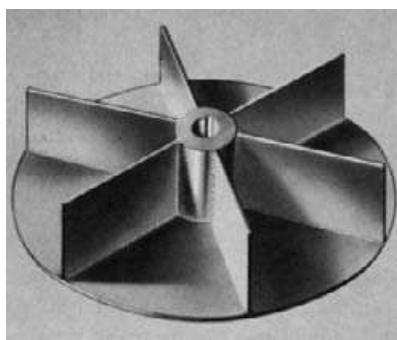
**Figure 7-6c** Curved-blade turbine creates a dual suction flow pattern the same as the flat blade. This design is used when relatively low shear is a requirement, when abrasion must be considered, and when many other variables are of prime importance. (Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



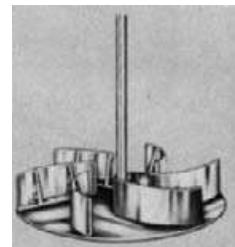
**Figure 7-6d** Gate paddle impeller is designed for materials of high viscosity and operates at low shaft speeds. It is most desirable for shallow, wide tanks and wherever low shear is a requirement. (Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



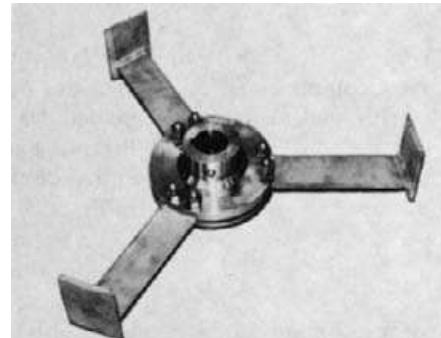
**Figure 7-6e** Marine propeller is designed with extra section thickness to give longest life in corrosion or abrasive materials. It is polished to a high finish and accurately balanced. Many special propeller types and alloys are available. This propeller type is satisfactory in 95% of applications. It drives liquid ahead in a helical cone while doing considerable "work" on material passing through it. (Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



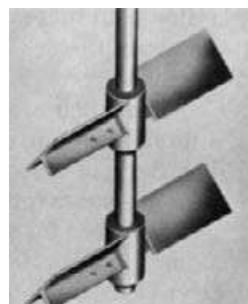
**Figure 7-6f** Lifter turbine is efficient for pumping large volumes against static heads of less than 36 in. As shown, it is used below a draft tube. Inverted, it is used above an orifice plane in tank bottom. (Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



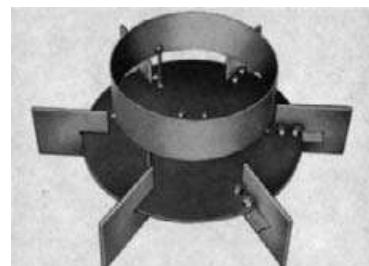
**Figure 7-6g** Curved-blade turbine, developed especially for agitating fibrous materials such as paper stock. Also used on oil well drilling muds. This impeller gives fast, thorough turnover without need for the usual tank baffling. (Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



**Figure 7-6h** A typical radial impeller agitator. Operates as an agitating turbine or a conventional propeller with a wide range of applications. (Courtesy of Struthers-Wells Corp., Warren, PA.)



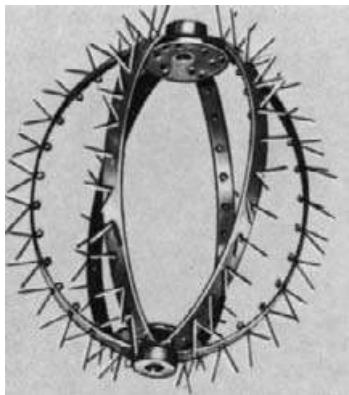
**Figure 7-6i** Flat-blade pitched paddle. A simple, low cost design that handles a wide variety of jobs. Operating at low speeds, it gives maximum pumping capacity with a minimum of turbulence. (Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



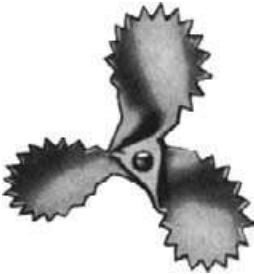
**Figure 7-6j** Bottom of flat-blade turbine with stabilizing ring to prevent shaft whip. (Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



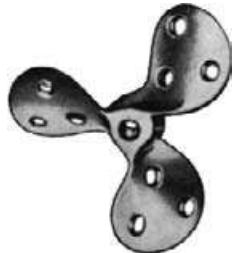
**Figure 7-6k** Plain cage beater. Imparts a cutting and beating action. It is usually combined with a standard propeller which supplies movement in the mix. (Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



**Figure 7-61** Studded cage beater. Enormous contact area gives extremely violent cutting and shredding action to certain emulsions, pulps, and so on. (Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



**Figure 7-6m** Saw-toothed propeller. Displaces a large amount of liquid and combines a cutting and tearing action. Suitable for fibrous materials. (Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



**Figure 7-6n** Perforated propeller. Occasionally recommended for wetting dry powders, especially those that tend to form into lumps. (Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



**Figure 7-60** Folding propeller. May be passed through a very small opening. Blades assume working position through centrifugal force only while rotating. (Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



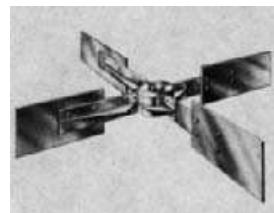
**Figure 7-6p** Propeller with ring guard. For extra safety where sounding rods are used or where samples are taken by hand dipping. (Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



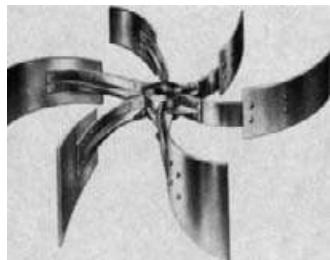
**Figure 7-6q** Weedless propeller. Handles long fibrous materials that would become entangled in the ordinary propeller. (Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



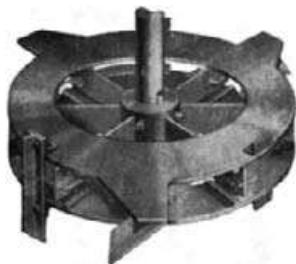
**Figure 7-6r** Cut-out propeller. Displaces a small amount of liquid combined with a high rate of shear for shredding, breaking up pulps, and so on. (Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



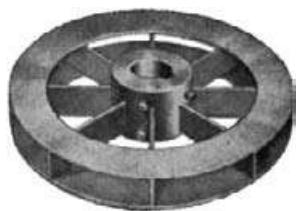
**Figure 7-6s** Four-blade, vertical flat-blade turbine impeller. Very versatile, mostly used in wide application range. (Courtesy of Philadelphia Gear Corp.)



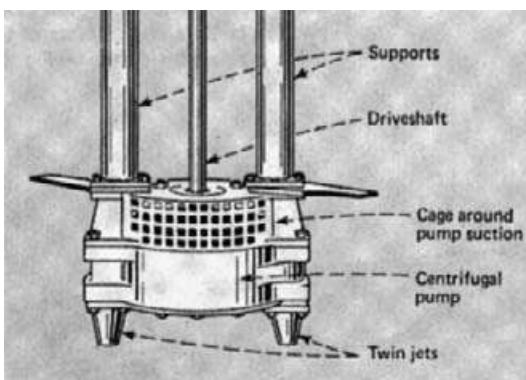
**Figure 7-6t** Standard six-blade vertical curve blade turbine impeller. Gives good efficiency per unit of horsepower for suspensions, mixing fibrous materials. Gives high pumping capacity. (Courtesy of Philadelphia Gear Corp.)



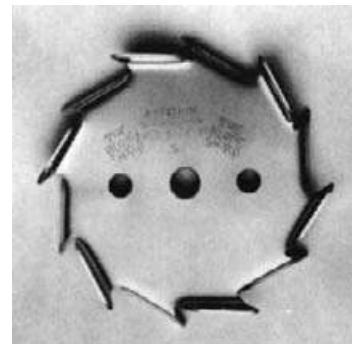
**Figure 7-6u** Shrouded turbine for high pumping capacity. Usually used with low static heads, creates minimum of direct shear. (Courtesy of International Process Equipment Co., Div. of Patterson Foundry and Machine Co.)



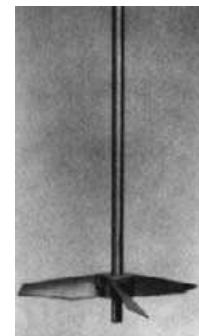
**Figure 7-6v** Turbine with three, four, or six radial blades. Handles a wide range of applications. (Courtesy of International Process Equipment Co., Div. of Patterson Foundry and Machine Co.)



**Figure 7-6w** Jet-flow mixer. Twin flow jets from submerged centrifugal pump allow for a maximum hydraulic shear per unit of power input, high velocities useful for thick slurries. (By permission from Penny W.R., *Chem. Eng.*, Mar 22, 1971, p. 97 [26].)



**Figure 7-6x** Type R-500. Very high shear radial flow impeller for particle size reduction and uniform dispersion in liquids. (By permission from Lightnin (Formerly Mixing Equipment Co.), a unit of General Signal.)



**Figure 7-6y** A-310 impeller. Develops 50% more action than ordinary propellers and is geometrically similar for accurate scale-up. (By permission from Lightnin (Formerly Mixing Equipment Co.), a unit of General Signal.)



**Figure 7-6z** A-410 Composite impeller. Strong axial flow at very high flow efficiency. Operates through a wide range of viscosities. (By permission from Lightnin (Formerly Mixing Equipment Co.), a unit of General Signal.)

## H. Paddle

1. Circulates radially, but has no vertical circulation unless baffles used
2. Covers wide viscosity range, blending
3. Not easily destroyed in operation
4. Not easily fouled
5. Flow capacity can be high for multiple blades
6. Cost: relatively low.

## I. Anchor, two blades, contoured [10]

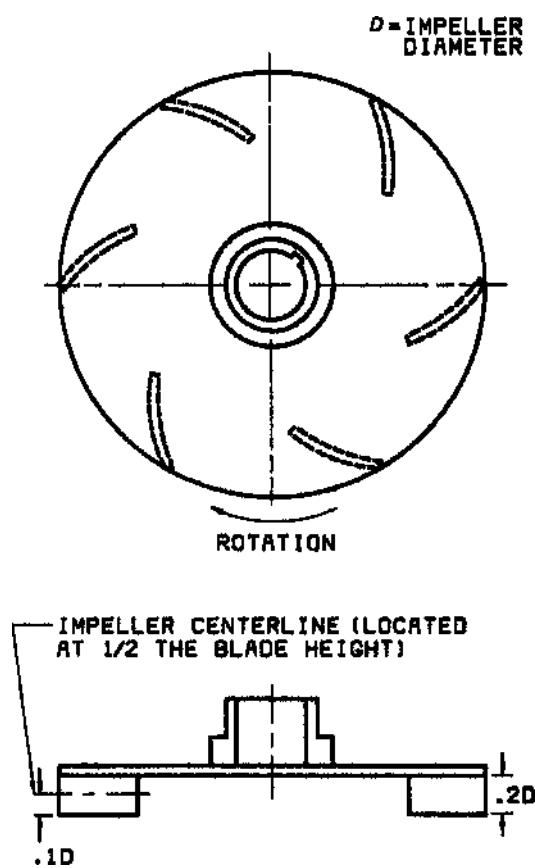
1. For higher viscosity applications: 40,000–50,000 cP
2. Nominal blade width is  $D/10$ , with little power change from  $D/8$  to  $D/12$  ( $D$  = impeller diameter)

3. Power requirements vary directly with the impeller height to diameter ratio
4. Used for blending and heat transfer for viscosities between 5000 and 50,000 cP. Pumping capacity falls off above 50,000 cP, as it "bores a hole" in the fluid. Speed ranges from 5.5 to 45 mph, for motor of 1–150 hp and impeller diameter 24–120 in.

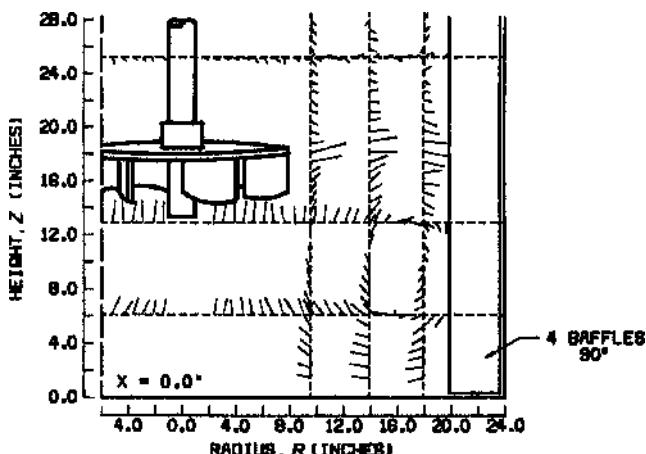
#### J. Lifter Turbine, Figures 7-7a and b.

This unit type [14] is used for a combination of pumping and mixing purposes. The unit has a closed disk on the top side. The feed flows into the unit and comes from directly below the rotating impeller. The performance is dependent on the size of the unit and the physical location with respect to the distance up from the bottom of the vessel. As this clearance increases, the head decreases for constant flow, and increases the power requirement.

Figures 7-4 and 7-6 illustrate a few of the impeller types used for mixing. They may be basically classified as axial, radial, and mixed. Generally, the most applicable are the three-bladed propeller, the flat-blade turbine, the curved-blade turbine, and the paddle. The many other designs are either modifications of these or specially designed for a very special purpose with respect to a fluid system and/or its performance. Figure 7-8 shows a class of axial flow impellers known as hydrofoil impellers. These impeller types have three or four blades made of rolled or bent metal plate. The blades are bolted or welded to a central hub that mounts on the

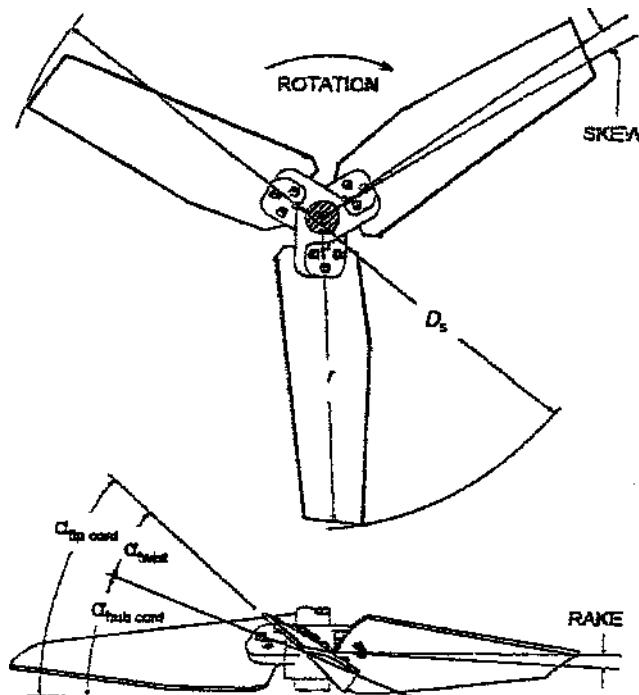


**Figure 7-7a** Drawing of typical lifter turbine. (By permission from Oldshue, J.Y., "Fluid Mixing Technology", *Chem. Eng.*, McGraw-Hill Publications Co., 1983 [14].)



**Figure 7-7b** Velocity vectors in R-Z plane (Lifter Turbine). (By permission, Oldshue, J.Y., "Fluid Mixing Technology", *Chem. Eng.*, McGraw-Hill Publications Co., 1983 [14].)

shaft. The width and shape of the blades may vary depending on the manufacturer and application. These impeller types have been designed to produce high flow with low turbulence and lower Power number. They often provide efficient liquid blending and solids suspension. Most of these axial flow impellers are proprietary, with each manufacturer offering a unique design. Figures 7-9 and 7-10 show improved impeller styles in use in process plants and snapshots of these figures and mixing flow patterns can be downloaded or viewed from the companion website. Figures 7-11 show flow patterns of different impellers types with and without baffles in a laboratory mixing tank. Snap shots of Figures 7-11 can also be downloaded or viewed from the companion website.

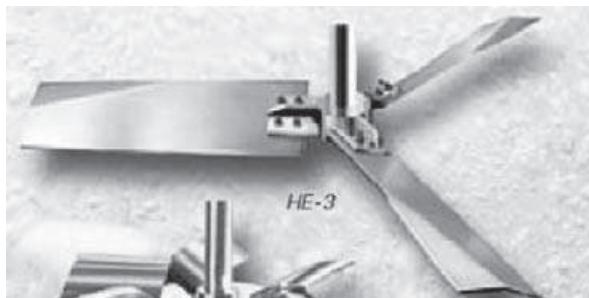


**Figure 7-8** Axial flow impeller.



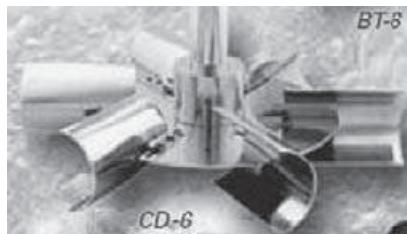
Application: Provides agitation in deep tanks. Produces flow of larger impellers without added weight or loss of efficiency.

**Figure 7-9a** SC-3 impeller. (Courtesy of Chemineer Inc.)



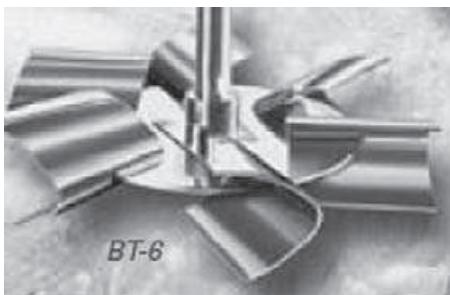
Application: Narrow blade, high efficiency impeller. High flow and low shear such as blending, turbulent heat transfer, and solids suspension.  $N_p = 0.27$ ,  $N_q = 0.5$  (turbulent)

**Figure 7-9b** HE-3 impeller. (Courtesy of Chemineer Inc.)



Application: Concave blade disc turbines (Smith turbine) are used extensively and economically for gas dispersion in large vessels for intermediate and high gas flows.

**Figure 7-9c** CD-6 impeller. (Courtesy of Chemineer Inc.)



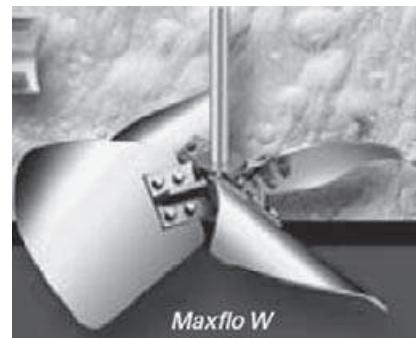
Application: Gas dispersion

**Figure 7-9d** BT-6 impeller. (Courtesy of Chemineer Inc.)



Application: Advanced side entering hydrofoil

**Figure 7-9e** Maxflo WSE impeller. (Courtesy of Chemineer Inc.)



Application: Blending abrasive solids suspension

**Figure 7-9f** Maxflo W impeller. (Courtesy of Chemineer Inc.)



Application: Flat-blade disc turbine (Rushton turbine). Gas dispersion, low and intermediate gas flows. Liquid-liquid dispersion.  $N_p = 5.5$ ,  $N_q = 0.75$

**Figure 7-9g** D-6 type impeller. (Courtesy of Chemineer Inc.)



Application: Straight-blade turbine. Local liquid motion for blending, dispersion, low-liquid level solids suspension. Keeping outlets clear from solids. Design for  $N_{Re} < 50$ ,  $N_p = 3.0$

**Figure 7-9h** S-4 type impeller. (Courtesy of Chemineer Inc.)



Application: Pitched-blade turbine. Substantial axial flow (with a significant radial component) design for wide changes in process viscosity. Blending, dispersion, and solids suspension where increased shear is needed. Heat transfer, surface motion,  $N_p = 1.25$ ,  $N_q = 0.7$  (turbulent)

**Figure 7-9i** P-4 impeller. (Courtesy of Chemineer Inc.)



Application: Ideal for small batches, handles higher viscosities.

**Figure 7-9j** JP-3 impeller. (Courtesy of Chemineer Inc.)



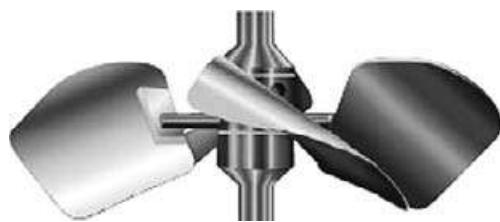
Application: Narrow-blade turbine. Liquid–liquid dispersion, solid–liquid dispersion local shear.

**Figure 7-9k** ChemShear. (Courtesy of Chemineer Inc.)



Application: Liquid–liquid (emulsions) and dense gas–liquid (foams) dispersion

**Figure 7-9l** Dispersion blade impeller. (Courtesy of Chemineer Inc.)



**Figure 7-9m** Smoothline impeller. (Courtesy of Chemineer Inc.)



Application: Helical ribbon. Blending and high viscosity laminar flow impeller. Highly effective heat transfer. Suitable for viscosities of 30,000 MPa s.

**Figure 7-9n** Double-helical ribbon impeller. (Courtesy of Chemineer Inc.)



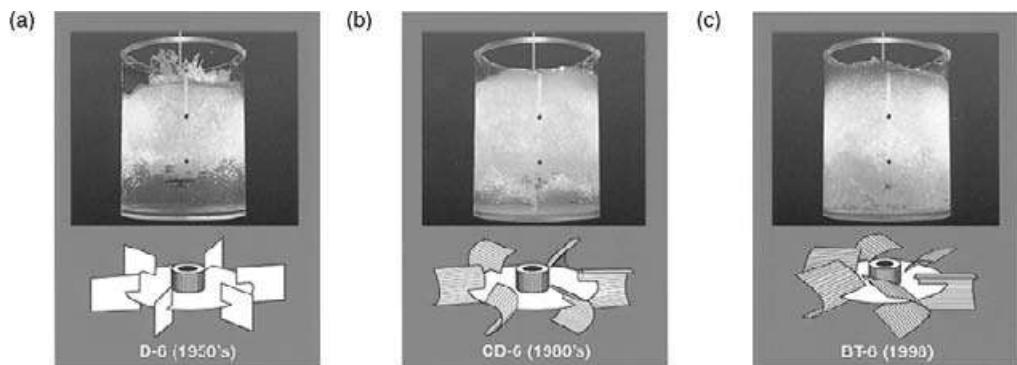
Application: Heat transfer in viscous media. Laminar flow impeller  $0.5 > N_{Re} > 0$ . Solves for heat transfer fouling problems and suitable for low liquid-level geometries.

**Figure 7-9o** Anchor impeller. (Courtesy of Chemineer Inc.)

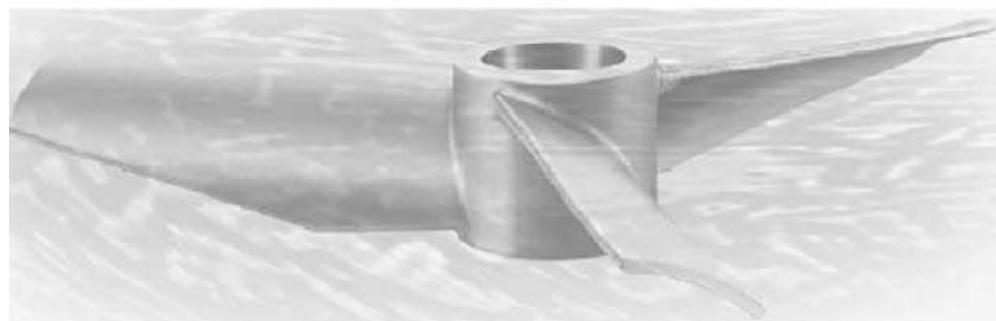


Application: Shear-sensitive uniform blending (polymers). Used in midly pseudoplastics applications.

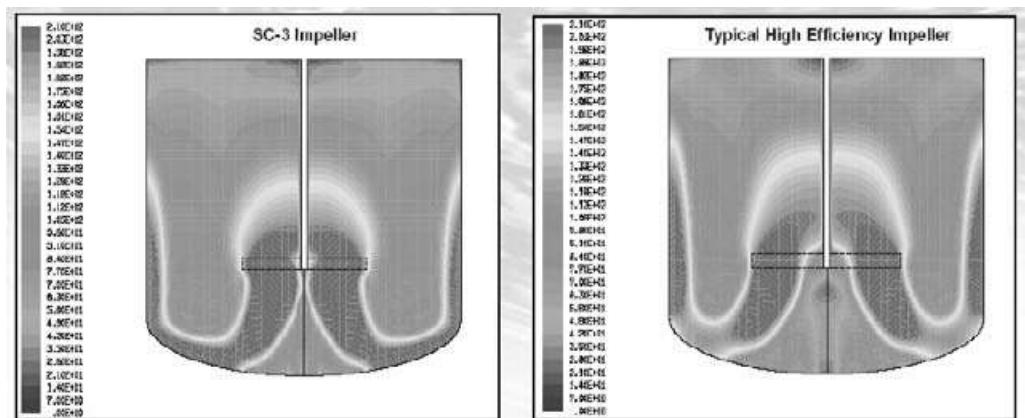
**Figure 7-9p** Screw (auger). (Courtesy of Chemineer Inc.)



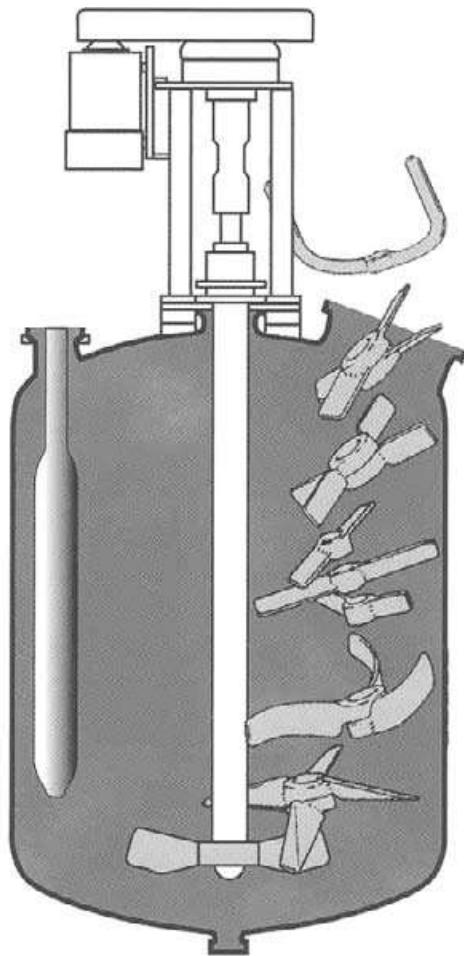
**Figure 7-9q** (a) D-6 impeller (b) CD-6 impeller (c) BT-6 impeller. (Courtesy of Chemineer Inc.)



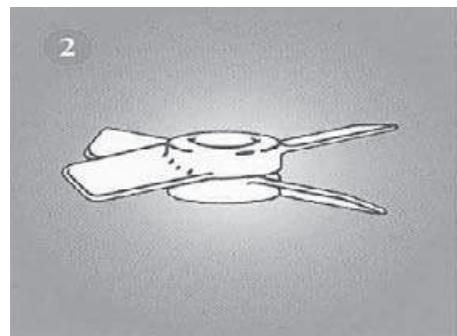
**Figure 7-9r** SC-3 High efficiency impeller. (Courtesy of Chemineer Inc.)



**Figure 7-9s** Average velocity profile comparison between SC-3 and a typical high efficiency impeller. (Courtesy of Chemineer Inc.)

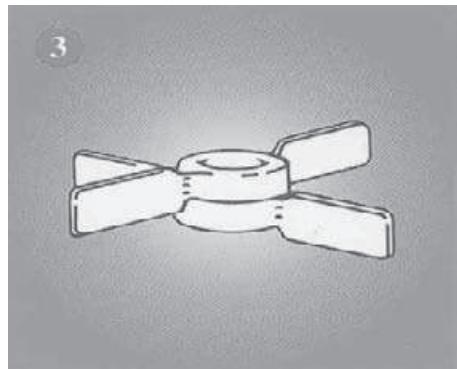


**Figure 7-10a** A continuous-stirred tank/batch reactor. (Courtesy of Pfaudler-Balfour Inc.)



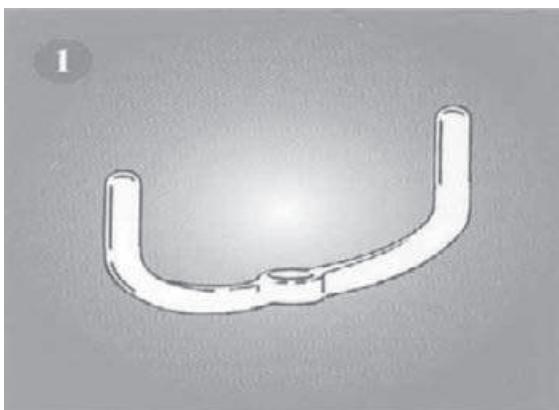
Application: Blending, heat transfer, emulsion, and solids suspension. Moderate shear (higher flow to shear ratio). Combined radial and axial flow and relatively high drive speed.

**Figure 7-10c** PBT Pitched-blade turbine. (Courtesy of Pfaudler-Balfour Inc.)



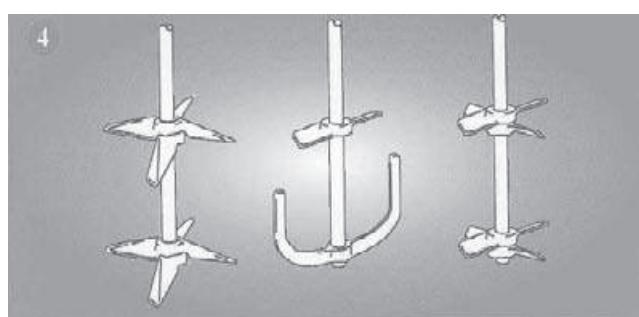
Application: Blending, heat transfer, emulsion, and gas dispersion. High shear load and pure radial flow.

**Figure 7-10d** FBT flat-blade turbine. (Courtesy of Pfaudler-Balfour Inc.)



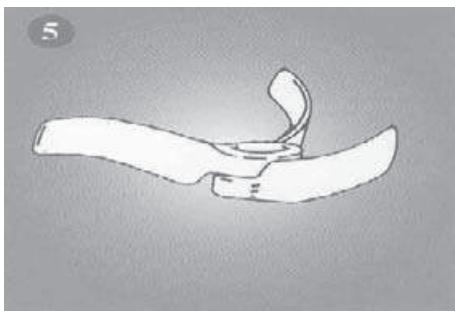
Application: Blending, heat transfer, and solids suspension. Low shear, tangential flow, high torque, for higher viscosity fluids and can operate at low liquid levels.

**Figure 7-10b** ANC Anchor impeller. (Courtesy of Pfaudler-Balfour Inc.)



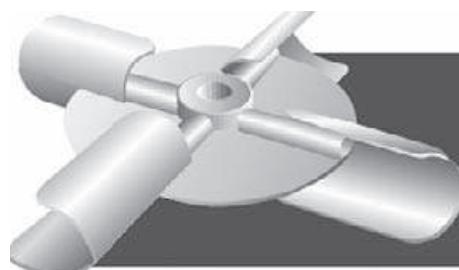
Application: Blending, heat transfer, emulsion, solids suspension. Where single impeller is inadequate, for high viscosity fluids and lightweight solids in narrow vessels.

**Figure 7-10e** MFI multi-flight impeller. (Courtesy of Pfaudler-Balfour Inc.)



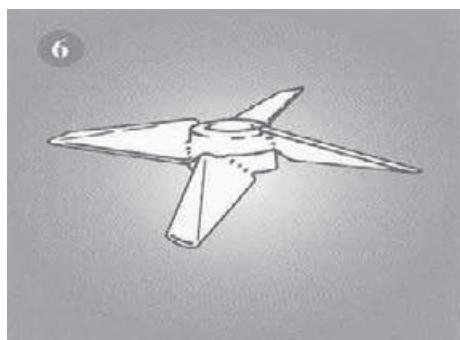
Application: Blending, heat transfer, solids suspension, and gas dispersion. High radial flow, wider baffling required. Insensitive to viscosity variations, suitable for low liquid levels.

**Figure 7-10f** RCI Retreat curve impeller. (Courtesy of Pfaudler–Balfour Inc.)



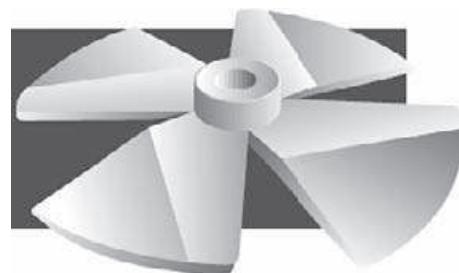
Application: Blending, heat transfer, and gas dispersion. Increases the mass transfer rate.

**Figure 7-10i** Gas dispersion turbine (GST). (Courtesy of Pfaudler–Balfour Inc.)



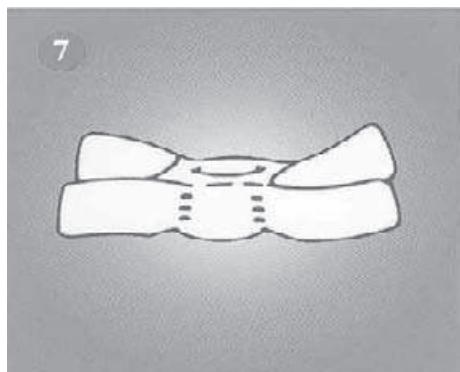
Application: Blending, heat transfer, and solids suspension. Low shear forces, high axial flow at relatively low baffling, low torque, and low power consumption.

**Figure 7-10g** TBF Turbofoil. (Courtesy of Pfaudler–Balfour Inc.)



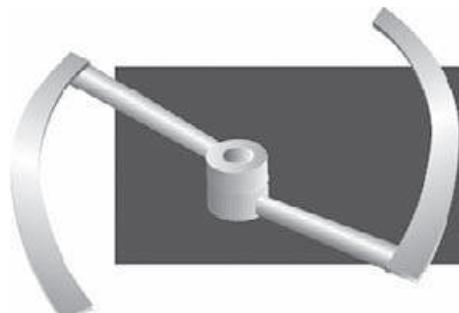
Application: A high efficiency impeller, designed to mix higher viscosity slurries or solutions which cannot be handled by traditional high efficiency impellers (e.g., turbofoil).

**Figure 7-10j** Maxflo turbine (MXT). (Courtesy of Pfaudler–Balfour Inc.)



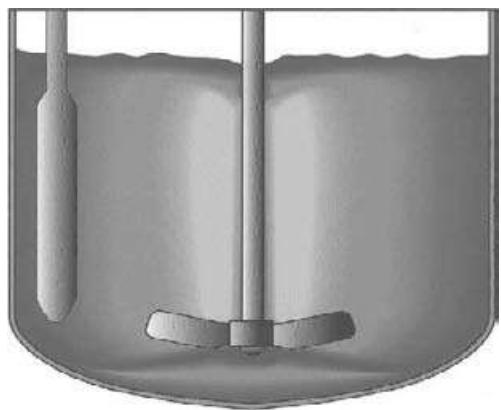
Application: Blending, heat transfer, solids suspension, and gas dispersion. High shear, radial flow, wider baffling required with low viscous fluids.

**Figure 7-10h** CBT Curve-blade turbine. (Courtesy of Pfaudler–Balfour Inc.)

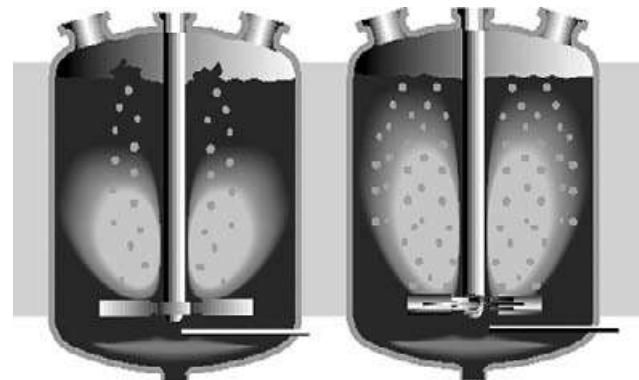


Application: Designed to handle high viscosity (>100,000 cP) blending.

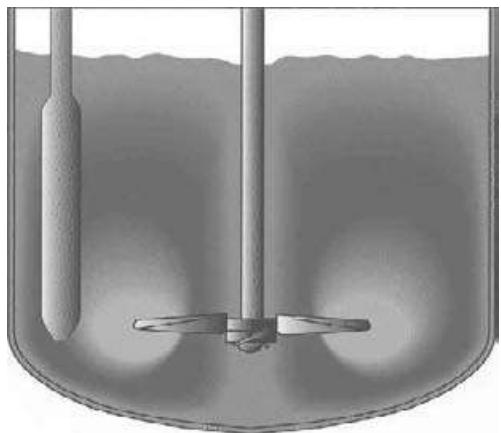
**Figure 7-10k** Helical ribbon (HLX). (Courtesy of Pfaudler–Balfour Inc.)



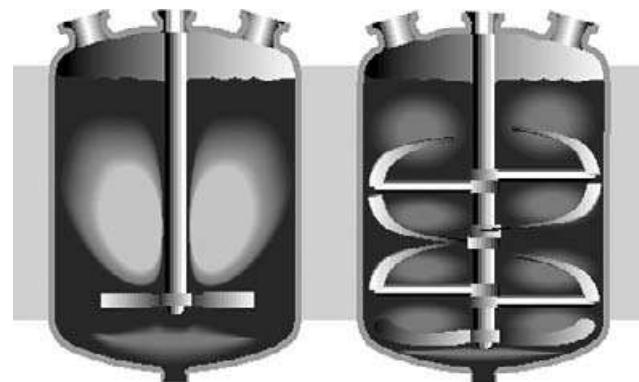
**Figure 7-10l** Typical flow patterns with curved-blade turbine. (Courtesy of Pfaudler-Balfour Inc.)



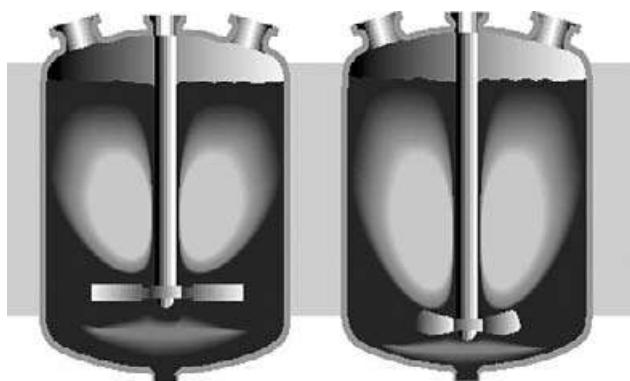
**Figure 7-10o** Typical flow patterns involving gas dispersion. (Courtesy of Pfaudler-Balfour Inc.)



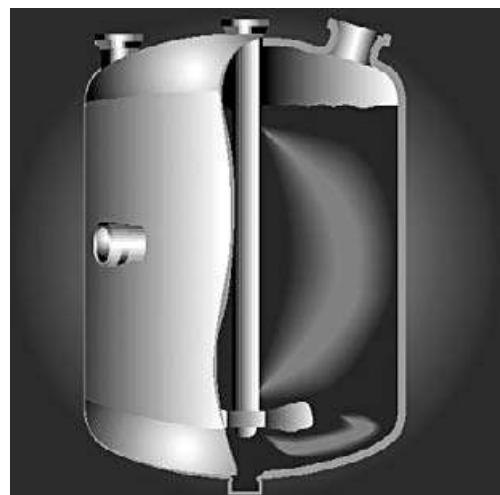
**Figure 7-10m** Typical flow patterns with Turbofoil impeller. (Courtesy of Pfaudler-Balfour Inc.)



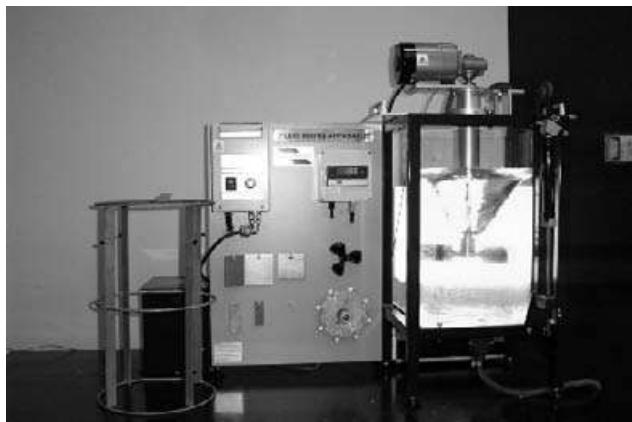
**Figure 7-10p** Typical flow patterns involving high viscosity mixing. (Courtesy of Pfaudler-Balfour Inc.)



**Figure 7-10n** Typical flow patterns involving low level agitation. (Courtesy of Pfaudler-Balfour Inc.)



**Figure 7-10q** Typical flow patterns in a reactor. (Courtesy of Pfaudler-Balfour Inc.)



**Figure 7-11a** Vortex formation: Paddle-type impeller in water at 186 rpm ( $H/D_T = 1$ ,  $D_A/D_T = 0.6$ ) and a ring of four baffles, dimensions (Baffle height = 440 mm, width = 25 mm). (By permission of Armfield Ltd.)



**Figure 7-11b** Paddle-type impeller in a baffled tank containing water at 187 rpm ( $H/D_T = 1$ ,  $D_A/D_T = 0.6$ ,  $J/D_T = 1/12$ ,  $L/D_A = 0.52$ ). (By permission of Armfield Ltd.)

Figure 7-12 is an analysis flowchart for examining types of turbine impeller performance requirements.

For some services there may be more than one impeller on the shaft, attached part-way up the shaft from the lower one (Figures 7-13a and b).

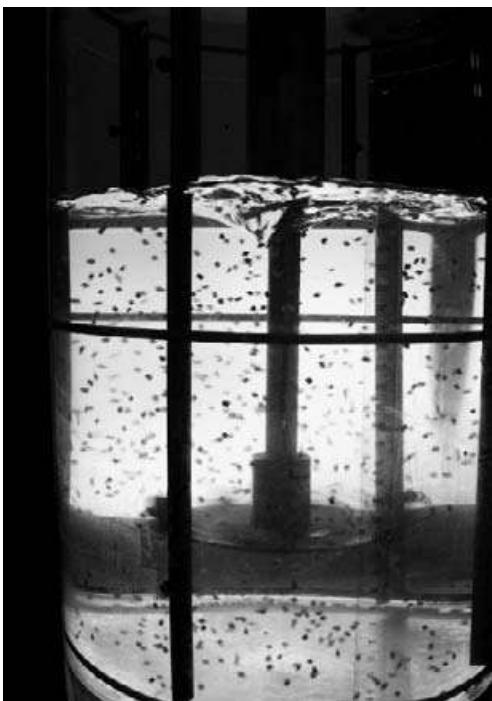
The use of dual impellers on a shaft should be determined by the physical properties and characteristics of the system, in general being a function of viscosity, impeller diameter, and liquid depth in the tank. Generally, dual impellers may be indicated for fluids



**Figure 7-11c** Propeller-type impeller in a baffled tank containing water at 440 rpm ( $H/D_T = 1$ ,  $D_A/D_T = 0.27$ ,  $J/D_T = 1/12$ ,  $L/D_A = 0.35$ ). (By permission of Armfield Ltd.)



**Figure 7-11d** Vortex formation: Propeller-type impeller in water at 443 rpm ( $H/D_T = 1$ ,  $D_A/D_T = 0.27$ ). (By permission of Armfield Ltd.)



**Figure 7-11e** Rushton turbine impeller in a baffled tank containing water at 428 rpm ( $H/D_T = 1$ ,  $D_A/D_T = 0.41$ ,  $J/D_T = 1/12$ ,  $L/D_A = 0.35$ ). (By permission of Armfield Ltd.)



**Figure 7-11f** Vortex formation, Rushton turbine impeller at 428 rpm ( $H/D_T = 1$ ,  $D_A/D_T = 0.41$ ). (By permission of Armfield Ltd.)

of 45 cP and greater and where the fluid travels more than 4 ft before being deflected.

The circulating capacity of three-blade square pitch propellers is theoretically of the magnitude given in Figure 7-14. The speed ranges indicated may be grouped as [27]:

high speed (1750 rpm) for low viscosity fluids, such as water  
medium speed (1150 rpm) for medium viscosity fluids, such as  
light syrups and varnishes

low speed (420 rpm) for high viscosity fluids such as oils, paints,  
or for tender crystals or fibers; or if foaming is a problem.

The mixing efficiency is generally higher (40–60%) for the slow 400 rpm speed and lower (25–45%) for the 1750 and 1150 rpm speeds. This is given in Figure 7-15 for general estimating use. Note that the turnover of tank capacity is involved through the selected impeller diameter and speed.

### 7.3 EQUIPMENT FOR AGITATION

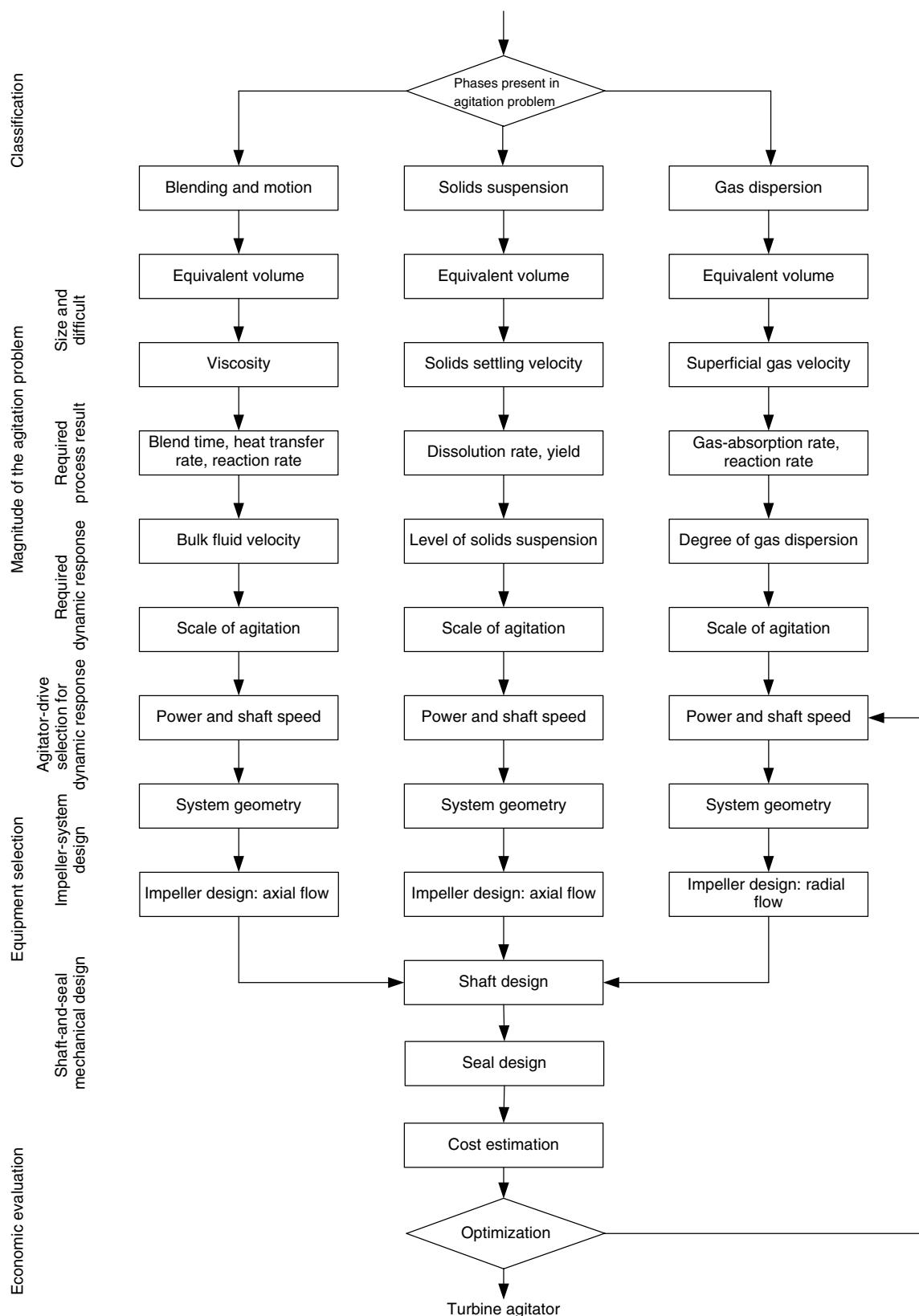
Various types of vessels and tanks of geometrical shapes and sizes are used for the mixing of fluids. The top of the vessel may be either opened or sealed. A typical batch reactor (see Figure 7-10a) is applicable in many operations. There are vessels where the tank bottom shape is not flat, but rounded to eliminate sharp corners or regions into which the fluid currents would not penetrate; dished ends are most common. The liquid depth is approximately equal to the diameter of the tank. An impeller is mounted on an overhung shaft, that is a shaft supported from above. The shaft is driven by a motor which is sometimes directly connected to the shaft but often through a speed-reducing gearbox (see Figure 7-1). Other attachments are inlet and outlet lines, coils, jackets, and wells for thermometers. Figures 7-16a and b show typical standard tank configuration for flat and dished tanks respectively. The geometric proportions of the agitation system, which are considered as a typical standard design, are given in Table 7-2. These relative proportions form the basis of the major correlations of agitation performance from various studies.

There are cases where  $\frac{W}{D_A} = \frac{1}{8}$  and  $\frac{J}{D_T} = \frac{1}{10}$  for some agitator correlations. The number of baffles often used is 4 and the clearance between the baffles and the wall is about  $0.1\text{--}0.15J$ . This ensures that the liquid does not form stagnant pockets between the baffle and the wall. The number of impeller blades varies from 4 to 16, but is generally between 6 and 8.

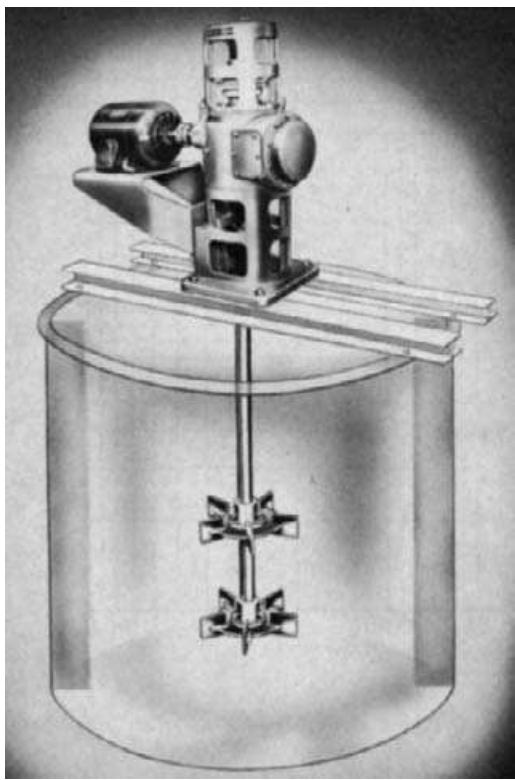
Mixing by agitation of liquids involves the transfer of momentum from an impeller to the liquid. An impeller, which is mounted on a shaft driven by an electric motor, is divided into two categories of operation:

1. Where momentum is transferred by shearing stresses, in which the transfer is perpendicular to the direction of flow. This category includes the rotating disc and cone agitators.
2. The momentum is transferred by normal stresses, in which the transfer is parallel to the direction of flow. This category includes the paddle, propeller, and turbo mixer agitators.

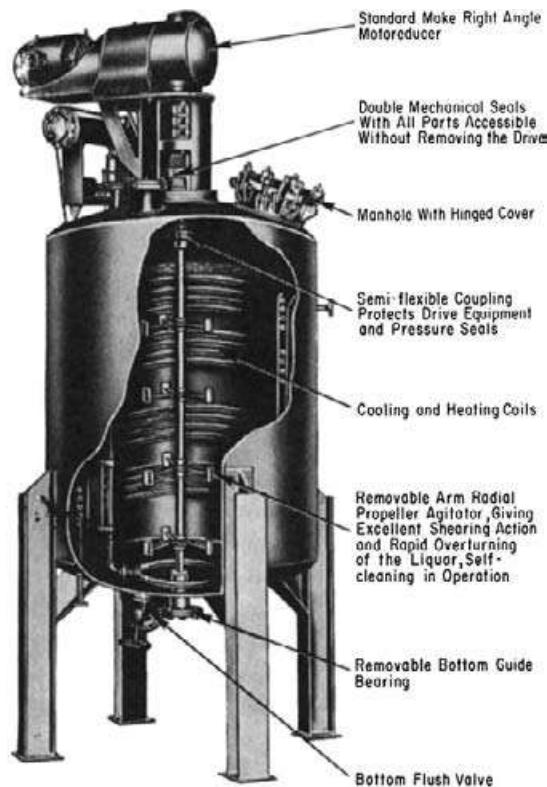
Agitation plays an essential role in the success of many chemical processes, and there is a wide range of commercially available impellers that give optimum agitation for any process (see Figures 7-6, 7-9, and 7-10). The problem arises in selecting the best impeller for the required process. Equipment manufacturers often provide expert guidance, but it is beneficial for designers and engineers to acquire fundamental knowledge of various types of impellers. The process objective of impellers is the primary factor that influences their selection. These objectives as summarized in Table 7-1 together with physical properties such as viscosity play



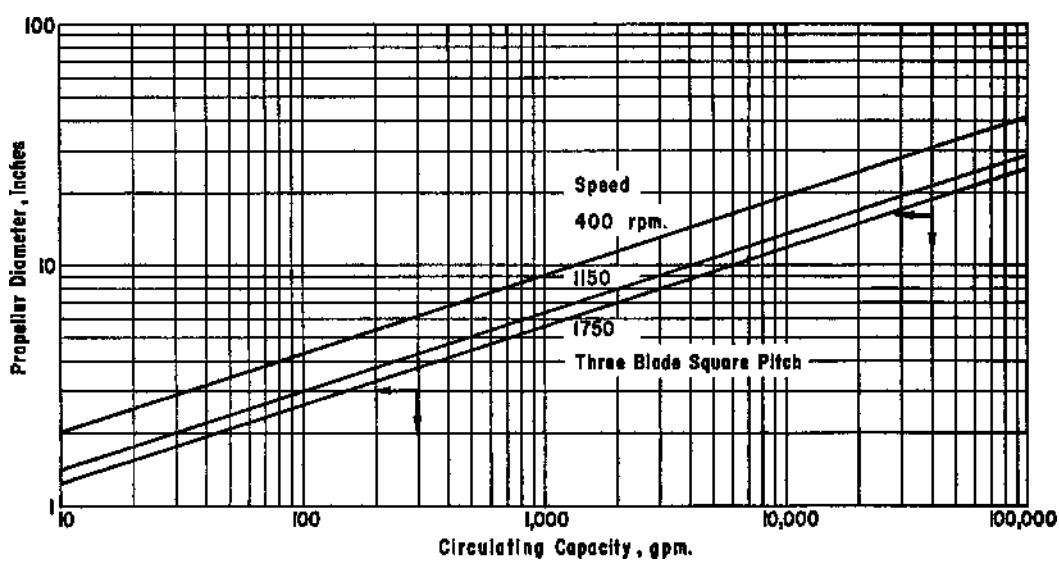
**Figure 7-12** Analysis flow chart for examining types of turbine impeller applications. (By permission from Gates, L.E. et al., *Chem. Eng.*, Dec 8, 1975, p. 110 [11].)



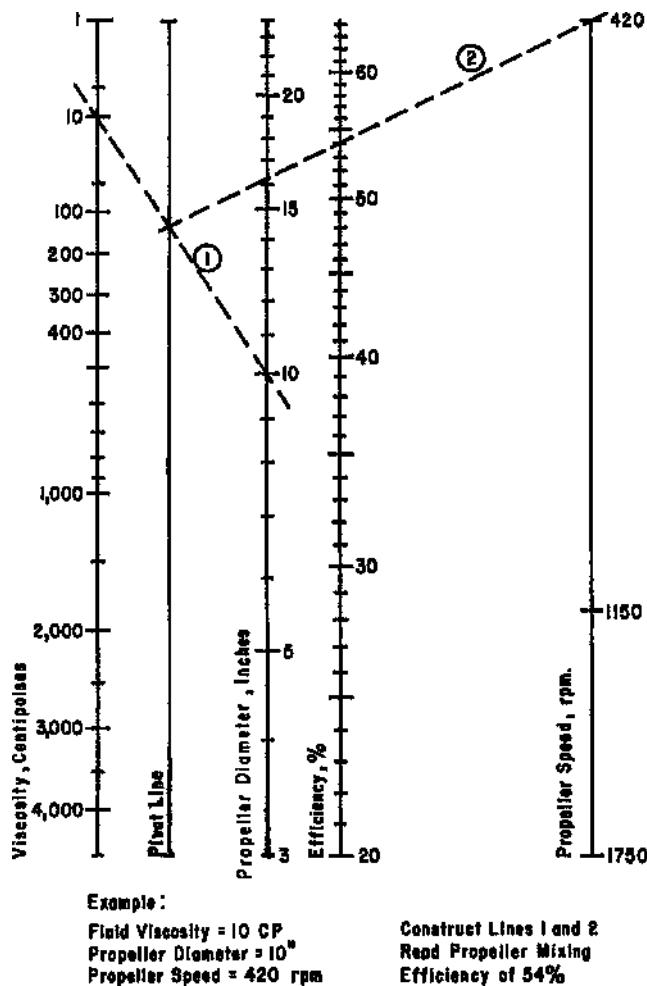
**Figure 7-13a** Dual impeller mixer and drive. (Courtesy of Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



**Figure 7-13b** Multiple impellers. (Courtesy of Struthers-Wells Corp.)



**Figure 7-14** Theoretical circulating capacity of single propeller mixers. (By permission from *Fluid Agitation Handbook*, Chemineer Inc.)



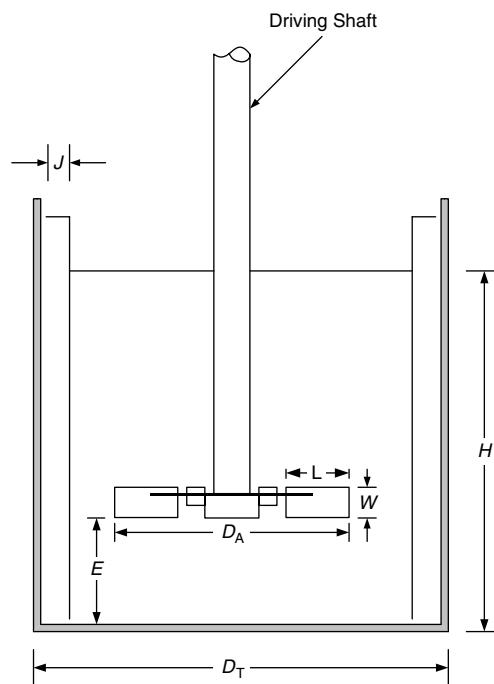
**Figure 7-15** Propeller circulation efficiency. This is used with theoretical propeller capacity to determine actual capacity. (By permission from *Fluid Agitation Handbook*, Chemineer Inc.)

an important role in the selection of impellers in laminar, transitional, and turbulent operations. Several methods of selecting an impeller are available [25, 28], and [29]. Figure 7-17 shows one method based on liquid viscosity and tank volume, and Table 7-3 illustrates another based on the liquid viscosity alone.

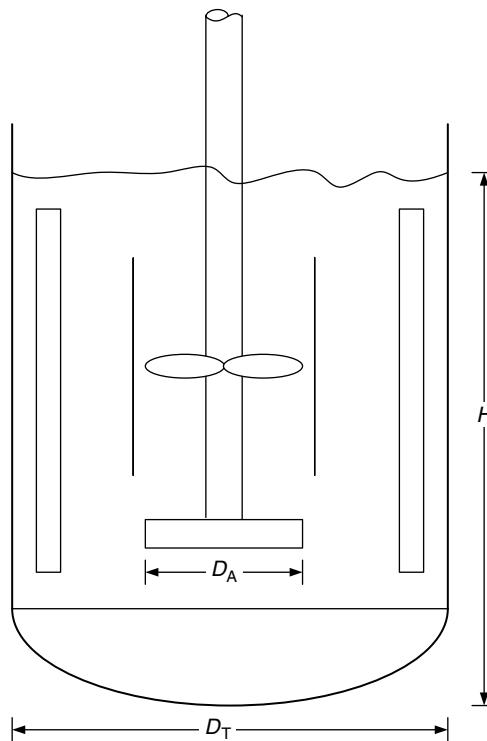
**TABLE 7-2 Geometric Proportions for a Standard Agitation System**

$\frac{D_A}{D_T} = \frac{1}{3}$	$\frac{H}{D_T} = 1$	$\frac{J}{D_T} = \frac{1}{12}$
$\frac{E}{D_A} = 1$	$\frac{W}{D_A} = \frac{1}{5}$	$\frac{L}{D_A} = \frac{1}{4}$

B = number of blades on impeller  
R = number of baffles  
D<sub>A</sub> = agitator diameter  
H = liquid height  
D<sub>T</sub> = tank diameter  
E = height of the agitator from the bottom of the tank  
J = baffle width  
L = agitator blade length  
W = agitator blade width.



**Figure 7-16a** Standard tank configuration, not to scale, showing a flat six-blade turbine impeller.



**Figure 7-16b** A basic stirred tank design, not to scale, showing lower radial impeller and an upper axial impeller housed in a draft tube. Four equally spaced baffles are standard. H = height of liquid level, D<sub>T</sub> = tank diameter, D<sub>A</sub> = impeller diameter. For radial impellers,  $0.3 \leq D_A/D_T \leq 0.6$ .

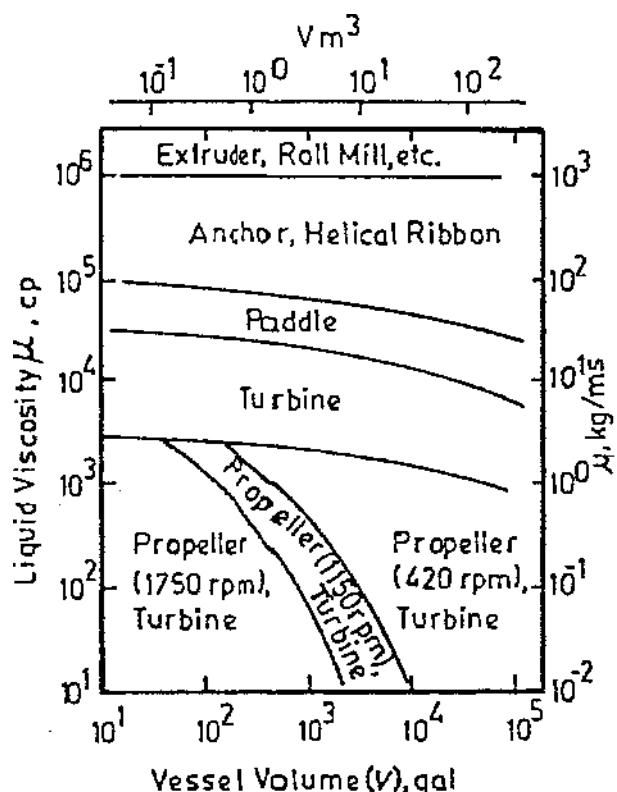


Figure 7-17 Impeller selection. (Source: Penny, W.R., "Guide to trouble free mixers", *Chem. Eng.*, Vol. 77, No. 12, 1970, p. 171.)

TABLE 7-3 Impeller Selection Guide

Type of Impeller	Range of liquid cP	Viscosity kg/m s
Anchor	$10^2$ – $2 \times 10^3$	$10^{-1}$ – $2$
Propeller	$10^0$ – $10^4$	$10^{-3}$ – $10^1$
Flat-blade turbine	$10^0$ – $3 \times 10^4$	$10^{-3}$ – $3 \times 10^1$
Paddle	$10^2$ – $3 \times 10^1$	$10^{-1}$ – $3 \times 10^1$
Gate	$10^3$ – $10^5$	$10^0$ – $10^2$
Helical screw	$3 \times 10^3$ – $3 \times 10^5$	$3$ – $3 \times 10^2$
Helical ribbon	$10^4$ – $2 \times 10^6$	$10^1$ – $2 \times 10^3$
Extruders	$>10^6$	$>10^3$

(Source: Holland, F.A. and F.S. Chapman, *Liquid Mixing and Processing in Stirred Tanks*, Reinhold, New York, 1966.)

Axial flow devices such as high efficiency (HE) (see Figures 7-9) impellers and pitched blade turbines give better performance to conventional pitched blade turbines. They are best suited to provide the essential flow patterns in a tank to keep the solids in suspension. High efficiency impellers effectively convert mechanical energy to vertical flow required to overcome the effects of gravity on solids in suspension. In addition, high efficiency impellers provide the same levels of solids suspension at reduced capital and operating costs.

## 7.4 FLOW PATTERNS

In fluid agitation, the direction as well as the magnitude of the velocity is essential. The directions of the velocity vectors

throughout an agitated vessel are referred to as the flow pattern. Since the velocity distribution is constant in the viscous and turbulent ranges, the flow pattern in an agitated vessel is fixed.

During mixing of fluids, it is essential to avoid solid body rotation and a large central surface vortex. When solid body rotation occurs, there is no adequate mixing taking place as the fluid rotates as if it were a single mass as shown in Figure 7-18 (also see Figures 7-11). Centrifugal force of the fluid causes a central surface vortex being thrown outward by the impeller. Entrainment of air results if the vortex reaches an impeller, resulting in reduced mixing of the fluids. This situation can be averted by installing baffles on the vessel walls, which impede rotational flow without interfering with radial or longitudinal flow (see Figures 7-11). An effective baffling is attained by installing vertical strips perpendicular to the wall of the tank. With the exception of large tanks, four baffles are adequate to prevent swirling and vortex formation. For propellers, the width of the baffle should not exceed one-eighteenth the diameter of the tank; for turbines, not more than one-twelfth the tank diameter.

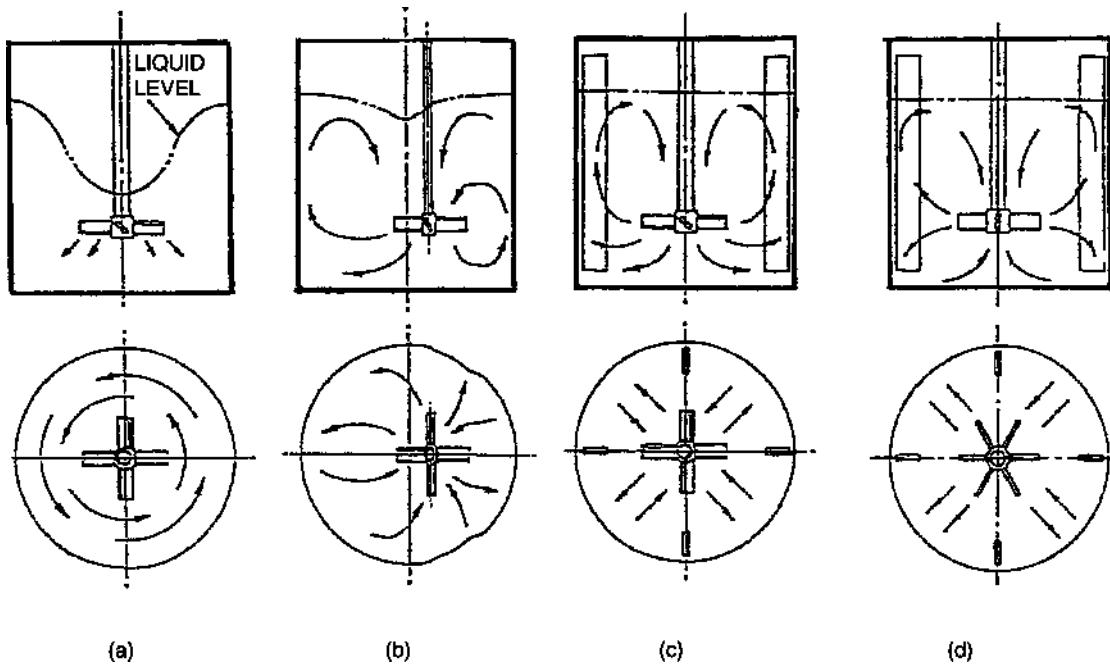
An alternative to reducing vortex formation is by placing an impeller in an off-center position. This creates an unbalanced flow pattern either reducing or eliminating the swirl, and thereby increasing or maximizing the power consumption. This position is critical, since too far or too little off-center in one direction or the other will cause greater swirling, erratic vortexing, and dangerously high shaft stresses. Changes in viscosity and tank size also affect the flow pattern in such vessels. Off-center mounting of radial or axial flow impellers is readily employed as a substitute for baffled tank installations. It is common practice with propellers, but less with turbine agitators. Off-center mounting can also be useful with turbine in the medium viscosity range, and with non-Newtonian fluids where baffles would cause stagnation with little swirl of the fluid. Off-center mountings have been quite effective in the suspension of paper pulp. The patterns illustrated in Figures 7-19a–k indicate that almost any pattern can be established, provided the particular impeller type is located in the proper position. This is easier to accomplish in some systems than others. Figure 7-19d shows flow patterns of an angular impeller in an off-center position.

Once swirling stops, the specific flow pattern in the tank depends on the type of impeller. Paddle agitators and flat-blade turbines give good radial flow in the plane of the impeller with the flow dividing the wall to form two separate circulation patterns (Figures 7-5). One portion flows downward along the wall and back to the center of the impeller from below, and the other flows upward toward the surface and back to the impeller from above.

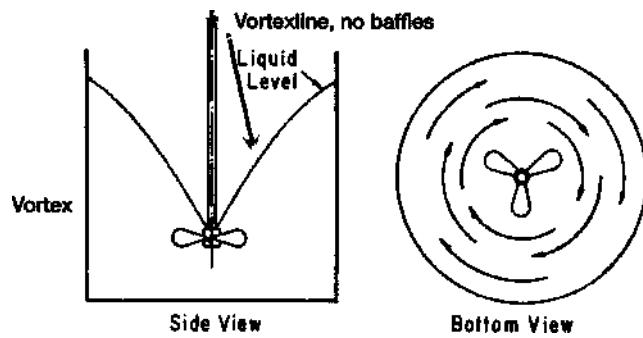
Propeller agitators drive the liquid down to the bottom of the tank, where the stream spreads radially in all directions toward the wall, flows upward along the wall and returns to the suction of the propeller from the top. The earlier Figure 7-5b shows the flow pattern of a propeller agitator. Propellers are employed when heavy solid particles are kept in suspension.

The use of vertical side wall baffles usually destroys the rotary and swirling motion in vertical tanks. This also can be accomplished to a degree by setting the mixer off-center. These baffles should be 1/10–1/12 of the width or diameter of the tank. Six baffles generally give slightly better performance than four; although four is the usual number, with three not being as good for most situations.

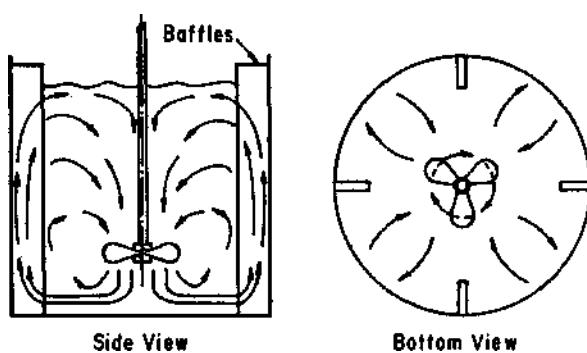
Table 7-4 shows flow patterns and applications of some commercially available impellers. In general, the axial flow pattern is most suitable for flow-sensitive operations such as blending, heat transfer, and solids suspension, while the radial flow pattern is ideal for dispersion operations that require high shear levels than are provided by axial flow impellers. Myers et al. [30] have provided a range of selection of impellers with applications. However, further details on selection are provided by [11, 13, 31, and 32].



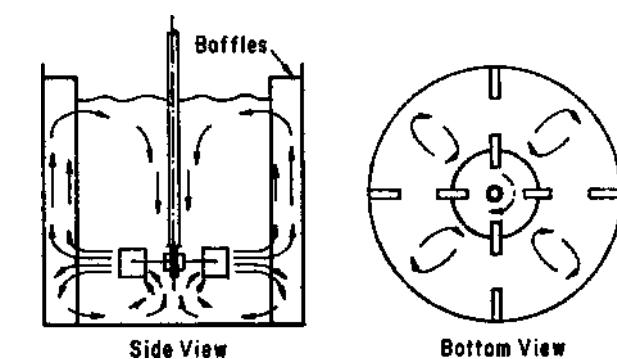
**Figure 7-18** Agitator flow patterns: (a) Axial or radial impellers without baffles produce vortex. (b) Off-center location reduces the vortex. (c) Axial impeller with baffles. (d) Radial impeller with baffles. (Source: Wales, S.M., *Chemical Process Equipment – Selection and Design*, Butterworths Series in Chemical Engineering, 1988.)



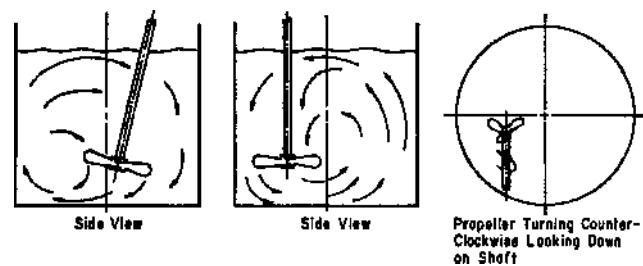
**Figure 7-19a** Fluid flow pattern for propeller mounted at center with no baffles. Note vortex formation. (By permission from Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



**Figure 7-19b** Fluid flow pattern for propeller mounted at center with baffles, axial flow pattern. (By permission from Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



**Figure 7-19c** Fluid flow pattern for turbine mounted on-center with baffles, radial flow pattern. (By permission from Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)



**Figure 7-19d** Fluid flow pattern for propeller mounted in angular off-center position. (By permission from Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)

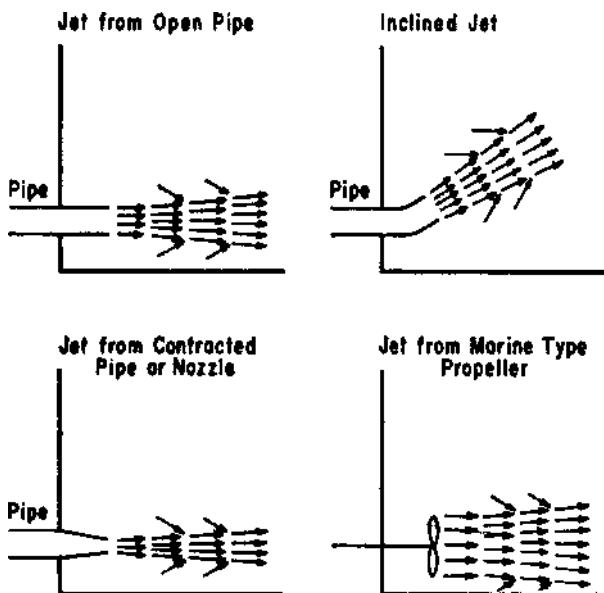


Figure 7-19e Entrainment mixing jets. (By permission from Rushton, J.H., *Petroleum Refiner*, Vol. 33, 1954, p. 101 [33].)

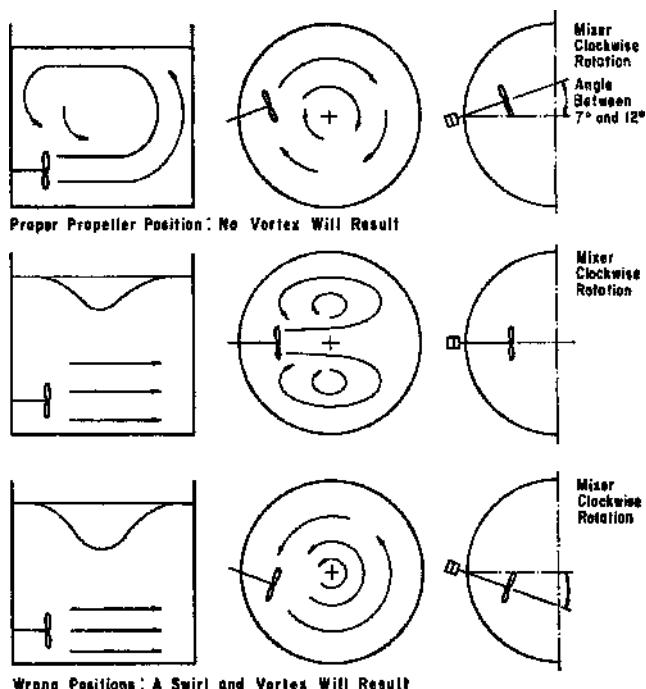


Figure 7-19g Side-entering propeller mixer position, large tanks. (By permission from Rushton, J.H., *Petroleum Refiner*, Vol. 33, 1954, p. 101 [33].)

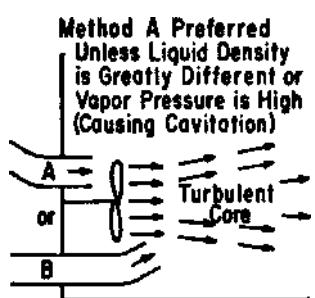


Figure 7-19f Introducing liquid during mixer operation. (By permission from Rushton, J.H., *Petroleum Refiner*, Vol. 33, 1954, p. 101 [33].)

## 7.5 FLOW VISUALIZATION

A necessary means of understanding mixer performance is flow visualization. Visualizing flow patterns and fluid motion help to better understand how different impellers work. Since flow patterns are somehow difficult to observe from surface motion, other means are necessary. Computational fluid dynamics (CFD) uses mathematical representations of the continuity equations to describe fluid flow. First, the agitated tank is divided into many small cells. Then equations for conserved properties (e.g., mass, momentum, and energy) are written for the fluid in the cell and connected mathematically to the adjacent cells. Successive calculations reduce the errors between initial cell values and conditions that satisfy internal and boundary conditions. The equations calculate radial, axial, and rotational components of the fluid velocity at each cell in the tank model [32]. Computational fluid dynamics calculations can create visual impressions of fluid motion, and models can be used to describe almost any fluid in any size tank. However, the limitation is the ability of the model to predict real results accurately. Experimental techniques such as laser Doppler anemometry (LDA) and time-averaged digital particle image velocimetry (DPIV) can create similar results to verify or validate the results of

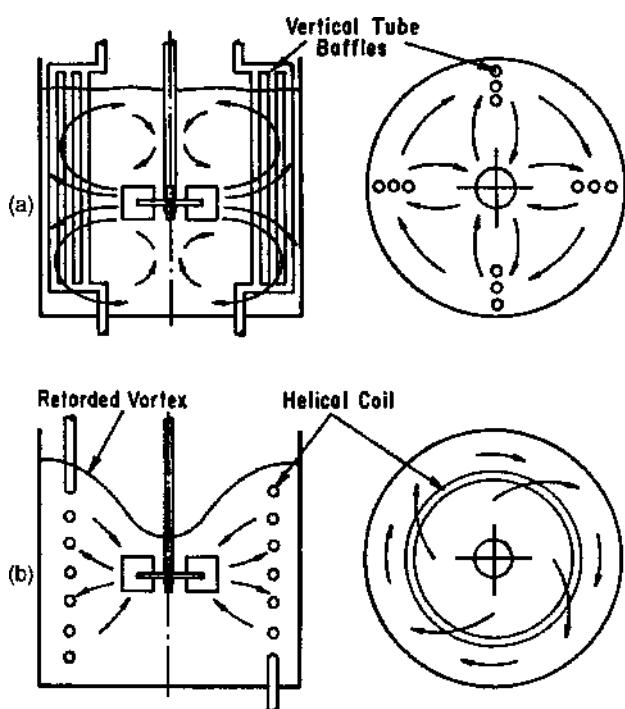
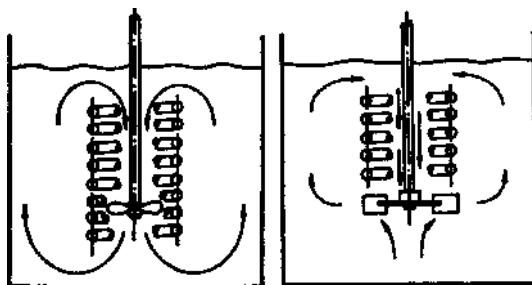
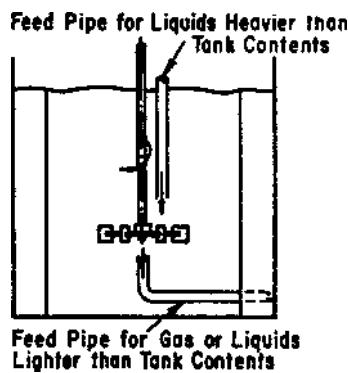


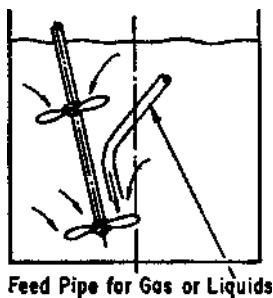
Figure 7-19h Liquid motion patterns: (a) Vertical-tube baffles. (b) Helical coil, no other baffles. (By permission from Dunlap, J.R., Jr and J.H. Rushton, AIChE Symp. Ser., Vol. 49, No. 5, 1953, p. 137 [34].)



**Figure 7-19i** Coil used as draft tube. (By permission from Dunlap, J.R., Jr and J.H. Rushton, AIChE, Symp. Ser., Vol. 49, No. 5, 1953, p. 137 [34].)



**Figure 7-19j** Feed of liquids and gases to turbine. (By permission from Dunlap, J.R., Jr and J.H. Rushton, AIChE, Symp. Ser., Vol. 49, No. 5, 1953, p. 137 [34].)



**Figure 7-19k** Feed of liquids and gases to dual propellers. (By permission from Dunlap, J.R., Jr and J.H. Rushton, AIChE, Symp. Ser., Vol. 49, No. 5, 1953, p. 137 [34].)

computer modeling. Additionally, laboratory models in transparent tanks also assist in further providing opportunities for flow visualization. Adequate work has been carried out to compare laboratory observations with CFD analysis to verify that time average velocity results can be reliably modeled, however, time average velocity profiles have limitations as much mixing occurs due to random and structured fluid motion.

Two forms of visualization, namely vector plots and raster plots, represent the liquid velocities in agitated tanks. The former represent velocity by arrows and show the radial and axial velocity components in a plane through a vertical section of the tank model.

The latter show velocity magnitudes as color intensity as represented by Figures 7-9s, 7-10l-p and do not attempt to show velocity direction. These plots show high velocities in bright colors and low velocities in dark colors. Furthermore, the plots can be scaled to show the same velocity magnitudes in different mixing conditions. The same color scales can be used to show plots of impellers and mixing conditions. Thus, a model showing larger regions of red representing a high velocity and smaller regions of blue representing low velocity should predict better mixing than a model showing lower velocities.

## 7.6 MIXING CONCEPTS, THEORY, FUNDAMENTALS

A mixer unit or impeller, regardless of its physical design features, is a pump of varying efficiency for pumping of fluids. Generally, all designs are low heads compared to a conventional centrifugal pump, because there is no defined confining casing for the mixing element.

The action of the impeller design produces flow of the fluid, head on the fluid, or shear in the fluid, all to varying degrees depending on the specific design. A general identification of these characteristics for several impeller types are given by [12] (*Note: Use consistent dimensions*).

Impeller Type	Flow Decreases from Top	Head (Shear) Increases from Top
Rakes, gates		
Spirals, anchors		
Paddles		
Propellers		
Axial flow turbines	Decreases	
Flat blade turbine		
Bar turbine		
Bladeless impeller		
Close clearance		
Impeller and stator		
Colloid mill, Homogenizer		Increases

The horsepower required for any impeller is partly used for pumping flow and partly for shear requirements. To accomplish a given mixing performance for a process operation, the objective usually becomes a matching of the quantity of flow from an impeller with the shear characteristics at a specific power input. The flow/shear input ratio to a fluid system can be shifted or changed by changing the type/physical characteristics of the impeller, not the dimensions of a specific impeller design. For particular dimensional features (angles of blades, height/depth of blades, number of blades, etc.), the performance will remain the same as long as the dimensions are in the same relative relationship as the impeller, that is, in the same performance family.

## 7.7 FLOW

The quantity of flow is defined as the amount of fluid that moves axially or radially away from the impeller at the surface or periphery of rotation. This flow quantity is never actually measured, but its relative relation to head characterizes the particular system. The flow rate,  $Q$ , is usually available from the manufacturer for a given impeller [23].

$$Q = K_1 ND^3 \cong ND^3 \quad (7-1)$$

**TABLE 7-4 Impellers and Flow Patterns**

Impeller	Flow Pattern	Name and Description	Applications
		HE-3 Narrow-blade, high-efficiency impeller	Blending, Turbulent heat transfer, Solid suspension,  Upper impeller for gas dispersion, $N_p = 027, N_q = 05$ (turbulent)
		P-4 Pitched-blade turbine	Blending, Dispersion, Solid suspension  Heat transfer, Surface motion, $N_p = 125, N_q = 07$ (turbulent)
		S-4 Straight-blade turbine	Local liquid motion for blending, Dispersion, keeping outlets dear from solids,  $N_p = 3.0$
		Maxflo T Wide-blade, high-efficiency impeller	Blending, Transitional flow, Simultaneous gas dispersion and solid suspension (like mining),  $N_p$ and $N_q$ vary with tip angle and number of blades
		ChemShear Narrow-blade turbine	Liquid-liquid dispersion, Solid-liquid dispersion, Local shear
		D-6 Flat-blade disc turbine (Rushton turbine)	Gas dispersion, low and intermediate gas flows, Liquid-liquid dispersion; $N_p = 5.5, N_q = 0.75$
		CD-6 Concave-blade disc turbine (Smithturbine)	Gas dispersion, intermediate and high gas flows
		Helical ribbon (Double flight shown)	Blending and heat transfer in viscous media ( $\mu > 50 \text{ Pa s}$ or $N_{Re} < 100$ ) $-N_p = 350/N_{Re}, N_{Re} < 100$
		Anchor	Heat transfer in viscous media $N_p = 400/N_{Re}, N_{Re} < 10$
		CD-6 / HE-3 / P-4	Gas dispersion and blending for tall reactors Fermentations (food products, pharmaceuticals)
		CD-6 / HE-3	Combined gas-dispersion, blending, and material drawdown (corn wet milling)
		Side-entering wide blade impeller (HE3-S or Mark II)	Oilstorage, Paper pulp, Waste water circulation, Flue gas desulphurisation

(Source: Myers, K. et al., Agitation for Success, *Chem. Eng.*, Oct 10, 1996.)

where

$Q$  = flow rate from impeller, ft<sup>3</sup>/s

$N$  = speed of rotation, rps

$D$  = impeller diameter, ft

$K_1$  = proportionality constant, a function of the impeller shape  
= 0.40 for three-blade propeller in water.

Figure 7-14 indicates the theoretical circulation from a propeller, and Figure 7-15 gives its efficiency for estimating purposes. Efficiency must be used in converting theoretical to actual horsepower, or in converting theoretical to actual circulation of the propeller.

### FLOW NUMBER (PUMPING NUMBER)

The primary pumping capacity of an impeller is determined by the impeller diameter, the Pumping number, and the rotational speed. This is probably the most important dimensionless group used to represent the actual flow during mixing in a vessel. The Flow or Pumping number,  $N_Q$ , is defined by

$$N_Q = \frac{Q'}{(N_m D^3)} \quad (7-2)$$

where

$N_m$  = impeller speed of rotation, rpm

$Q'$  = flow rate or pumping capacity, ft<sup>3</sup>/min

$D$  = impeller diameter, ft

In Metric units:

$$N_Q = \frac{Q_p}{ND_A^3} \quad (7-2a)$$

where

$Q_p$  = effective pumping capacity, m<sup>3</sup>/s

$N$  = impeller rotational speed, s<sup>-1</sup>

$D_A$  = impeller diameter, m.

The Pumping number is used to determine the pumping rate  $Q_p$  of an impeller. Figure 7-20 shows plot of Pumping number  $N_Q$  as a function of Reynolds number  $N_{Re}$  for various ratios of impeller diameter to tank diameter ( $D_A/D_T$ ).  $N_Q$  is strongly dependent on the flow regime, Reynolds Number,  $N_{Re}$ , and installation geometry of the impeller. The flow from an impeller is only that produced by the impeller and does not include the entrained flow which can be a major part of the total "motion" flow from the impeller. The entrained flow refers to fluid set in motion by the turbulence of the impeller output stream [12]. To compare different impellers, it is important to define the type of flows being considered. The plots on Figure 7-20 can be transformed into a series of equations and incorporated into a computer program. Vlyssides et al. [35] have provided steps in transforming a nomograph and Figure 7-20 into a series of equations as a subroutine in a computer program. This exercise can also be incorporated into an Excel spreadsheet.

### 7.8 POWER

Power is the external measure of the mixer performance. The power put into the system must be absorbed through friction in viscous and turbulent shear stresses and dissipate as heat. The flow mechanism in a mixing tank is very complex. Various

techniques, including computational fluid dynamics (CFD) and computational fluid mixing (CFM) tools, are employed together with experimental data to establish improvements in mixing with increased yield. Estimating the power consumption for any agitator is essential for design. Generally, the desired requirements for the system to be mixed will categorize the type of impeller to be used. Laboratory tests on the system can establish the appropriate speed for the maintenance of isotropic turbulence in the mixing vessel. Therefore, estimating the power consumption for the large-scale mixing operation must include scale-up considerations. These requirements may be determined from Navier-Stokes equation or analyzed by a dimensional analysis of the mixing operation. The power consumed by an agitator depends on its impeller shape, size, speed of rotation, fluid density, viscosity, vessel dimensions, internal attachments, and position of the impeller in this enclosed system. Since there is a possibility of a gas-liquid surface being distorted, as in the formation of a vortex, gravity forces must be considered.

Consider a stirred tank vessel having a Newtonian liquid of density  $\rho$  and viscosity  $\mu$  is agitated by an impeller of diameter  $D_A$ , rotating at a rotational speed  $N$ . Let the tank diameter be  $D_T$ , the impeller width  $W$ , and the liquid depth  $H$ . We can express the power  $P$  required for agitation of a single-phase liquid as

$$P = f(\rho^a, \mu^b, N^c, g^d, D_A^e, D_T^f, W^g, H^h) \quad (7-3)$$

There are nine variables and three primary dimensions, and therefore by Buckingham's theorem, Eq. (7-3) can be expressed by dimensionless groups. Employing dimensional analysis, Eq. (7-3) in terms of the three basic dimensions (mass M, length L, and time T) yields:

$$P = ML^2T^{-3}$$

Substitution of the dimensions into Eq. (7-3) gives

$$ML^2T^{-3} = f \left[ (ML^{-3})^a, (ML^{-1}T^{-1})^b, T^{-c}, (LT^{-2})^d, L^e, L^f, L^g, L^h \right] \quad (7-4)$$

Equating the exponents of M, L, and T on both sides of Eq. (7-4) gives

$$M: 1 = a + b \quad (7-5)$$

$$L: 2 = -3a - b + d + e + f + g + h \quad (7-6)$$

$$T: -3 = -b - c - 2d \quad (7-7)$$

From Eq. (7-5)

$$a = 1 - b \quad (7-8)$$

Substituting Eq. (7-8) into Eq. (7-6) gives

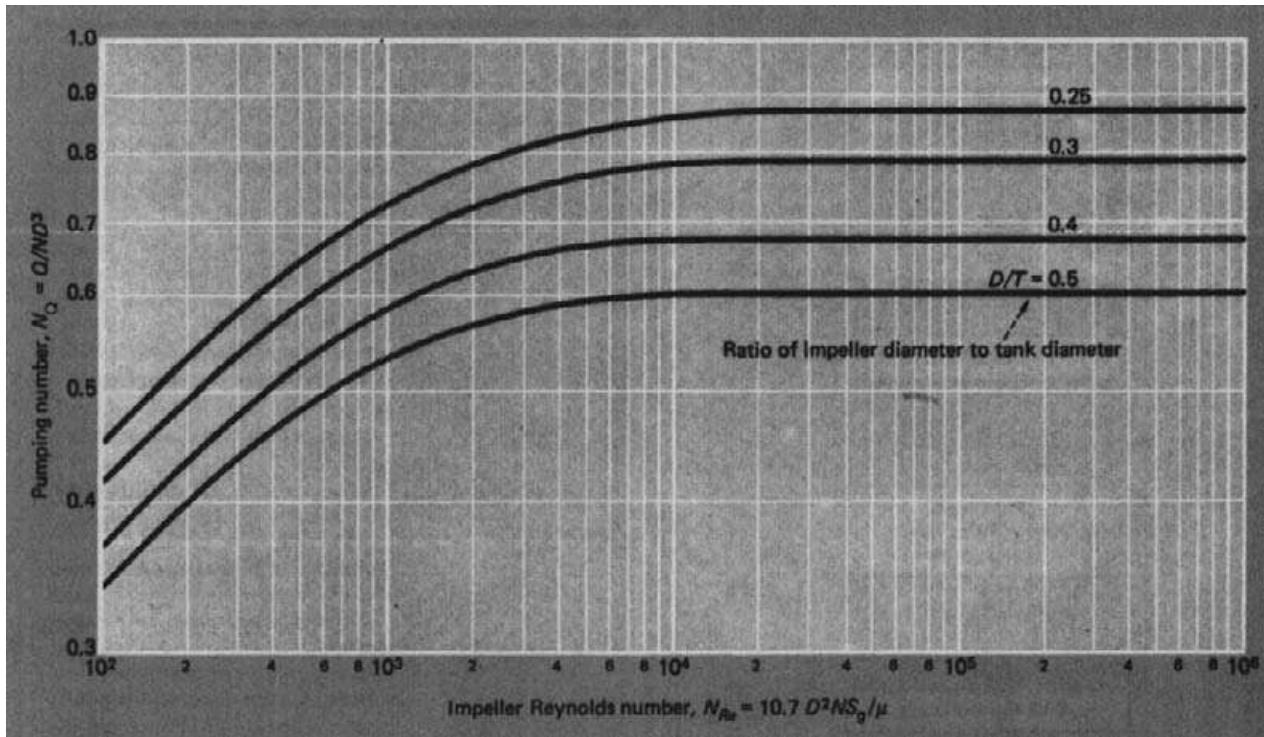
$$2 = -3(1 - b) - b + d + e + f + g + h$$

$$5 = 2b + d + e + f + g + h \quad (7-9)$$

From Eq. (7-7)

$$-3 = -b - c - 2d$$

$$b = 3 - c - 2d$$



**Figure 7-20** Pumping number is the basis for design procedures involving blending and motion. (By permission from Hicks, R.W. et al., *Chem. Eng.*, Apr 26, 1976, p. 104 [13].)

or

$$c = 3 - b - 2d \quad (7-10)$$

From Eq. (7-9)

$$e = 5 - 2b - d - f - g - h \quad (7-11)$$

Substituting a, c, and e on the right side of Eq. (7-3) yields

$$P = K (\rho^{1-b}, \mu^b, N^{3-b-2d}, g^d, D_A^{5-2b-d-f-g-h}, D_T^f, W^g, H^h) \quad (7-12)$$

Re-arranging and grouping the exponents yields

$$P = K \left[ \rho N^3 D_A^5 \left( \frac{\mu}{\rho N D_A^2} \right)^b \left( \frac{g}{N^2 D_A} \right)^d \left( \frac{D_T}{D_A} \right)^f \left( \frac{W}{D_A} \right)^g \left( \frac{H}{D_A} \right)^h \right] \quad (7-13)$$

or

$$\frac{P}{\rho N^3 D_A^5} = K \left[ \left( \frac{\mu}{\rho N D_A^2} \right)^b \left( \frac{g}{N^2 D_A} \right)^d \left( \frac{D_T}{D_A} \right)^f \left( \frac{W}{D_A} \right)^g \left( \frac{H}{D_A} \right)^h \right] \quad (7-14)$$

The dimensionless parameters are as follows:

$$\text{The power number, } N_p = \frac{P g_C}{\rho N^3 D_A^5}$$

$g_C$  = conversion factor

$$= 32.174 \frac{\text{lb}_m}{\text{lb}_f} \cdot \frac{\text{ft}}{\text{s}^2}$$

$$= 1 \text{ kg m/N s}^2$$

$$\text{The Reynolds number, } N_{Re} = \frac{\rho N D_A^2}{\mu}$$

$$\text{The Froude number, } N_{Fr} = \frac{N^2 D_A}{g}$$

Substituting these dimensionless numbers into Eq. (7-14) yields

$$N_p = K \left[ N_{Re}^{-b} N_{Fr}^{-d} \left( \frac{D_T}{D_A} \right)^f \left( \frac{W}{D_A} \right)^g \left( \frac{H}{D_A} \right)^h \right] \quad (7-15)$$

The power requirements cannot always be calculated for any system with a great degree of reliability. However, for those systems and/or configurations with known data, good correlation is the result. The relations are [23] as follows:

$$P = Q \rho H \quad (7-16)$$

where

$Q$  = Flow rate or pumping capacity from impeller  $\text{ft}^3/\text{s}$

$\rho$  = density,  $\text{lb}/\text{ft}^3$

$H$  = total potential head during flow, ft of liquid.

**TABLE 7-5 Baffled Cylindrical Tanks**

	$K_2$ Viscous	$K_3$ Turbulent
Propeller, 3-blade, pitch = diameter	41.0	0.32
Propeller, 3-blade, pitch = 2 diameters	43.0	1.00
Turbine, flat blade, 4 blades	70.0	4.50
Turbine, flat blade, 6 blades	71.0	6.30
Turbine, flat blade, 8 blades	72.0	7.80
Fan turbine, blades at 45°, 6 blades	70.0	1.65
Shrouded turbine, stator ring	172.50	1.12
Flat paddles, 2 blades (single paddle), $D/W = 4$	43.0	2.25
Flat paddles, 2 blades, $D/W = 6$	36.5	1.60
Flat paddles, 2 blades, $D/W = 8$	33.0	1.15
Flat paddles, 4 blades, $D/W = 6$	49.0	2.75
Flat paddles, 6 blades, $D/W = 6$	71.0	3.82

(Source: By permission from Rushton, R.H. and J.Y. Oldshue, *Chem. Eng. Prog.*, Vol. 49, No. 161, (1953).)

Viscous flow,  $N_{Re}$  less than 10–300 is expressed as:

$$P = \frac{K_2}{g_c} \mu N_s^2 D_A^3 \quad (7-17)$$

$K_2$  = from Table 7-5

$P$  = power, *not* power number,  $N_p(P_o)$ .

Fully developed turbulent flow,  $N_{Re}$  over 10,000, in a tank containing four equally spaced baffles having a width of 10% of the tank diameter:

$$P = \frac{K_3}{g_c} \rho N_s^3 D_A^5 \quad (7-18)$$

$K_3$  = from Table 7-5

where

$g_c$  = conversion factor, 32.174 lb (mass) ft/lb (force) s<sup>2</sup>

$H$  = total potential head during flow, ft

$P$  = power, ft lb<sub>f</sub>/s

$W$  = impeller blade width, ft

$\mu$  = viscosity, lb/ft s

$\rho$  = density, lb/ft<sup>3</sup>

$N_s$  = revolution per second, rps.

$$\text{Horsepower (HP)} = P/550 \quad (7-19)$$

$$\text{Impeller HP} = \frac{N_p N_m^3 D_i^5 S_g}{1.524 \times 10^{13}} \text{ Turbine} \quad (7-20)$$

Note: (HP) (33,000) = ft lb/min

(HP) (550) = ft lb/s

Symbols below [14]

Table 7-5 shows that in a cylindrical tank, four baffles, each 1/12 tank diameter above flat bottom, liquid depth is equal to tank diameter, impeller shaft is vertical and at centerline of tank.

The Reynolds number  $N_{Re}$  for mixing is (dimensionless) defined as:

$$N_{Re} = \frac{D_A^2 N_m \rho}{\mu} \quad (7-21)$$

where

$D_A$  = impeller diameter, m

$N_m$  = rotational speed, rps

$\rho$  = density, kg/m<sup>3</sup>

$\mu$  = fluid viscosity, Pa s.

or [14].

In Imperial units,

$$N_{Re} = \frac{(10.754 N_m D_i^2 S_g)}{(\mu')} \quad (7-22)$$

where

$N_m$  = impeller speed, rpm

$D_i$  = impeller diameter, in.

$\mu'$  = fluid viscosity, cP

$S_g$  = fluid specific gravity (not density).

Typical values [2]:

Turbulent mixing  $N_{Re} > 20,000$

Transition  $10 < N_{Re} < 20,000$

Laminar mixing  $N_{Re} < 10$

**Power number,  $N_p(P_o)$**

$$N_p = \frac{P}{\rho N^3 D_A^5} \quad (7-23)$$

where

$P$  = impeller power, W

$\rho$  = density, kg/m<sup>3</sup>

$N$  = rotational speed, rps

$D_A$  = impeller diameter, m.

In Imperial units,

$$N_p = \frac{1.524 \times 10^{13} P}{(S_g N^3 D_A^5)} \quad (7-24)$$

where

$P$  = impeller power, hp

$S_g$  = specific gravity

$N$  = rotational speed, rpm

$D_A$  = impeller diameter, in.

Typical values (depend on impeller type) [2]:

Low  $0.1 < N_p < 0.5$

For axial flow impellers such as propellers, hydrofoil impellers, and so on

Middle  $0.5 < N_p < 3.0$

For mixed flow impellers such as pitched-blade turbines, simple paddles, and so on

High  $3.0 < N_p < 7.0$

For radial flow impellers such as straight-blade turbines, disc-style turbines, and so on.

Figure 7-21 is used in most correlations to represent the relationship to system performance for turbulent flow in a baffled tank. For tanks containing no baffles, the fluid motion remains swirling and a vortex develops. These conditions are characterized by the lower curves in Figures 7-22 and 7-23, which include the Froude effect. This effect is not prominent in baffled tanks.

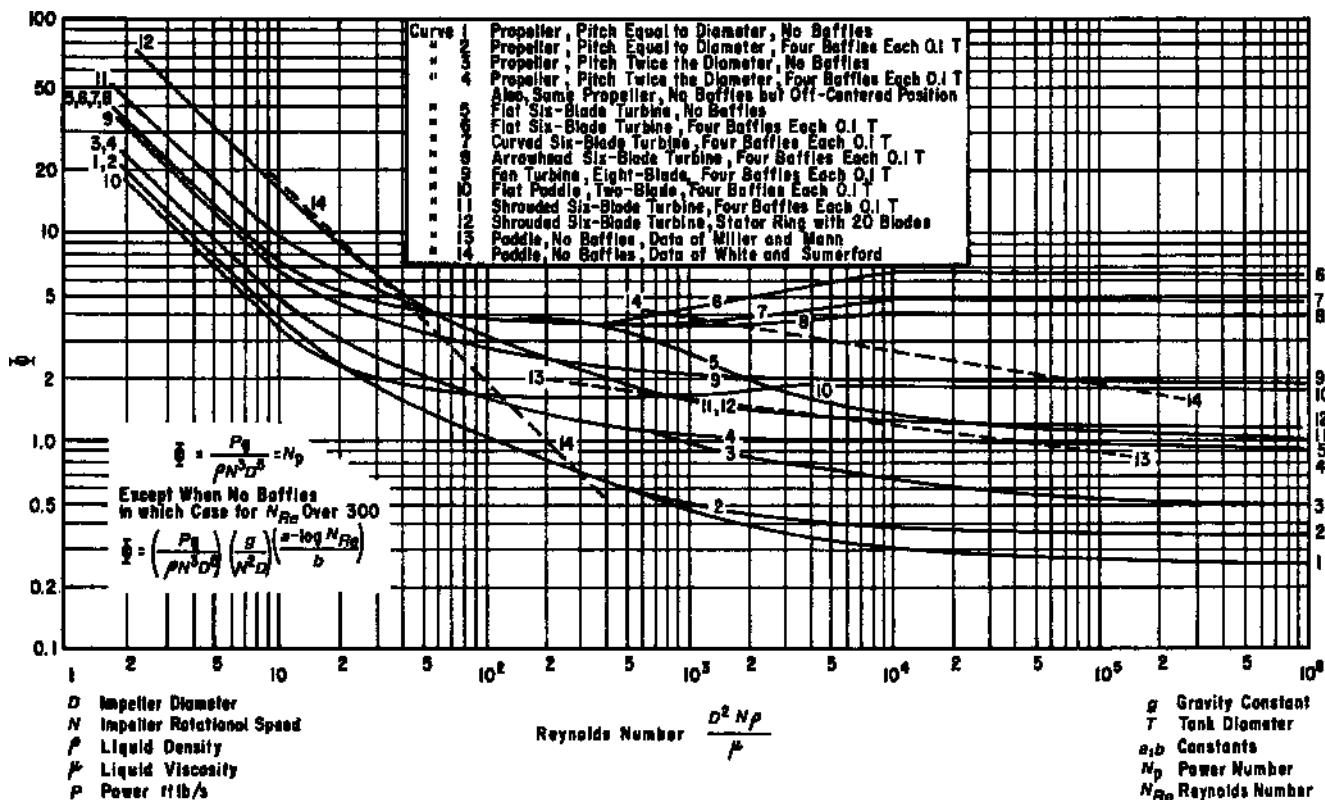


Figure 7-21 Power consumption of Impeller. (Source: Rushton, J.H. et al., *Chem. Eng. Prog.*, Vol. 46, Nos 8 and 9, 1950 [36].)

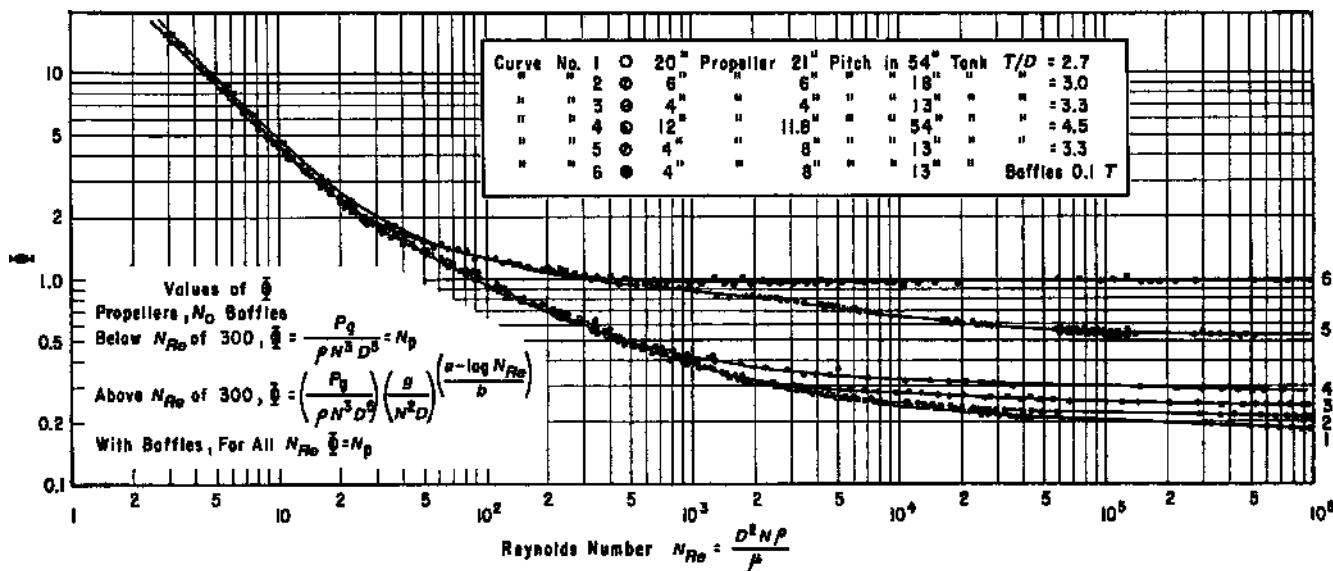


Figure 7-22 Reynolds number correlation for propellers. (By permission from Rushton, J.H. et al. [36].)

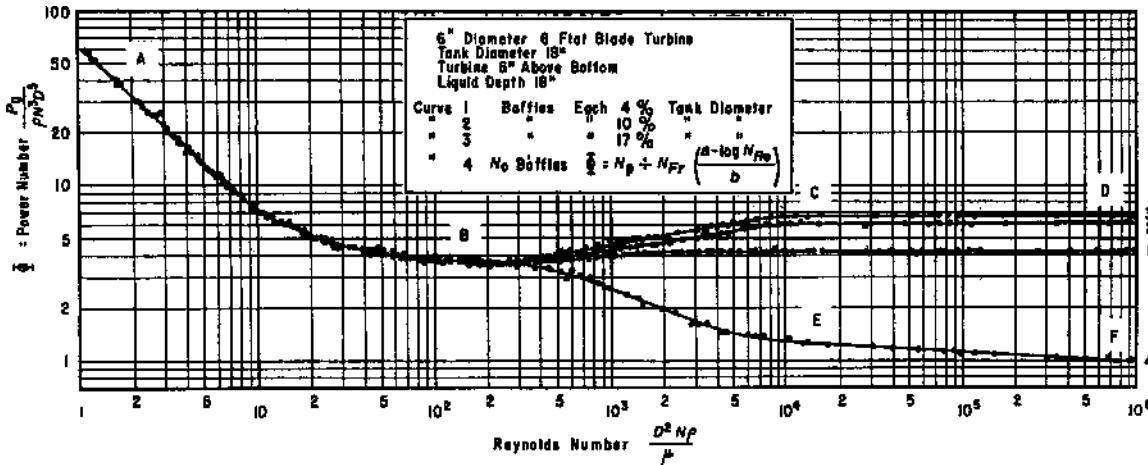


Figure 7-23 Reynolds number correlation for a flat-blade turbine. (By permission from Rushton, J.H. et al. [36].)

### EXAMPLE 7-1

Calculate the power for agitating a liquid of density  $950 \text{ kg/m}^3$  and viscosity of  $250 \text{ cP}$ , given the following configuration: Number of blades  $B = 6$ , diameter of the agitator  $0.61 \text{ m}$ , operating at  $90 \text{ rpm}$ . Other geometrical ratios are shown in Figure 7-16a. A disc-mounted flat-turbine is used.

*Solution*

The Reynolds number for mixing is

$$N_{Re} = \frac{\rho N D_A^2}{\mu}$$

$$N = \text{the number of rps is } \left( \frac{90}{60} \right) = 1.5 \text{ rps}$$

$$N_{Re} = \frac{(950)(1.5)(0.61^2)}{250 \times 10^{-3}} \quad \left\{ \frac{\text{kg}}{\text{m}^3} \cdot \frac{1}{\text{s}} \cdot \frac{\text{m}^2}{\text{ms}} \right\}$$

$$N_{Re} = 2121$$

Using curve 6 in Figure 7-21, the power number is  $N_p = 5.0$ . The theoretical power for mixing is

$$\begin{aligned} P &= N_p \rho N^3 D_A^5 \\ &= 5.0 \times 950 \times 1.5^3 \times 0.61^5 \quad \left\{ \frac{\text{kg}}{\text{m}^3} \cdot \frac{\text{rev}^3}{\text{s}^3} \cdot \text{m}^5 \right\} \\ &= 1353.99 \text{ W} \\ &= 1.35 \text{ kW}(1.82 \text{ hp}) \end{aligned}$$

(Note:  $1 \text{ kW} = 1.341 \text{ hp}$ )

An Excel spreadsheet program from the Companion website (Example 7-1.xls) has been developed to determine the theoretical power of any agitator type with given fluid physical properties and tank geometry.

**The Froude number is [16]:**

$$N_{Fr} = \frac{D_A N^2}{g} \quad (7-25)$$

$g = \text{acceleration due to gravity, } 32 \text{ ft/s}^2$

Estimated turbine impeller diameter [13]:

$$D_A = 394 \left[ \frac{(H_p)}{(n S_g N_m^3)} \right]^{1/5} \quad (7-26)$$

Calculate Reynolds number,  $N_{Re}$ , from Eq. (7-22), then correct for viscosity effects.

$$D_{cor} = D_A C_F \quad (7-27)$$

Representative  $C_F$  values from [13] are shown in the table below:

$N_{Re}$	$C_F, \text{ in.}$
700	1.0
400	0.98
200	0.95
100	0.91
70	0.89
60	0.88
50	0.87

In general, below a Reynolds number of 50, all impellers give viscous flow mixing; between 50 and 1000, the pattern is in the transition range; and from  $N_{Re}$  above 1000, the action is turbulent.

For  $N_{Re} \leq 10$ , the liquid motion moves with the impeller, and off from the impeller, the fluid is stagnant [37]. The Froude number accounts for the force of gravity when it has a part in determining the motion of the fluid. The Froude number must be equal in scale-up situations for the new design to have similar flow when gravity controls the motion [38].

Oldshue [14] points out that to identify the turbulent range as beginning at a specific  $N_{Re}$  may not be exactly correct, as it actually varies with different impeller designs. This range may vary from  $N_{Re} \cong 10^3$  to  $N_{Re} \cong 10^5$ , so for common use  $N_{Re} = 10^5$  is taken as the turbulent range for all impellers.

## SIMILARITY

Equality of all groups in Eq. (7-15) assures similarity between systems of different sizes. The types of similarity are geometric, kinematic, and dynamic. The last three terms of Eq. (7-15) represent the conditions of geometric similarity, which require that all corresponding dimensions in systems of different sizes have the same ratio to each other.

For geometric similarity, Eq. (7-15) becomes

$$N_p = K N_{Re}^{-b} N_{Fr}^{-d} \quad (7-28)$$

The constant  $K$  and the exponents  $b$  and  $d$  must be determined for the particular type of agitator, its size and location in the tank, the dimensions of the tank, and the depth of the liquid.

Kinematic similarity exists between two systems of different sizes when they are not only geometrically similar, but when the ratios of velocities between corresponding points in one system are equal to those in the other.

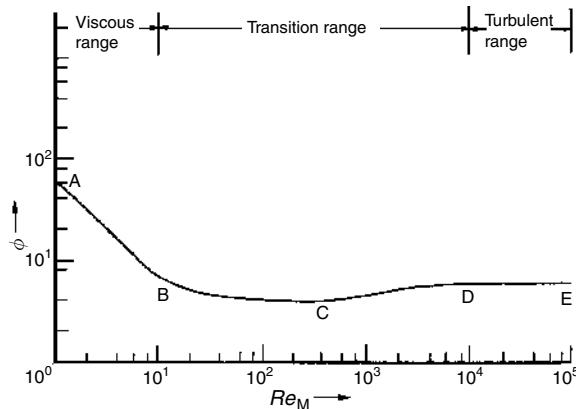
Dynamic similarity exists between two systems when, in addition to being geometrically and kinematically similar, the ratios of forces between corresponding points in one system are equal to those in the other.

The value of  $N_{Re}$  determines whether the flow is laminar or turbulent, and hence is a significant group affecting the power consumption. The Froude number  $N_{Fr}$  representing the ratio of inertial to gravitational forces is only significant when the liquid in the tank swirls to such an extent that a deep vortex is formed and the wave or surface effects become important. In an unbaffled vessel, a balance between the inertial and gravitational forces determines the shape of any vortex. The Power number  $N_p$  may be considered as a drag coefficient or friction factor.

Experimental data on power consumption are generally plotted as a function of the Power number  $N_p$  vs Reynolds number  $N_{Re}$ , that is, by rearranging Eq. (7-28).

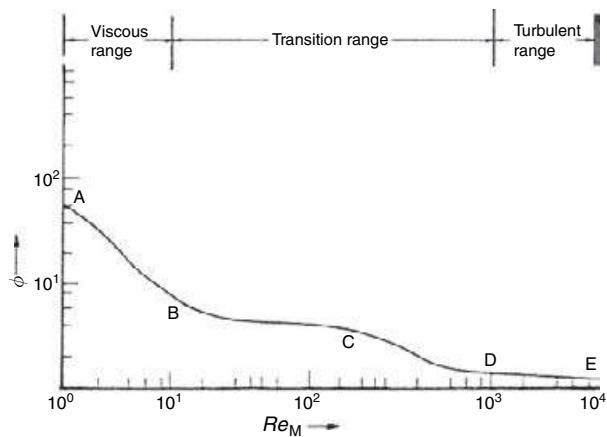
$$\Phi = \frac{N_p}{N_{Fr}^{-d}} = K N_{Re}^{-b} \quad (7-29)$$

For a fully baffled tank,  $b = 0$  and  $\Phi = N_p$ . A generalized plot of Eq. (7-29) is shown in Figure 7-24. The power correlation indicates three ranges of liquid motion: laminar (viscous), transition, and turbulent. The laminar or viscous range occurs below a Reynolds number of 10. The expected result of Power number being inversely

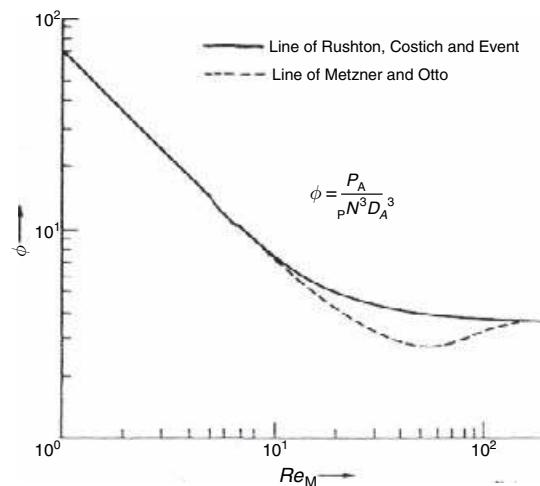


**Figure 7-24** Power curve for the standard tank configuration. (Source: Holland, F.A. and R. Bragg, *Fluid Flow for Chemical Engineers*, 2nd ed., Edward Arnold, 1995.)

proportional to the Reynolds number is also confirmed by experimental data. The Froude effects are unimportant and a logarithmic plot of the relation between Power number and Reynolds number gives a slope of  $-1$  in this range. Fully turbulent agitation occurs above a Reynolds number of 10,000. The range between these limits can be described as transition flow since flow patterns change depending on the Reynolds numbers. Figure 7-25 shows the Power number vs Reynolds number plot for the unbaffled system. Both Figures 7-24 and 7-25 are identical to point C where  $N_{Re} \cong 300$ . As the Reynolds number for mixing increases beyond point C in the unbaffled system, vortexing increases and the power number falls sharply. Figure 7-26 shows the Power number as a function of the Reynolds number for shear thinning fluids. The full line gives the Newtonian Power number obtained by Rushton et al. [36] for a flat-blade turbine system, while the dashed line shows Metzner and Otto's [39] plot for shear thinning liquids. Figure 7-26 illustrates that at no point is the shear thinning power curve higher than the Newtonian power curve. Therefore, the use of the Newtonian



**Figure 7-25** Power curve for the standard tank configuration without baffles. (Source: Holland, F.A. and R. Bragg, *Fluid Flow for Chemical Engineers*, 2nd ed., Edward Arnold, 1995.)



**Figure 7-26** Deviation from Newtonian power curve for shear thinning liquids. (Source: Holland, F.A. and R. Bragg, *Fluid Flow for Chemical Engineers*, 2nd ed., Edward Arnold, 1995.)

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power curve to determine the power will give a conservative value when used for shear thinning liquids.

Rushton et al. [36] performed extensive measurements of the power requirements on geometrically similar systems and found that for baffled tanks, the Froude number does not play any part in determining the power requirements, as vortices do not form in such systems. For unbaffled systems, the Froude number plays a part above  $N_{Re}$  of about 300. They reported  $N_p = 6.3$  in the turbulent range of 10,000. After extensive curve fitting of their experimental data, they found that a single curve is obtained for any particular unbaffled configuration. If we plot  $\Phi$  as a function of  $N_{Re}$  where  $\Phi$  is defined as

$$\Phi = N_p \quad \text{for } N_{Re} < 300 \quad (7-30)$$

$$\Phi = \frac{N_p}{N_{Fr}^{[(a-\log N_{Re})/b]}} \quad \text{for } N_{Re} > 300 \quad (7-31)$$

where  $a$  and  $b$  are constants for any configuration [36].

1. Propellers,  $p = D$ ;  $R = 0$   
 $a = 2.1$ ,  $b = 18$ ;  $B = 3$
2. Propellers,  $p = D$ ;  $R = 4$   
 $J = 0.1T$ ;  $B = 3$
3. Propellers,  $p = 2D$ ;  $R = 4$   
 $a = 1.7$ ;  $b = 18$ ;  $B = 3$
4. Propellers,  $p = 2D$ ;  $R = 4$   
 $J = 0.1T$ ;  $B = 3$
5. Flat-blade disc turbine,  
 $R = 0$ ;  $a = 1$ ;  $b = 40.0$
6. Flat-blade disc turbine,  
 $R = 4$ ;  $J = 0.1T$ ;  $B = 6$
7. Curved-blade disc turbine,  
 $R = 4$ ;  $J = 0.1T$ ;  $B = 6$
8. Arrow-head disc turbine,  
 $R = 4$ ;  $J = 0.1T$ ,  $B = 6$
9. Pitched-blade turbine,  
 $R = 4$ ;  $J = 0.1T$ ,  $B = 8$
10. Flat paddle,  
 $R = 4$ ;  $J = 0.1T$ ;  $B = 2$
11. Shrouded turbine,  
 $R = 4$ ;  $J = 0.1T$ ,  $B = 6$
12. Diffuser ring shrouded turbines stator ring  
having 20 blades  $B = 6$

Dickey and Fenic [16] observed that the impeller characteristics have significant influence on the Power number correlation.

The following are the two characteristics of Figure 7-21:

- 1) At low  $N_{Re} < 1.0$ ,  $N_p \propto \frac{1}{N_{Re}}$ , independent of the presence of baffles.
- 2) At high Reynolds, at which most mixing operations are carried out in practice, the Power number is constant, that is

$$N_p \propto \frac{P}{\rho N^3 D_A^5} = \text{constant} \quad (7-32)$$

Rushton et al. [36] carried out preliminary investigations on the effect of varying the tank geometrical ratios and the correlation of Power number with Reynolds number. At high Reynolds number, they inferred that

- $\Phi$  is relatively unchanged when  $D_T/D_A$  is varied from 2 to 7 for turbine- and propeller-agitated baffled systems.
- $\Phi$  is unchanged when  $H/D_A$  is varied from 2 to 4.
- $\Phi$  is unaltered when  $E/D_A$  is changed from 0.7 to 1.6.

- $\Phi$  changes to  $\Phi \propto (J/D_A)^{0.3}$  when  $J/D_T$  is changed from 0.05 to 0.17.
- $\Phi$  depends on the number of blades in the turbine impeller as:

$$\Phi \propto (B/6)^{0.8} \quad \text{if } B < 6 \text{ and } \Phi \propto (B/6)^{0.7} \quad \text{if } B > 6.$$

- If off-centered and inclined propellers without baffles or side-entering propellers without baffles are used, no vortex forms and the  $\Phi$  vs  $N_{Re}$  curve for the corresponding baffled tank can be used to estimate power requirements.

These conclusions are speculative and experimental curves must be generated if more than one geometrical ratio differs from the standard value.

The power consumed by an agitator at various rotational speeds and physical properties (e.g., viscosity and density) for a system's geometry can be determined from the Power number correlation. The procedure involves:

- Calculating the Reynolds number,  $N_{Re}$ , for mixing.
- Reading the Power number  $N_p$  from the appropriate curve, and calculate the power  $P$  given by

$$P = N_p \rho N^3 D_A^5 \quad (7-33)$$

or

$$P = \Phi \rho N^3 D_A^5 N_{Fr}^{[(a-\log N_{Re})/b]} \quad (7-34)$$

Equations (7-33) and (7-34) are the power consumed by the agitator. Additional power is required to overcome electrical and mechanical losses. A contingency of motor loading as a percentage (e.g., 85%) is added when selecting the motor. Equation (7-33) can also be rearranged to determine impeller diameter when it is desired to load an agitator impeller to a given power level.

For unbaffled tanks,  $\Phi$  is read from the charts using the constants  $a$  and  $b$  given in Figure 7-27. Figure 7-28 shows the power correlations for the Chemineer standard 4BP and HE-3 impellers as a function of  $D/T$  at a  $C/T$  of 1/3. Figure 7-29 is useful for the determination of horsepower during turbulent flow involving various impellers types, and Figure 7-30 is useful for laminar flow. Also see Figure 7-31.

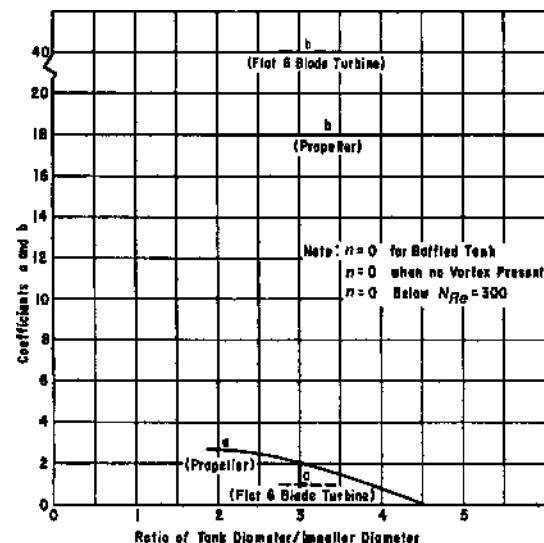
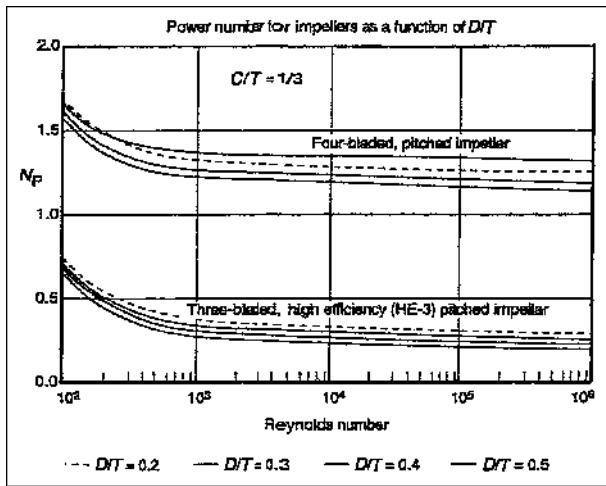


Figure 7-27 Factors in Froude number exponent (By permission, Rushton, J.H. et al. [36].)



**Figure 7-28**  $N_p$  vs  $N_{Re}$  for 4BP and HE-3 impellers as a function of  $D/T$  at  $C/T = 1/3$ . ( $D/T$  dependence: sequentially from top curve going down to bottom curve: 4BP - 0.5, 0.2, 0.4; HE-3 - 0.2, 0.3, 0.4, 0.5). (Chem. Eng., Aug 1984, p. 112.)

Flow and Power numbers each decrease as the Reynolds number increases. In unbaffled tanks, a vortex forms that takes over the flow regime and does not allow the usual relationship to describe the performance of the mixing operation. It is proper and good practice to provide baffles in all vessels.

At high  $N_{Re}$ , the Power number,  $N_p$ , stays reasonably constant, thus, viscosity has little effect on the power requirements. When moving to lower  $N_{Re}$  through the laminar region into the viscous region, the viscosity effect increases. In the laminar range [14]

$$N_p \propto 1/N_{Re}, N_{Re} < 50 \quad (7-35)$$

or

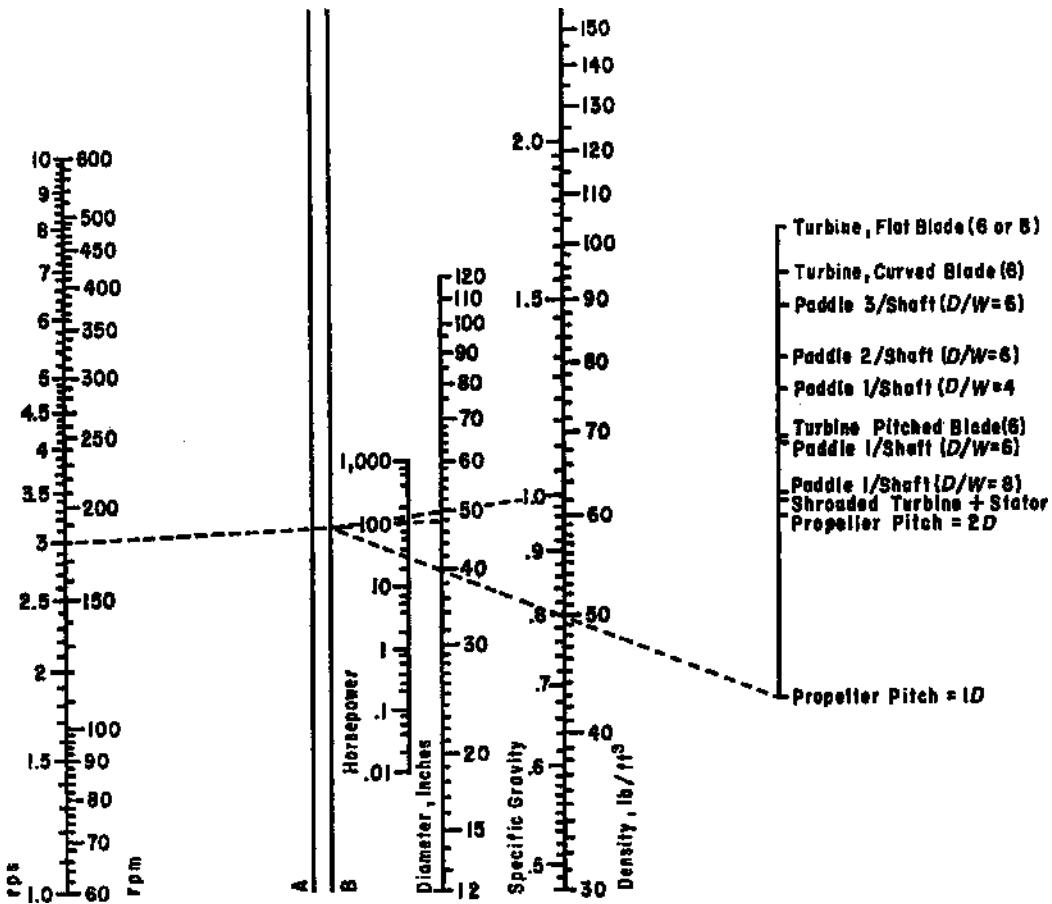
$$N_p \propto \mu \quad (7-36)$$

For  $50 < N_{Re} < 1000$  [14] is the transition range. In the immediate impeller area, the flow is fully turbulent; however, in the extremities of a vessel, the motion would be laminar. In this case, as in all others, the tank baffling is a major factor for performance of the system, the power, and flow results.

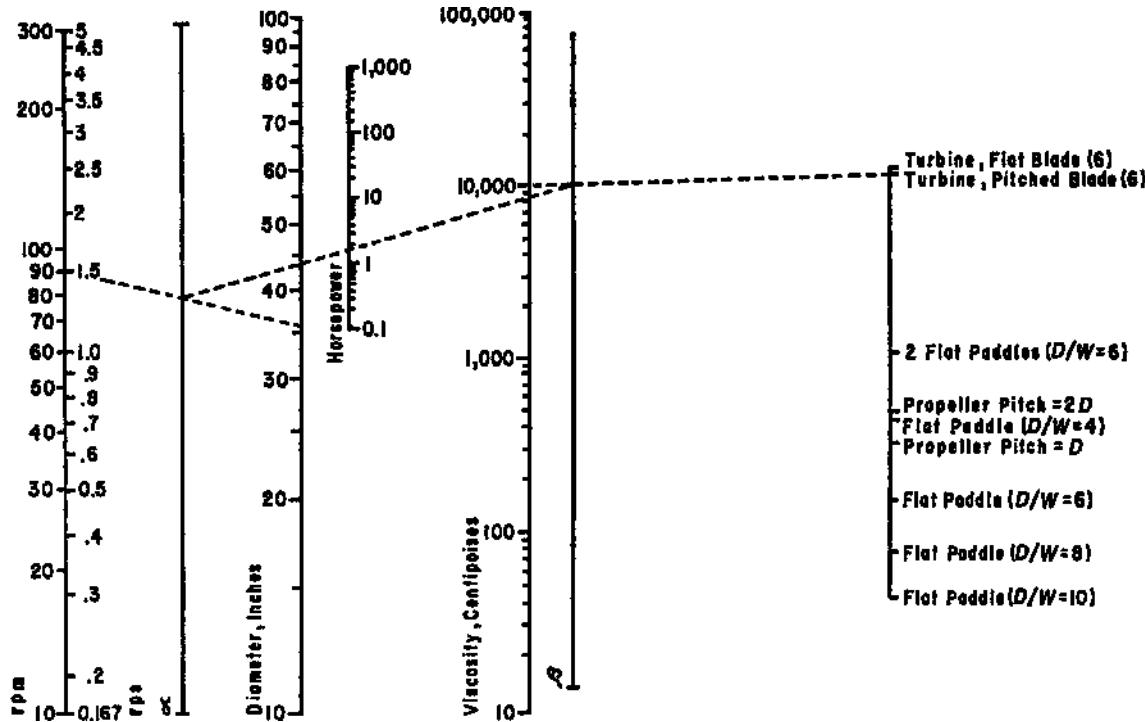
For  $N_{Re} > 1000$ , the properly baffled tank is turbulent throughout.  $N_Q$  and  $N_p$  are independent of  $N_{Re}$ . If the tank is not baffled, a "forced vortex" dominates the flow in the vessel.

For  $N_{Re} > 1000$ , the fully baffled tank is turbulent.

$$N_p = \frac{P}{\rho N^3 D_i^5} \quad (7-37)$$



**Figure 7-29** Power consumption by impeller type/dimensions for turbulent flow conditions. Knowing impeller type, diameter, speed, and batch density, connect RPM with diameter. The intersection with "A", connected to the density scale, makes an intersection on "B". A line from this point to the impeller scale intersects the horsepower scale at the correct value. (By permission from Cullen, C.S., Chem. Eng., Jun 1954, p. 177 [22].)



**Figure 7-30** Laminar flow mixing. For known impeller type, diameter, speed, and viscosity, this nomograph will give power consumption. Connect RPM and diameter, also viscosity and impeller scale. The intersection of these two separate lines with alpha and beta respectively is then connected to give horsepower on the HP scale. (By permission, Cullen, C.S., *Chem. Eng.*, Jun 1954, p. 177 [22].)

Pumping effectiveness or pumping per power is important for flow-controlled processes [14]. The shape, size, and baffling of a specific mixing vessel significantly influences the Reynolds number, flow, and Power numbers.

$$D_i = 394 \left( \frac{HP}{n S_g N_m^3} \right)^{1/5} \quad (7-38)$$

Other relationships [14] for one type of impeller (not different types) are given below:

$$\frac{Q}{P} = \frac{N_Q}{N_p} \left( \frac{1}{\rho N^2 D^2} \right), \text{ ratio of flow to power} \quad (7-39)$$

This is dependent on the impeller type, speed, diameter, and the geometry of the installation.

#### Torque, $\tau$

The torque delivered to the fluid by an impeller from its speed and power draw is defined as:

$$\tau = \frac{P}{2\pi N} = \frac{N_p \rho N^2 D_A^5}{2\pi} \text{ N m} \quad (7-40)$$

where

$\tau$  = torque, N m

$P$  = power, W

$N$  = rotational speed, rps.

In Engineering units,

$$\tau = \frac{63,025 P}{N} \quad (7-41)$$

where

$\tau$  = torque, in. lb

$P$  = power, hp

$N$  = rotational speed, rpm

$$\text{Torque } \tau = \frac{N_p (\rho N^2 D^5)}{2\pi}$$

$$\text{Lateral fluid forces on mixer } F = N_F \rho N^2 D^4 \quad (7-42)$$

For two different impellers, comparing the performance of power, flow, and flow per power at constant flow and speed [14] in terms of flow,  $Q$ , at speed,  $N$ :

$$\text{Fluid forces, } F = \frac{N_F (\rho N^{2/3} Q^{4/3})}{N_Q^{4/3}} \quad (7-43)$$

$$\text{Torque, } \tau = \frac{N_p \rho N^{1/3} Q^{5/3}}{N_Q^{5/3} (2\pi)} \quad (7-44)$$

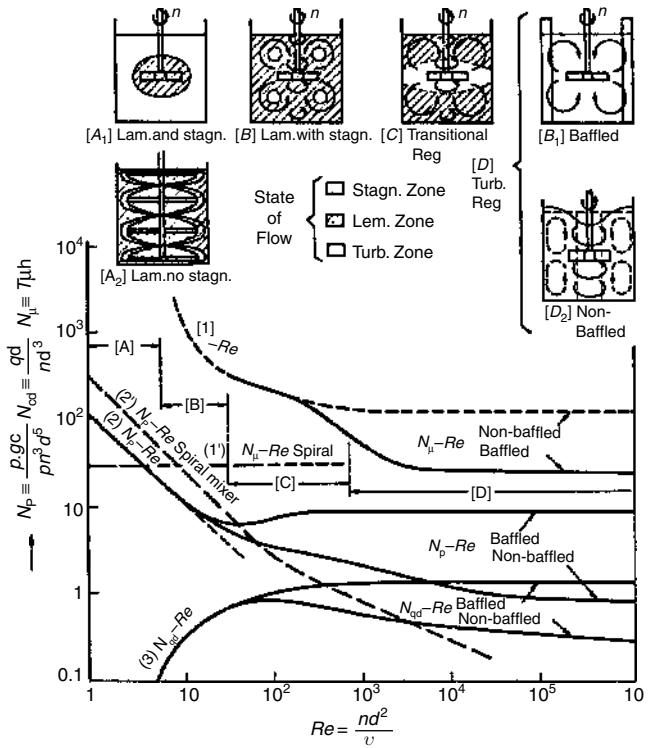
Flow per power input:

$$\frac{Q}{P} = \frac{\left( \frac{N_Q^{5/3}}{N_p} \right)}{(N^{4/3} Q^{2/3} \rho)} \quad (7-45)$$

Power, input, net to impeller shaft:

$$P = \left( \frac{N_p}{N_Q^{5/3}} \right) (\rho N^{4/3} Q^{5/3}) \quad (7-46)$$

$$\text{Diameter, } D = \frac{1}{N_Q^{1/3}} \left( \frac{Q^{1/3}}{N^{1/3}} \right) \quad (7-47)$$



$N_p$  = power number  
 $D$  = tank diameter  
 $d$  = impeller diameter  
 $Z$  = height of liquid in tank  
 $n$  = revolutions per minute  
 $T_M$  = time

$Re$  = Reynolds number =  $N_0 = d^2 \eta p / \mu$   
 $N_{qd}$  = discharge flow number =  $q_d / nd^3$   
 $p$  = pressure  
 $q_d$  = discharge flow rate  
 $\theta_m$  = mixing time

**Figure 7-31** Characteristic curves: flow, power, and mixing time ( $H/D = 1$ , 8-blade paddle  $H/D = 1/2$ ,  $d/D = 1/10$ ,  $H_D/D = 1/2$ ). (By permission, Nagata, S., *Mixing Principles and Applications*, Kodansha Scientific, Halsted Press, John Wiley & Sons, p. 125 [37].)

where using consistent units:

$P$  = impeller power draw,  $F L/t$  or  $ML^2/t^3$   
 $t$  = time  
 $L$  = length  
 $F$  = fluid force on turbine, perpendicular to shaft,  $ML/t^2$   
 $D$  = impeller diameter,  $L$   
 $Q$  = volumetric flow,  $L^3/t$   
 $T$  = tank diameter,  $L$   
 $\rho$  = fluid density,  $M/L^3$   
 $\mu$  = fluid viscosity,  $M/(Lt)$   
 $\tau$  = torque,  $FL$ , or  $ML^2/t^2$   
 $Z$  = liquid depth,  $L$   
 $N_p$  = Power number, dimensionless.

In Engineering units,

$$P_o \cong N_p \cong \frac{P g_c}{\rho N^3 D^5} \quad (7-48a)$$

or

In S.I. units

$$P_o \cong N_p \cong \frac{P}{\rho N^3 D^5} \quad (7-48b)$$

where

$P$  = Power, W  
 $\rho$  = fluid density,  $kg/m^3$   
 $N$  = revolution, rev/s  
 $D$  = impeller diameter, m  
 $N_Q$  = flow number, dimensionless  
 $g_c$  = conversion factor,  $32.174 \text{ lb}_m/\text{lb}_f \cdot \text{ft/s}^2$   
 $N_Q$  = flow number, dimensionless.

$$N_Q \cong \frac{Q'}{N_m D^3} \quad (7.2)$$

or

$$N_Q = \frac{1.037 \times 10^5 Q}{ND^3} \quad (7-49a)$$

$N_{Fr}$  = Froude number, dimensionless.

$$N_{Fr} = 7.454 \times 10^{-4} N^2 D \quad (7-49b)$$

where

$D$  = impeller diameter, in.  
 $N$  = rpm of impeller shaft.  
 $Q$  = volumetric pumping rate,  $ft^3/s$   
 $N_{Re}$  = Reynolds number, dimensionless.

$$N_{Re} = \frac{\rho ND^2}{\mu} \quad (7.21)$$

where

$$N_F = \text{force number}, N_F \cong F/(PN^2 D^4) \quad (7-50)$$

when

$M$  = mass  
 $L$  = length  
 $T$  = temperature  
 $t$  = time  
 $F$  = force,  $ML/t^2$ .

Oldshue [14] expresses pumping effectiveness as pumping per power and recognizes it as a key function for processes that are flow-controlled or need more flow than head or shear.

$$Q = N_Q ND^3 \quad (7-51)$$

$$P = N_p \rho N^3 D^5 \quad (7-33)$$

Then

$$\frac{Q}{P} = \left( \frac{N_Q}{N_p} \right) \left( \frac{1}{\rho N^2 D^2} \right) \quad (7-39)$$

When comparing flow (or pumping) per power, we determine that it is dependent on the impeller type, speed, diameter, and geometry of the installation. The mixer is not fully specified until torque,  $\tau$ , and lateral loads (fluid force,  $F$ ) are included in the analysis [14].

$$\tau = \frac{N_p \rho N^2 D^5}{(2\pi)} \quad (7-40)$$

$$F = N_F \rho N^2 D^4 \quad (7-52)$$

Table 7-6 presents the effects of expected performance on various parameters and relationships for mixing. To actually calculate a numerical result of comparing impeller performances, the dimensionless numbers for flow, power, and force are needed. Note that in Table 7-6 the constant basis is across the horizontal top of the chart and the function to be examined or compared is along the vertical left side. The functions in the body of the table are used as ratios for conditions (1) and (2), holding the basis constant.

For example, referring to Table 7-6, if power input,  $P$ , and impeller diameter,  $D$ , are kept constant, then speed,  $N$ , is proportional to  $1/N_p^{1/3}$ ; or holding flow,  $Q$ , and speed,  $N$ , constant then:

$$\frac{Q}{P} \propto \frac{N_Q^{5/3}}{N_p}$$

Thus

$$\frac{(Q/P)_1}{(Q/P)_2} = \frac{(N_Q^{5/3}/N_p^{3/4})_1}{(N_Q^{5/3}/N_p^{3/4})_2} \quad (7-53)$$

This is a valuable relationship as expressed in Table 7-6, because it expresses the working relationship between all the important variables. Note that as one variable changes, all others are changed. One variable cannot be changed alone without affecting the others.

**TABLE 7-6 Performance Relationships for Mixing Variables: More than one Variable Changing or Held Constant**

Function	Basis N,D	* Q, N	P, N, t	P, D	Q, P	Q, D
$N \sim$	1	1	1	$\frac{1}{N_p^{1/3}}$	$\frac{N_Q^{5/4}}{N_p^{3/4}}$	$\frac{1}{N_Q}$
$D \sim$	1	$\frac{1}{N_Q^{1/3}}$	$\frac{1}{N_p^{1/5}}$	1	$\frac{N_p^{1/4}}{N_Q^{3/4}}$	1
$Q \sim$	$N_Q$	1	$\frac{N_Q}{N_p^{3/5}}$	$\frac{N_Q}{N_p^{1/3}}$	1	1
$P \sim$	$N_p$	$\frac{N_p}{N_Q^{5/3}}$	1	1	1	$\frac{N_p}{N_Q^3}$
$t \sim$	$N_p$	$\frac{N_p}{N_Q^{5/3}}$	1	$N_p^{1/3}$	$\frac{N_p^{3/4}}{N_Q^{5/4}}$	$\frac{N_p}{N_Q^2}$
$* QP \sim$	$\frac{N_Q}{N_p}$	$\frac{* N_Q^{5/3}}{N_p}$	$\frac{N_Q}{N_p^{3/5}}$	$\frac{N_Q}{N_p^{1/3}}$	1	$\frac{N_Q^3}{N_p}$
$F \sim$	$N_F$	$\frac{N_F}{N_Q^{4/3}}$	$\frac{N_F}{N_Q^{4/5}}$	$\frac{N_F}{N_p^{2/3}}$	$\frac{N_F}{N_Q^{1/2} N_p^{1/2}}$	$\frac{N_F}{N_Q^2}$

(Source: By permission from Oldshue, "Fluid Mixing Technology," *Chem. Eng.*, McGraw-Hill Publications Co., Inc., 1983 [14].)

\* Example:  $Q/P$  is proportional to  $N_Q^{5/3}/N_p$  on a comparison basis of keeping flow  $Q$  and speed  $N$  constant.

### EXAMPLE 7-2

Calculate the theoretical power for a flat, six-blade turbine without baffles, but with standard tank configuration shown in Table 7-2. Use the same data as in Example 7-1.

*Solution*

Since the tank is unbaffled, the Froude number is a factor, and its effect is calculated from

$$N_{Fr} = \frac{N^2 D_A}{g}$$

$$= \frac{(1.5^2)(0.61)}{9.81} \quad \left\{ \frac{\text{rev}}{\text{s}^2} \cdot \frac{\text{m}}{\text{s}^2} \right\}$$

$$= 0.14$$

$$N_{Re} = 2121$$

The constants  $a$  and  $b$  for an unbaffled tank  $R = 0$  are  $a = 1.0$  and  $b = 40$ . Using curve 4 in Figure 7-23, the power number is  $N_p = 2.0$

$$N_{Fr}^m \text{ where } m = \frac{a - \log_{10} N_{Re}}{b}$$

$$m = \frac{1.0 - \log_{10} 2121}{40} = -0.0582$$

$$N_{Fr}^m = 0.14^{-0.0582} = 1.1212$$

Therefore

$$\begin{aligned} \text{Power } P &= N_p \rho N^3 D_A^5 N_{Fr}^m \\ &= 2.0 \times 950 \times 1.5^3 \times 0.61^5 \times 1.1212 \\ &= 607.24 \text{ W} \\ &= 0.61 \text{ kW}(0.81 \text{ hp}) \end{aligned}$$

A Microsoft Excel spreadsheet program from the companion website (Example 7-2.xls) has been developed for an unbaffled tank.

Studies on various turbine agitators have shown that geometric ratios that vary from the standard design can cause different effects on the Power number  $N_p$  in the turbulent regions [40].

- For the flat, six-blade open turbine,  $N_p \propto (W/D_A)^{1.0}$ .
- For the flat, six-blade open turbine, varying  $D_A/D_T$  from 0.25 to 0.5 has no effect on  $N_p$ .
- Where two six-blade open turbines are installed on the same shaft and the spacing between the two impellers (vertical distance between the bottom edges of the two turbines) is at least equal to  $D_A$ , the total power is 1.9 times a single flat-blade impeller. For two six-blade, pitched-blade ( $45^\circ$ ) turbines, the power is about 1.9 times that of a single pitched-blade impeller.
- A baffled vertical square tank or a horizontal cylindrical tank has the same Power number as a vertical cylindrical tank.

## 7.9 SCALE OF AGITATION, $S_A$

Hicks et al. [13] introduced the scale of agitation,  $S_A$ , as a measure for determining agitation intensity in pitched-blade impellers. The scale of agitation is based on a characteristic velocity,  $v$ , defined by

$$v = \frac{Q_p}{A_v} \quad (7-54)$$

where

$v$  = characteristic velocity, m/s

$A_v$  = cross-sectional area of the tank,  $\text{m}^2$

$Q_p$  = effective pumping capacity,  $\text{m}^3/\text{s}$ .

The characteristic velocity can be expressed as:

$$v \propto N_Q N D_A \left( \frac{D_A^2}{D_t^2} \right) \quad (7-55)$$

In geometrically similar systems, the characteristic velocity becomes

$$v \propto N_Q N D_A \quad (7-56)$$

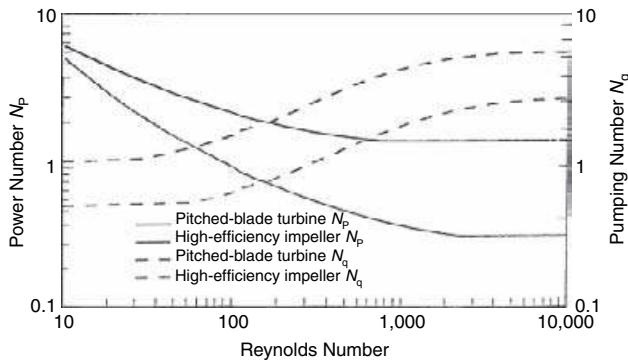
Thus, during geometric scale-up, the characteristic velocity can be held constant by holding  $N_Q N D_A$  constant.  $Q_p$  is determined from the Pumping number and Figure 7-32a.  $S_A$  is a linear function of the characteristic velocity and is determined by

$$S_A = 32.8v \quad (7-57)$$

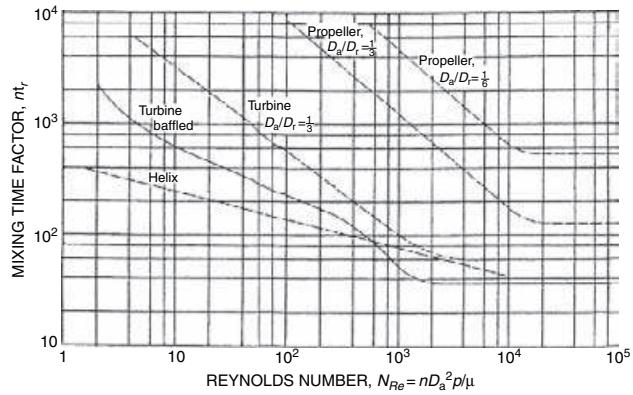
Accordingly, a value of  $S_A$  equal to 1 represents a low level and 10 shows a high level of agitation intensity. The 1–10 range of agitation intensity accounts for about 95%, or more of all turbine-agitation applications, enabling it to be suited for a wide range of process operations. Gates et al. [1] gave guidelines on how to relate  $S_A$  to specific process applications.

## 7.10 MIXING TIME CORRELATION

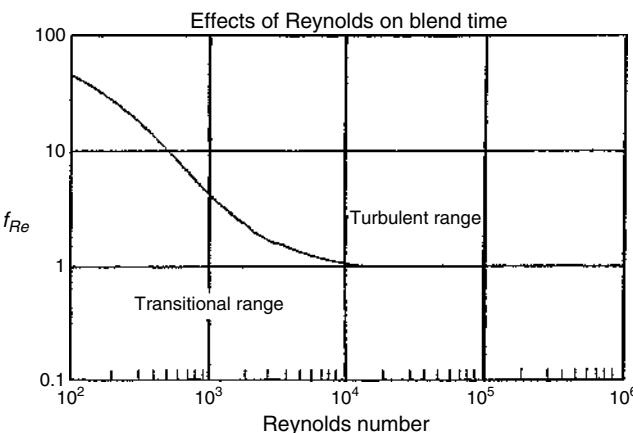
A distinction was made earlier between mixing and agitation. The third term in liquid mixing is blending. This refers to the intermingling of miscible fluids to produce some degree of uniformity. A criterion for good mixing may be visual. For example, it could be a particular color from two different color liquids, or the color change of an acid-base indicator to determine the liquid blending



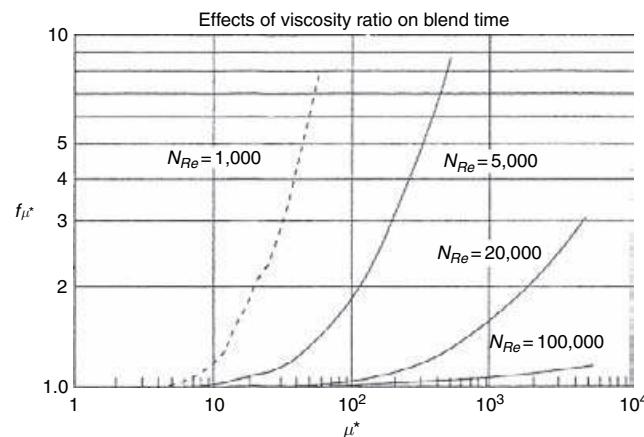
**Figure 7-32a** Power number and Pumping number as functions of Reynolds number for a pitched-blade turbine and high-efficiency impeller. (Source: Bakker, A. and L.E. Gates, “Properly Choose Mechanical Agitators for Viscous Liquids”, *Chem. Eng. Prog.*, pp. 25–34, 1995.)



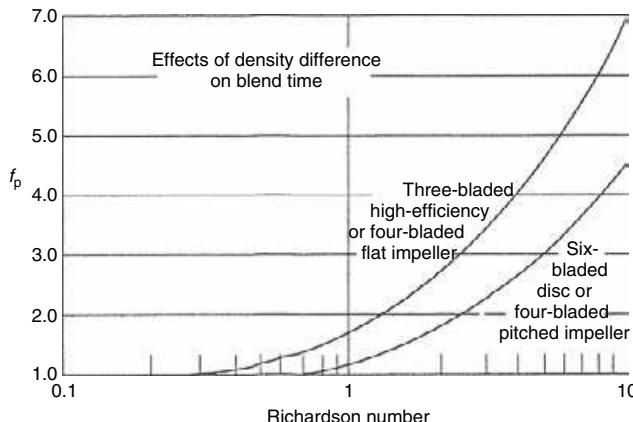
**Figure 7-32b** Mixing times in agitated vessels. Dashed lines represent unbaffled tanks; solid lines represent a baffled tank. (Source: McCabe, W.L. et al., *Unit Operations of Chemical Engineering*, 4th ed., McGraw-Hill Book Co., New York, 1985.)



**Figure 7-32c** Effect of impeller Reynolds number on blend time for  $N_{Re} < 10,000$ . (Source: Couper, J.R. et al., *Chemical Process Equipment – Selection and Design*, Elsevier Inc., 2005.)



**Figure 7-32d** Effect of viscosity ( $\mu^*$ ) ratio on blend time. (Source: Couper, J.R. et al., *Chemical Process Equipment – Selection and Design*, Elsevier Inc., 2005.)



**Figure 7-32e** Effect of density difference on blend time. (Source: Couper, J.R. et al., *Chemical Process Equipment – Selection and Design*, Elsevier Inc., 2005.)

**TABLE 7-7 Methods for Determining Blending Time**

Technique	Tracer	Blend Time Reached When
Grab sample	Any material that can be analyzed	Samples do not vary more than $\pm X\%$ from final concentration.
Dye introduction	Dyed fluid	Uniform color is attained
Conductivity cell	Concentration of salt solution	Measured conductivity that represents concentration is within $\pm X\%$ of final concentration
Acid-base Indicator	Acid (or base)	Neutralization is complete as determined by color change of indicator

(Source: Dickey D.S., "Dimensional analysis for fluid agitation system", *Chem. Eng.*, Jan 5, 1976.)

times. Characterization of blending in agitated vessels is usually in terms of mixing time. This is the time required to achieve some specified degree of uniformity after introduction of a tracer. Table 7-7 gives various techniques for determining blending time.

Each technique measures a different degree of uniformity; therefore, the time required for blending may vary from one method to the other. The correlation of blending time as derived from dimensional analysis is applicable to all techniques. Uhl and Gray [31] have summarized many of the experiments and correlations on blending and mixing time. For a given tank and impeller or geometrically similar systems, the mixing time is predicted to vary inversely with the stirrer speed, as confirmed in various studies [41–44]. Figure 7-32b shows plots of mixing time ( $N_t$ ) against the Reynolds number  $N_{Re}$  for several systems and Figure 7-32c illustrates the effect of impeller Reynolds number on blend time for  $N_{Re} < 10^4$ . An example, a turbine with  $D_A/D_T = 1/3$  and  $D_T/H = 1$ , the value of  $N_t$  is 36, for  $N_{Re} > 10^3$ , compared with a predicted value of 38.

Prochazka and Landau [45] developed a mixing time correlation for a single Rushton turbine impeller in a baffled tank in the standard configuration for  $N_{Re} > 10^4$ :

$$N_t = 0.905 \left( \frac{D_T}{D_A} \right)^{2.57} \log \left( \frac{X_o}{X_c} \right) \quad (7-58)$$

For a propeller, the mixing time is given by:

$$N_t = 3.48 \left( \frac{D_T}{D_A} \right)^{2.05} \log \left( \frac{X_o}{X_c} \right) \quad (7-59)$$

For a pitched-blade turbine, the mixing time is:

$$N_t = 2.02 \left( \frac{D_T}{D_A} \right)^{2.20} \log \left( \frac{X_o}{X_c} \right) \quad (7-60)$$

where

$X_o$  = initial value of the degree of inhomogeneity which varies between 1 and 3. A value of 2 is recommended.

$X_c$  = final integral mean value of the local degree of inhomogeneity and is defined as:

$$X_c = \left[ \frac{C(t) - C_x}{C_x - C_i} \right] \quad (7-61)$$

where

$C(t)$  = instantaneous concentration

$C_i$  = initial concentration

$C_x$  = final concentration.

$X_c = 0.05$  for most configurations. Moo-Young et al. [46] correlated their mixing results from

$$N_t = KN_{Re}^a \quad (7-62)$$

where  $K = 36$  and  $a = 0$  for turbines in baffled tanks for  $1000 < N_{Re} < 10^5$ .

Sano and Usui [47] developed an expression for mixing times by tracer injection for turbines as:

$$N_t = 3.8 \left( \frac{D_A}{D_T} \right)^{-1.80} \left( \frac{W}{D_T} \right)^{-0.51} n_p^{-0.47} \quad (7-63)$$

where

$n_p$  = number of blades.

Gray [48] found that the mixing times of helical ribbon impellers to be of the form

$$N_t = 30 \quad (7-64)$$

where

$N$  is the rotational speed of the helical ribbon impeller

$t$  is the batch mixing time.

Fasano et al. [49] expressed the blend time for turbulence conditions in a standard baffled tank (i.e.,  $N_{Re} > 10,000$ ) as:

$$t_{99} = \frac{4.065}{aN \left( \frac{D_A}{D_T} \right)^b \left( \frac{D_T}{H} \right)^{0.5}} \quad (7-65)$$

where  $a$  and  $b$  are the mixing rate constants. Table 7-8 shows values of  $a$  and  $b$  for different impeller types. The constants are for surface addition; however, blend times for similar fluids are relatively insensitive to addition location. Equation (7-65) is limited to the following:

- Newtonian fluids of nearly the same viscosity and density as the bulk fluid

**TABLE 7-8 Mixing Rate Constants for fully Turbulent Flow Regimes ( $N_{Re} > 10,000$ )**

Impeller Type	a	b
Six-bladed disc	1.06	2.17
Four-bladed flat	1.01	2.30
Four-bladed 45° pitched	0.641	2.19
Three-bladed high efficiency	0.272	1.67

(Source: Fasano et al., "Advanced Impeller Geometry Boosts Liquid Agitation", *Chem. Eng.*, Vol. 102, No. 8, Aug 1994.)

- Additions of 5% or less, of the fluid volume
- Additions made to a vessel already undergoing agitation (blend times of stratified fluids can be considerably longer).

The estimated blend time for 95% uniformity ( $t_{b,95\%}$ ) using a standard double flight, helical ribbon impeller with  $P_i/D_A = 1$ ,  $W/D_A = 0.1$ , and  $D_A/D_T = 0.96$  is given by

$$\text{For } N_{Re} \leq 100 : t_{b,95\%} = \frac{75}{N} \quad (7-66)$$

For anchor impellers of standard geometry ( $W/D_A = 0.1$ ,  $D_A/D_T = 0.98$ , and  $H/D_T = 1.0$ ), the estimated  $t_b$  for  $100 < N_{Re} < 10,000$  is given by

$$t_{b,95\%} = \exp(12.9 N_{Re}^{-0.135}) \quad (7-67)$$

where

$D_A$  = impeller diameter, m

$D_T$  = tank diameter, m

$H$  = impeller or helix height, m

$N$  = impeller rotational speed,  $s^{-1}$

$P_i$  = pitch of a helical ribbon impeller, m

$W$  = blade width, m.

Bakker and Gates [50] compared both Eqs (7-66) and (7-67) and inferred that at a Reynolds number of 100, it will take an anchor impeller more than 13 times as long to achieve 95% uniformity as a helical ribbon impeller operating at the same speed. These impellers require cooling to remove the excess heat due to their high power input. The mixing time that was considered relates to tanks operating in closed systems (e.g., batch reactors). In a continuous feed tank, the mixing time is generally shorter than in a closed tank. Figures 7-32d and e show the effect of physical properties such as viscosity and density on blend time.

## 7.11 SHAFT

The proper size of shaft is very important to avoid whip and vibration, destruction of bearings, gears, and damage to the vessel. The manufacturers usually take a conservative view of this problem; nevertheless it is well to understand the expected operating conditions for any installation. Normally, an impeller-shaft system should operate at about 40% of the critical speed. However, the turbine with bottom-side stabilizer can go as high as 80% of critical speed. The manufacturer should provide this information for the specific system.

## 7.12 DRIVE AND GEARS

Most mixers are driven by electric motors or in some cases by mechanical turbines, with gears ratioed to give the proper performance speed of the impeller. A variable or two-speed driver or

gear system often proves worth the extra cost, since it is difficult to predict the exact speed requirements for some new installations. This is particularly true in continuous chemical processes where the general nature of the fluid remains constant but the viscosity, density, or solid particle content may change as the plant progresses from "just erected" to steady production and even on to new and different products. This transition may take from a few months to several years and should be economically evaluated. The gear mechanism is not a place to reduce costs for this equipment, since improper application can create costly maintenance.

Usual practice, particularly for good estimating, is to assume that the gear drive requires 5% of the impeller horsepower and that system "surging or variations" require a minimum of 10% of this impeller horsepower. Thus

$$\text{Actual motor hp} = \text{impeller required hp}/0.85 \text{ (minimum)}$$

(7-68)

When the actual *maximum* gear box horsepower is known from the manufacturer, it should be used as long as it is equal to or greater than the 5% allowance noted above. The impeller/fluid horsepower allowable variation should still be 10% or greater. For example, if the calculated required motor drive (or turbine drive) hp = 23 (i.e., 19.55/0.85), the next *standard* motor is 25 hp; so use this, but never less than the 23 indicated above because 23 hp is non-standard, and no such motor hp exists.

Figure 7-33 illustrates a vertical propeller mixer assembly, with vertical mounting with gear box and motor. Figure 7-34 is a typical right angle, vertical impeller shaft with horizontal gear and motor drive.

The mixer manufacturer should always be consulted for proper mechanical features design and strength characteristics, such as horsepower, gear rating AGA, shaft diameter, shaft deflection, critical speeds, bottom steady bearing, and side-shaft bearings.

## 7.13 STEADY BEARINGS

The installation of mixers on long shafts in tall tanks may become a problem if "whip" of the shaft develops. To reduce this possibility, a bearing support in the bottom of the tank will hold the shaft steady. Lubrication is by the tank fluid. Therefore this has limited application if abrasive particles are present. Normally the manufacturers' designs avoid this extra bearing. Sometimes a guide bearing is installed about midway in the tanks to steady the shaft at this point. Again it is preferable to avoid this, if possible, and the manufacturer should make recommendations for the installation.

## MATERIALS OF CONSTRUCTION

In general, just about any material that can be worked into the impeller design is available, including steel, stainless alloys, copper alloys, nickel and alloys, hard rubber, and lead, rubber, and plastic coatings on impellers and shafts.

## DESIGN

Normally the proper impeller selections and horsepower requirements are handled in a cooperative manner with the manufacturer of this equipment in order to obtain the best analysis of a given application. There is no substitute for performing the proper test runs to evaluate one or more types of impellers in a particular application. Even if this is carried out on a small scale, proper evaluation can lead to the correct selection of impeller and horsepower. The horsepower seems to be the factor that is missed in



**Figure 7-33** Portable Vektor™ vertical propeller mixer assembly. (By permission from Lightnin (formerly Mixing Equipment Co.), a unit of General Signal.)

some evaluations. Any foreseeable process changes, or ranges of operation, must be considered in order to have sufficient power for start-up as well as normal running.

#### SPECIFICATIONS

The suggested specification sheet of Figure 7-35 is helpful as a general checklist for the mixing inquiry and can be used in setting forth the known as the desired information with a manufacturer. Generally, the specification sheet should not be expected to establish the whole story or information on a mixing problem, unless the problem is known to be fairly straight forward or data is known which can be given to the manufacturer (e.g., blending, dispersing or dissolving crystals, etc.). For the unique problems, one-of-a-kind, laboratory data should be taken under the guidance of technical advice from the manufacturer or other qualified authority in order that adequate scale-up data will be taken and evaluated. It is important to both describe and give dimensions for the vessel to be used, or request the mixer manufacturer to recommend the type best suited to the service.

#### 7.14 DRAFT TUBES

The application of draft tubes as related to various mixing operations is shown in Figures 7-19a and 7-36a–i. The draft tubes are basically a tube or shell around the shaft of the mixer including the usual axial impeller, which allows a special or top-to-bottom fixed flow pattern to be set up in the fluid system. The size and location of the tube are related to both the mechanical and mixing performance characteristics as well as peculiar problems of the system. Usually they are used to ensure a mixing flow pattern that cannot or will not develop in the system. Weber gives the following points for draft tubes [51]:

With a draft tube inserted in a tank, no sidewall baffles are required, and the flow into the axial impeller mounted inside the tube is flooded to give a uniform and high flow pattern into the inlet to the impeller. The upflow in the annulus around the tube has sufficient velocity to keep particles in suspension, if necessary.

##### A. Increase mixing efficiency

1. Prevent short-circuiting of fluid; define a specific path
2. Improve heat transfer coefficient by forcing flow past coil surfaces
3. Provide more complete reaction in a gas–liquid system by recirculation of unreacted gases
4. Minimize areas of inadequate turbulence in vessel
5. Accentuate the direct mechanical shearing action of the mixing impeller upon the fluid
6. Amplify mixing action by effectively increasing the ratio of mixer to container diameter.

##### B. Decrease design problems

1. Reduce required shaft diameter and length, while maintaining complete mixing effectiveness
2. Limit or eliminate the need for submerged or internal guide bearings.

#### 7.15 ENTRAINMENT

Entrainment is an important element in the mixing operation and involves incorporation of low velocity fluid into the mass of the fluid stream or jet issuing from a source such as a mixing impeller. The axial flow from a propeller under proper physical conditions serves as a circular cross-section jet to produce mixing by turbulence and entrainment. The flat-blade turbine issues a jet for entrainment at the top and bottom areas of the ring [27]. It is significant to estimate the relative amount of liquid involved due to entrainment, as this helps to describe the effectiveness of the operation.

From a propeller, the entrainment by circular jet is [52] given by:

$$Q_e = Q \left[ 0.23 \left( \frac{X}{D_o} \right) - 1 \right] \quad (7-69)$$

where

$Q_e$  = volume entrained,  $\text{ft}^3/\text{s}$

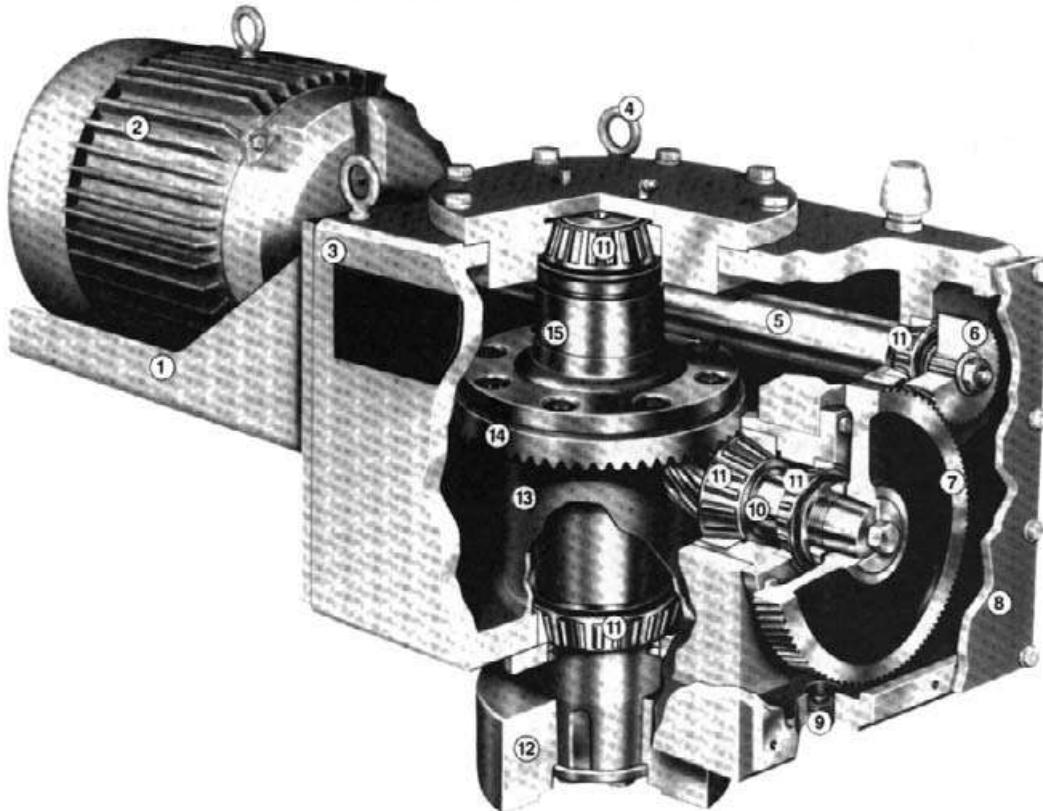
$X$  = distance from impeller source, not to exceed 100 jet diameters, ft

$D_o$  = diameter of jet at origin, ft.

This relation is sufficiently accurate for large-scale design.

The maximum  $Q/P^{1/3}$  for a circular jet is at  $X = 17.1D_o$  [23, 52] or, in other words, the optimum jet origin diameter is 1/17.1 of the distance desired for effective entrainment. Since the entrainment

- |                                |                                   |                                       |
|--------------------------------|-----------------------------------|---------------------------------------|
| 1. Motor bracket               | 6. Change pinion                  | 11. Timken tapered roller bearings    |
| 2. Standard foot-mounted motor | 7. Change gear                    | 12. Removable low-speed coupling half |
| 3. Fabricated housing          | 8. Change gear cover              | 13. Dry well oil seal                 |
| 4. Lifting eyes                | 9. Drain plug                     | 14. Spiral bevel gear                 |
| 5. High speed shaft            | 10. Spiral bevel pinion cartridge | 15. Low-speed shaft                   |



**Figure 7-34** Right angle drive for vertical impeller shaft. (By permission from Chemineer, Inc. Bulletin 711.)

efficiency does not fall off too rapidly, it is not necessary to use only the ratio given, but rather to stay in close proximity, say  $\pm 25$  to 35%. Large diameter jet streams are more effective for the same power than small streams [33]. Data on flat-blade turbines have not been fully evaluated.

## 7.16 BATCH OR CONTINUOUS MIXING

Often pilot plant or research data for developing a process are obtained on a batch operation. Later, a continuous process will usually prove that smaller equipment can be used and that the operation will be more economical. Normally batch mixing requires 10–25% more power than continuous [14] for stable conditions; however, the reaction time for continuous flow is always longer than the reaction time for batch flow, but the practical result may show batch time cycle is increased by filling, cleaning, and emptying the reactor (Figures 7-37a and b).

In batch operations, mixing takes place until a desired composition or concentration of chemical products or solid/crystals is achieved. For continuous operation, the feed, intermediate, and exit streams will not necessarily be of the same composition, but the objective is for the end/exit stream to be of constant composition as a result of the blending, mixing, chemical reaction, solids suspension, gas dispersion, or other operations of the process. "Perfect" mixing is rarely totally achieved, but represents the instantaneous

conversion of the feed to the final bulk and exit composition (Figure 7-38).

When conducting pilot plant testing to develop a process involving mixing, which later may be used in the design of a large-scale plant, it is wise to discuss the testing with a mixing specialist and outline the needed pilot data required to later scale-up the process, generally from batch pilot plant to continuous commercial process.

## SCALE-UP AND INTERPRETATION

Scale-up techniques for using the results of pilot plant or bench scale test work to establish the equivalent process results for a commercial or large-scale plant mixing system design require careful specialized considerations and usually are best handled by the mixer manufacturer's specialist. The methods to accomplish scale-up will vary considerably, depending on whether the actual operation is one of blending, chemical reaction with product concentrations, gas dispersions, heat transfer, solids suspensions, or others.

These scale-up methods will necessarily at times include fundamental concepts, dimensional analysis, empirical correlations, test data, and experience [53].

Similarity concepts use physical and mathematical relations between variables to compare the expected performance of

**486 MIXING OF LIQUIDS**

Job No. \_\_\_\_\_

SPEC. DWG.NO.

B/M No. \_\_\_\_\_

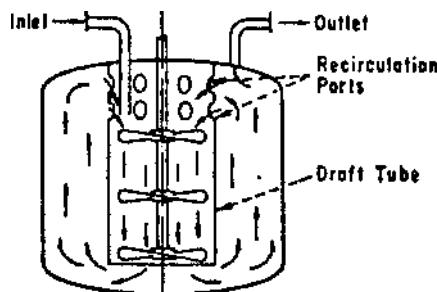
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**MIXING EQUIPMENT SPECIFICATIONS**

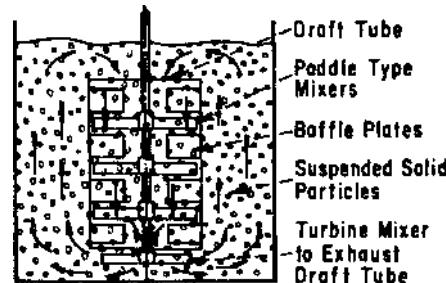
PERFORMANCE				
Component	Wt.%	Sp. Gr.	Viscosity, Cp	Temp. °F
Sp. Gr. of Mixture	Viscosity of Mixture			
Solids : <input type="checkbox"/> Soluble <input type="checkbox"/> Insoluble <input type="checkbox"/> Abrasive <input type="checkbox"/> Sticky <input type="checkbox"/> Crystalline <input type="checkbox"/> Fluffy	Lbs/Gal. Mixture      Sp. Gr.      Particle Size			
Suspension Sp. Gr.	Settling Velocity _____ FPM			
Class : <input type="checkbox"/> Blending <input type="checkbox"/> Dissolving <input type="checkbox"/> Suspending Solids <input type="checkbox"/> Cooking <input type="checkbox"/> Emulsifying <input type="checkbox"/> Heat Transfer <input type="checkbox"/> Gas Dispersion	<input type="checkbox"/> _____ Time Req'd _____			
Mixing Type : <input type="checkbox"/> Violent <input type="checkbox"/> Medium <input type="checkbox"/> Mild      Foam: <input type="checkbox"/> Slight <input type="checkbox"/> Average <input type="checkbox"/> Bad				
Cycle: <input type="checkbox"/> Batch:      Smallest _____ Gal.      Normal _____ Gal.      Largest _____ Gal.				
<input type="checkbox"/> Continuous:      Rate _____ GPM				
Mixer <input type="checkbox"/> Will <input type="checkbox"/> Will Not      be operated during filling.	Sequence of Addition _____			
Vessel Specs: Dwg. No. _____				
MATERIALS OF CONSTRUCTION				
Vessel _____	Shaft _____	Impeller _____		
Mtg. Flg. _____	Stuffing Box _____	Steady Bearings _____		
Packing _____	Other Wetted Parts _____			
SELECTION				
Manufacturer _____	Model _____			
Req'd Vessel Opening Size _____	Pressure Class _____ Facing _____			
Mixer Location on Vessel _____				
Mixer Angle _____ ° with _____				
DESIGN				
Impeller: Diameter _____	Type _____	No. _____	Speed _____	RPM _____
Normal BHP (Excluding Gear) _____				
Type Bearings _____	Steady Bearing Req'd?	<input type="checkbox"/> Yes	<input type="checkbox"/> No.	Guide Bearings Req'd? <input type="checkbox"/> Yes <input type="checkbox"/> No
Shaft Seal: <input type="checkbox"/> Packing <input type="checkbox"/> Mechanical.	Make _____	Type _____		
Seal Coolant _____	Stuffing Box Lubrication _____			
Shaft Coupling:      Type _____	Make _____			
Gear: Manufacturer _____	Type _____			
Size _____	Red. Ratio _____	Rated H.P. _____	Max. BHP _____	
Mech. Elf. _____ %.	No Reductions _____	Output _____	RPM Spec'd Changable <input type="checkbox"/> Yes <input type="checkbox"/> No	
Driver: Manufacturer _____	Type _____	Speed _____	RPM _____	
Elect. Power: _____ Volts _____	Phase _____	Cycle. PHB _____		
Service Factor _____	Frame _____			
REMARKS				
By _____	Chkd' _____	App. _____	Rev. _____	Rev. _____

P.O. To: \_\_\_\_\_

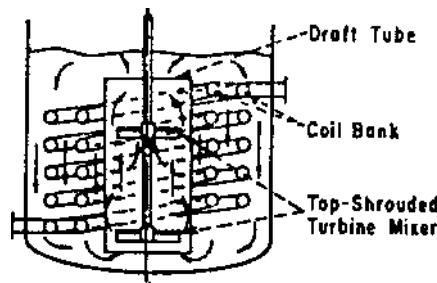
**Figure 7-35** Mixing equipment specifications.



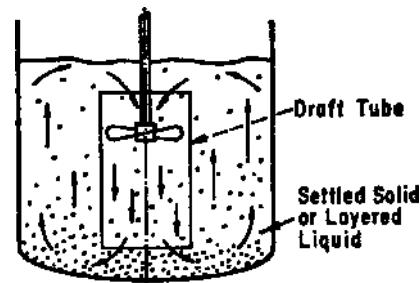
**Figure 7-36a** Draft tubes prevent short-circuiting of liquid from inlet to outlet in a continuous mixing vessel. (By permission from Weber, A.P., *Chem. Eng.*, Oct 1953, p. 183 [51].)



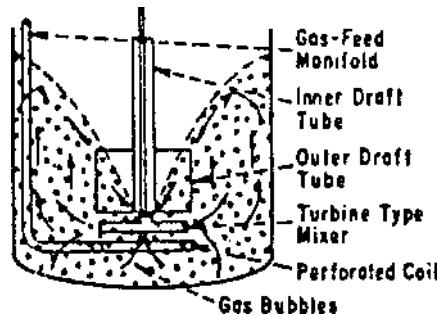
**Figure 7-36e** Baffles positioned in the draft tube accentuate the direct mechanical action of low speed mixing elements. (By permission from Weber, A.P., *Chem. Eng.*, Oct 1953, p. 183 [51].)



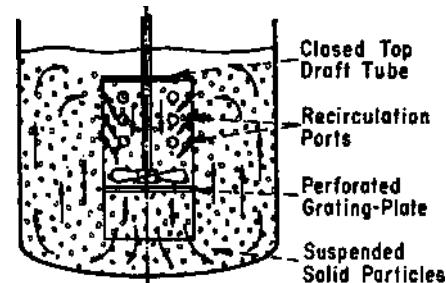
**Figure 7-36b** Forced convection across heat transfer surface improves the overall coefficient of heat transfer. (By permission from Weber, A.P., *Chem. Eng.*, Oct 1953, p. 183 [51].)



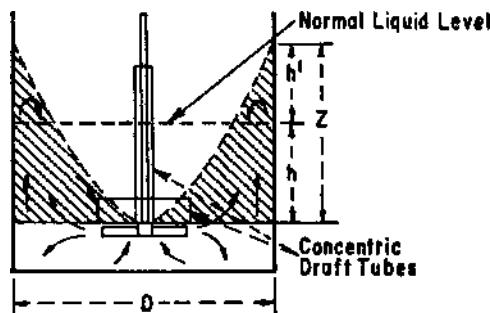
**Figure 7-36f** Settled solids or layered liquids are quickly dispersed by the directionalized flow from the draft tube. (By permission from Weber, A.P., *Chem. Eng.*, Oct 1953, p. 183 [51].)



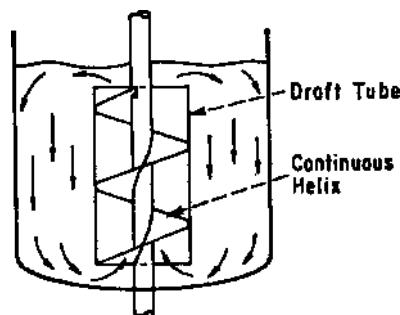
**Figure 7-36c** Gas-liquid mixing is more complete when concentric draft tubes are used to recirculate gases. (By permission from Weber, A.P., *Chem. Eng.*, Oct 1953, p. 183 [51].)



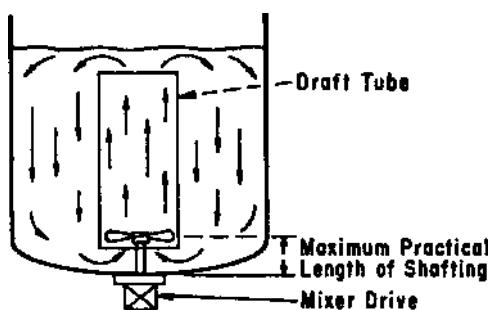
**Figure 7-36g** Direct mechanical action can be increased by the addition of a grating plate to the draft tube. (By permission from Weber, A.P., *Chem. Eng.*, Oct 1953, p. 183 [51].)



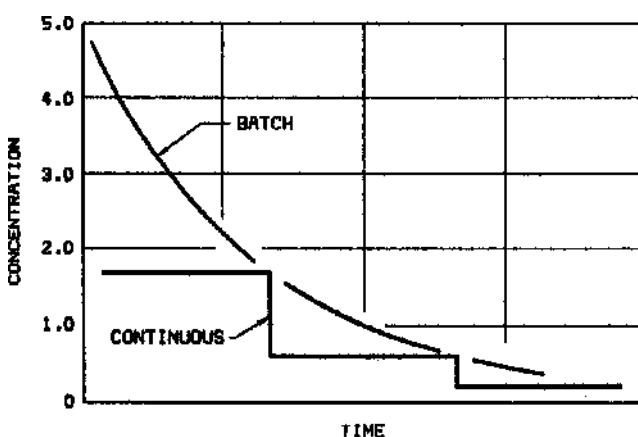
**Figure 7-36d** Capacity of a draft tube assembly to suck in gases is a function of the liquid height above the rotor hub. (By permission from Weber, A.P., *Chem. Eng.*, Oct 1953, p. 183 [51].)



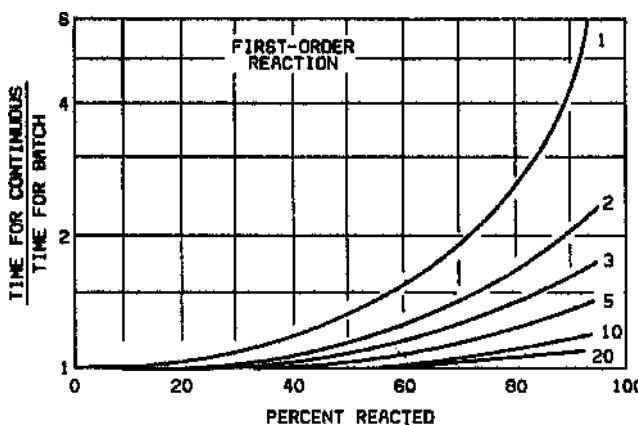
**Figure 7-36h** Helix-in-draft-tube assemblies are effective for crutting pastry or fibrous materials. (By permission from Weber, A.P., *Chem. Eng.*, Oct 1953, p. 183 [51].)



**Figure 7-36i** Mechanical design problems may be solved by using a draft tube to amplify the action of the mixer. (By permission from Weber, A.P., *Chem. Eng.*, Oct 1953, p. 183 [51].)



**Figure 7-37a** Illustration of chemical reaction in a batch system in which concentration decreases with time, with a three-stage continuous mixing system superimposed. (By permission from Oldshue, J.Y., "Fluid Mixing Technology", *Chem. Eng.*, McGraw-Hill Publications Co. Inc., 1983, pp. 340, 347, and 348 [14].)



**Figure 7-37b** For a first-order reaction, the ratio of time in a continuous tank to the time in a batch tank for various percentages of reaction completion. (By permission from Oldshue, J.Y., "Fluid Mixing Technology", *Chem. Eng.*, McGraw-Hill Publications Co. Inc., 1983, pp. 340, 347, and 348 [14].)

mixing/agitation in different-sized systems [16]. This is usually only a part answer to the scale-up problem.

Geometric similarity is often considered the most important feature to establishing similarity in mixing, basing the scaled-up larger unit on the smaller initial model or test unit.

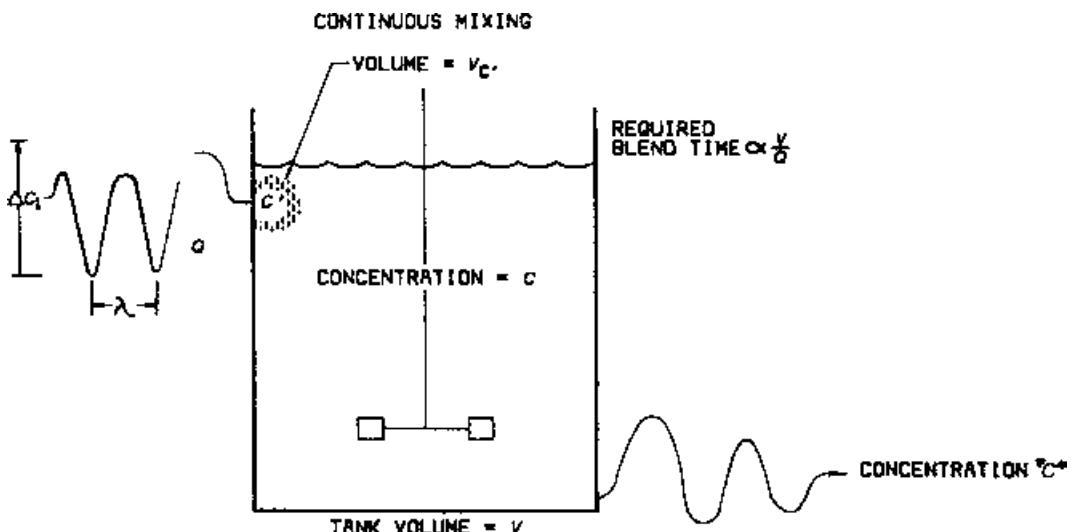
The scale-up of mixing data has been treated with a variety of approaches, some to rather disastrous results. The principles are now well established, and it is a matter of truly understanding the particular systems that present the real problem. The important similitude concept involves the following:

1. Geometric similarity requires all corresponding dimensions of a new system to have the same ratio with a test model which has proven acceptable. These dimensions should include vessel diameter and liquid level, baffle width and number in vessel, impeller diameter, number of blades, and width ratio. For example, a tank four times the diameter of the original model also requires a turbine ten times the diameter of the original turbine.
2. Kinematic similarity requires geometric similarity and requires corresponding points in the system to have the same velocity ratios and move in the same direction between the new system and the model.
3. Dynamic similarity requires geometric and kinematic similarity in addition to force ratios at corresponding points being equal, involving properties of gravitation, surface tension, viscosity, and inertia [23, 54]. With proper and careful application of this principle, scale-up from test model to large-scale systems is often feasible and quite successful. Tables 7-9 and 7-10 present the relationships of the major variables for the two most important cases of mixing.

Often, exact or true kinematic and dynamic similarity cannot be achieved in a system requiring small-scale testing to determine the effect of the design or flexibility in design to allow for final design "trimming". Consideration should definitely be given to such flexibility as (a) mixing impeller designs that can be modified without excessive cost, or the need to build a completely new/larger/smaller unit, (b) multiple gear ratios for the gear drive, with spare ratio gears to adjust speeds, and (c) either variable speed driver or oversized driver to allow for horsepower adjustments.

The dynamic response used to describe fluid motion in the system is bulk velocity. Kinematic similarity exists with geometric similarity in turbulent agitation [53]. To duplicate a velocity in the kinematically similar system, the known velocity must be held constant; for example, the velocity of the tip speed of the impeller must be constant. Ultimately, the process result should be duplicated in the scaled-up design. Therefore, the geometric similarity goes a long way in achieving this for some processes, and the achievement of dynamic and/or kinematic similarity is sometimes not that essential.

For scale-up, the "shear-rate" of the fluid, which is a velocity gradient that can be calculated from velocity profiles at any point in the mixing tank [14], is an important concept. The shear rate is the slope of the velocity vs distance curve. Using the time average velocity yields shear rate values between the adjacent layers of fluid that operate on large particles of about  $200\mu$  or greater. In Figure 7-39, usually a maximum shear rate will exist at the impeller jet boundary. The average shear rate is primarily a function of the time average velocity and impeller speed, and is not a function of any geometric type of impeller or the impeller diameter [14]. The maximum shear rate exists at the jet boundary and is a direct function of impeller diameter and speed, which is related to the peripheral speed of the impeller. Thus, on scale-up, the maximum impeller zone shear rate tends to increase while the average impeller zone shear rate tends to decrease [14].



**Figure 7-38** Concept of perfect mixing, in which the feed is dispersed instantly into the tank and the exit concentration is equal to the tank concentration. (By permission from Oldshue, J.Y., "Fluid Mixing Technology", *Chem. Eng.*, McGraw-Hill Publications Co. Inc., 1983, pp. 340, 347, and 348 [14].)

**TABLE 7-9 Impeller and Flow Characteristics for Turbulent, Baffled Systems – Simple Ratio Relationships**

At Constant ↓	$P_r$	$D_r$	$N_r$	$Q_r$	$H_r$	$(Q/H)_r$
Impeller Diameter, $D$	$N_r^3$	–	$P_r^{1/3}$	$N_r$	$N_r^2$	$N_r^{-1}$
Speed, $N$	$D_r^5$	$P_r^{1/5}$	–	$D_r^3$	$D_r^2$	$D_r$
Power, $P$	–	$N_r^{-3/5}$	$D_r^{-5/3}$	$N_r^{-4/5}$ or $D_r^{4/3}$	$N_r^{4/5}$ or $D_r^{-4/3}$	$N_r^{-8/5}$ or $D_r^{8/3}$

(Source: By permission from Chemineer, Inc., Dayton, OH [54].)

**TABLE 7-10 Impeller and Flow Characteristics for Viscous, Baffled, or Unbaffled Systems – Simple Ratio Relationships**

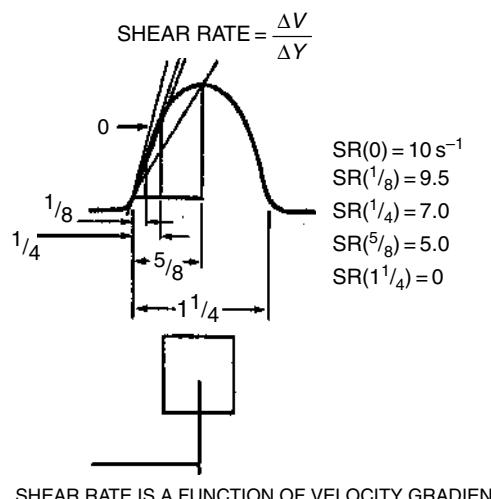
At Constant ↓	$P_r$	$D_r$	$N_r$	$Q_r$	$H_r$
Impeller Diameter, $D$	$N_r^2$	–	$P_r^{1/2}$	$N_r$	$N_r$
Speed, $N$	$D_r^3$	$P_r^{1/3}$	–	$D_r^3$	–
Power, $P$	–	$N_r^{-2/3}$	$D_r^{-3/2}$	$N_r^{-1}$ or $D_r^{3/2}$	$D_r^{-3/2}$ or $N_r$

(Source: By permission from Chemineer, Inc., Dayton, OH [54].)

The fluid shear stress actually brings about the mixing process, and is the multiplication of fluid shear rate and viscosity of the fluid [14].

The pumping capacity of the impeller is important in establishing the shear rate due to the flow of the fluid from the impeller.

There is no constant scale-up factor for each specific mixing system/process [14]. The two independent impeller variables come from speed, diameter, or power, because once the impeller type/style has been selected, the two variables can be established. The third variable is tied through the power curves (plot of power number vs  $N_{Re}$ , see Figures 7-21–7-23). Figure 7-40 shows that geometric and dynamic similarity can develop useful relationships for some situations [14], but not all, and it is not truly possible to prepare one dimensionless group expressing a process relationship. This suggests that care must be used in resorting

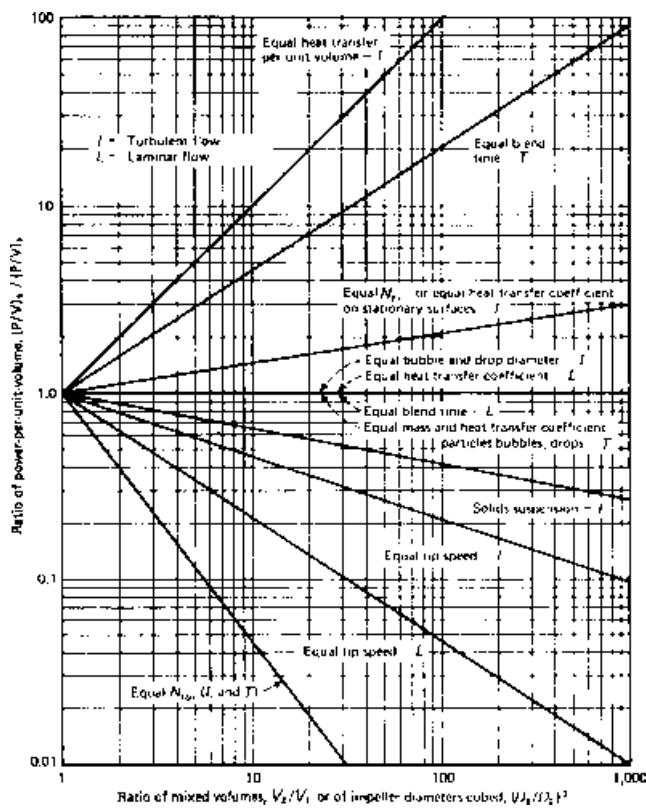


SHEAR RATE IS A FUNCTION OF VELOCITY GRADIENT

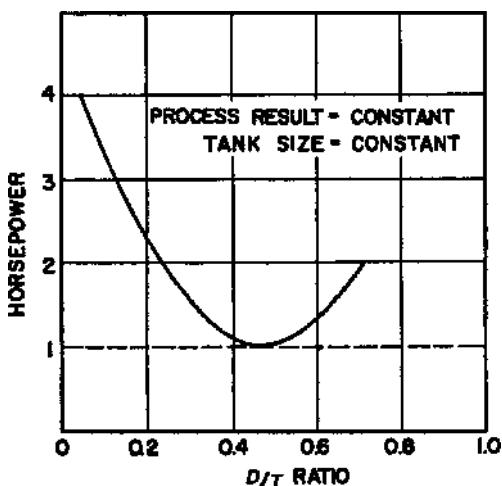
**Figure 7-39** Shear rate is a function of velocity gradient. (By permission from Lightnin Technology; Lightnin Technology Seminar, 3rd ed., 1982. Lightnin (Formerly Mixing Equipment Co.), a unit of General Signal, p. 1, Section 2A [12].)

only to a dimensionless number for process correlations. Also see Figures 7-41 and 7-42.

Because the most common impeller type is the turbine, most scale-up published studies have been devoted to that unit. Almost all scale-up situations require duplication of process results from

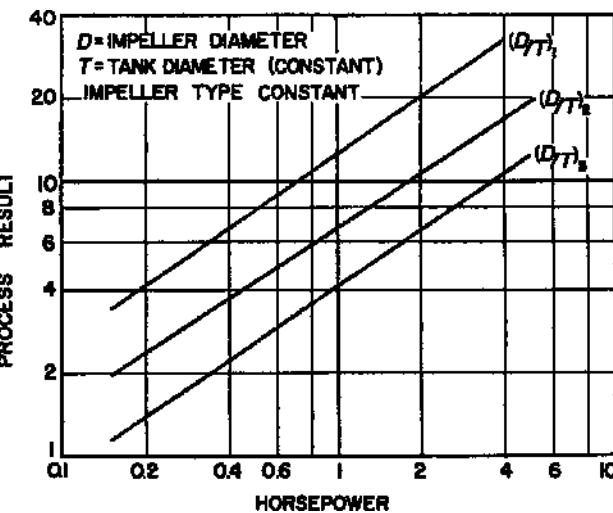


**Figure 7-40** Mixing scale-up factors referenced to experienced ratios of power per unit volume. (By permission from Penny, W.R., *Chem. Eng.*, Mar 22, 1971, McGraw-Hill Inc. p. 83 [15].)

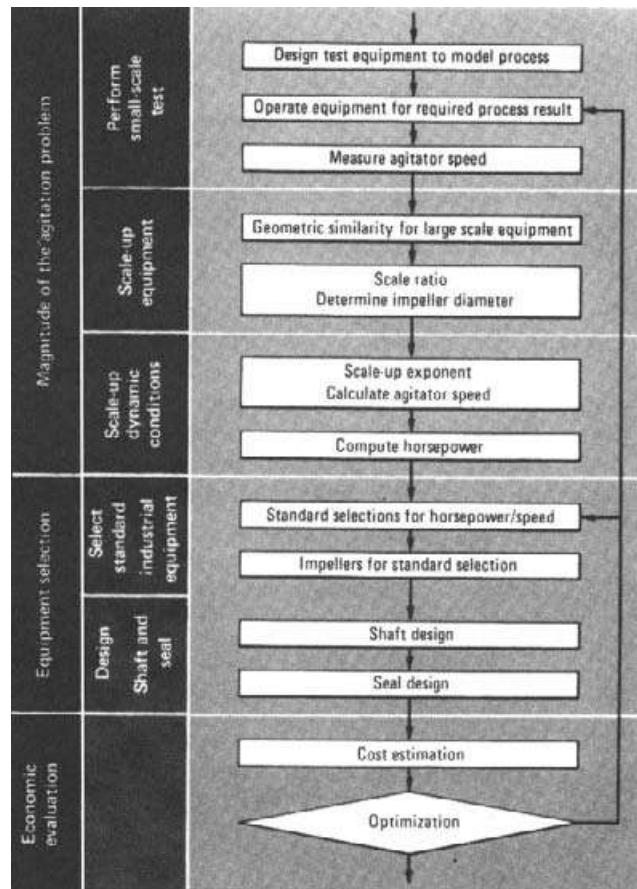


**Figure 7-41** Effect of  $D/T$  on power requirement for a given process result. (By permission from Fluid Mixing, Lightnin (Formerly Mixing Equipment Co.), a unit of General Signal.)

the initial scale to the second scaled unit. Therefore, this is the objective of the outline to follow, from [53]. The dynamic response is used as a reference for agitation/mixer behavior for a defined set of process results. For turbulent mixing, kinematic similarity occurs with geometric similarity, meaning fixed ratios exist between corresponding velocities.



**Figure 7-42** Effect of power on process result with constant  $D/T$  ratio.  
 (By permission from Fluid Mixing, Lightnin (Formerly Mixing Equipment Co.), a unit of General Signal.)



**Figure 7-43** Design procedure for agitator scale-up. (By permission from Rautzen, R.R. et al., *Chem. Eng.*, Oct 25, 1976, p. 119 [53].)

For scale-up procedure, refer to Figure 7-43 which outlines the steps involved in selecting commercial or industrial mechanical agitation equipment when based on test data.



If  $P/V$  is held constant in two different systems, then

$$N_1^3 D_1^2 = N_2^3 D_2^2 \quad (7-79)$$

or,

$$N_2 = N_1 \left( \frac{D_1}{D_2} \right)^{2/3} = N_1 \left( \frac{1}{R} \right)^{2/3} \quad (7-80)$$

Figure 7-45 provides a selection grid for establishing the industrial standard for the driving equipment, that is, input horsepower of motor and speed of agitator. Select the calculated point in the grid, then move to the nearest speed and input horsepower (not motor hp). Usually adjustments from actual design will be required, but often the incremental adjustments will not be great. To aid in these adjustments, equal torque will give equal liquid motion or solids suspension over a narrow range. To use equal torque, set up diagonals from lower left to upper right for the available equipment. Adjust or back-calculate (if necessary) the expected performance and dimensions based on the adjustment. Higher speeds require higher horsepower.

For equivalent mass transfer, equal power changes are used.

To recheck final design, the diameter of a single pitched-blade turbine, for turbulent conditions, is given by:

$$D_T = 394 \left( \frac{P_{hp}}{S_g N^3} \right)^{1/5}, \text{ in.} \quad (7-81)$$

$D_T$  = impeller diameter, turbulent operation, in.

$P_{hp}$  = horsepower used by system (not motor horsepower of driver)

$S_g$  = specific gravity of fluid

$N$  = agitator impeller speed, rpm.

Of course economics enters into the solution, so some alternate designs may be helpful.

Five forces that can usually be used for scale-up are given below:

### 1. Input force from mixer, function of [10]

impeller speed  
impeller diameter.

### 2. Opposing forces, functions of

viscosity  
density of fluid  
surface tension.

### 3. Dynamic similarity requires that the ratio of input force, viscosity, density, and surface tension be equal. For the same fluid, only two of these four forces need be equal, because the density and viscosity will be the same [37, 52].

The geometric and dynamic similarity can use dimensionless groupings.

### 4. Geometric [14]:

$$\frac{X_m}{X_p} = X_R, \text{ common ratio} \quad (7-82)$$

$X_m$ ,  $X_p$  = dimension of model and scale-up unit, respectively

$X_R$  = ratio of dimensions.

### 5. Dynamic [14]:

$$\frac{(F_1)_m}{(F_1)_p} = \frac{(F_v)_m}{(F_v)_p} = \frac{(F_G)_m}{(F_g)_p} = \frac{(F_\sigma)_m}{(F_\sigma)_p} = F_R \quad (7-83)$$

$F$  = force

Subscripts:  
 1 = inertia force  
 v = viscosity force  
 G = gravity  
 $\sigma$  = interfacial tension  
 R = ratio  
 m = model  
 p = prototype.

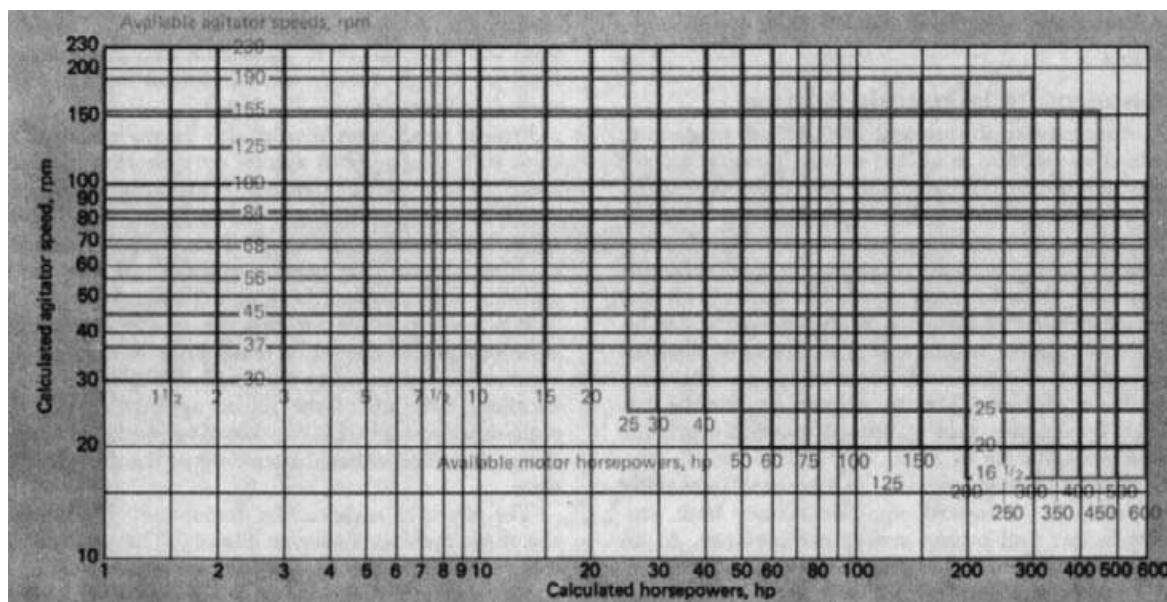


Figure 7-45 Scaled-up horsepower/speed requirement for an agitator system is readily related to industrial equipment. (By permission from Rautzen, R.R. et al., *Chem. Eng.*, Oct 25, 1976, p. 119 [53].)

Force ratios:

$$\frac{F_1}{F_v} = N_{Re} = \frac{ND^2\rho}{\mu} \quad (7-84)$$

$$\frac{F_1}{F_G} = N_{Fr} = \frac{N^2 D}{g} \quad (7-85)$$

$N_{Fr}$  = Froude number

$$\frac{F_1}{F_\sigma} = N_{We} = \frac{N^2 D^3 \rho}{\sigma} \quad (7-86)$$

$N_{We}$  = Weber number

### EXAMPLE 7-3

Scrapper blades set at 35 rpm are used for a pilot plant addition of liquid ingredients into a body wash product. What should the speed of the blades be in a full-scale plant, if the pilot and the full-scale plants are geometrically similar in design? Assume scale-up is based on constant tip speed, diameter of the pilot plant scrapper blades is 0.6 m, and diameter of the full-scale plant scrapper blades is 8 ft.

*Solution*

The diameter of the full-scale plant scrapper blades = 8.0 × 0.3048 = 2.4384 m (2.4 m).

Assuming constant tip speed,

$$\frac{N_2}{N_1} = \frac{D_{A1}}{D_{A2}} \quad (7-87)$$

where

$N_1$  = scrapper speed of pilot plant

$N_2$  = scrapper speed of full-scale plant

$D_{A1}$  = diameter of pilot plant scrapper blades

$D_{A2}$  = diameter of full-scale plant scrapper blades.

$$\begin{aligned} N_2 &= \frac{N_1 D_{A1}}{D_{A2}} \\ &= \frac{(35)(0.6)}{(2.4)} \\ &= 8.75 \text{ rpm} \end{aligned}$$

### EXAMPLE 7-4

A turbine agitator having six flat blades and a disk has a diameter of 0.203 m and is used in a tank having a diameter of 0.61 m and height of 0.61 m. The width  $W = 0.0405$  m. Four baffles are used having a width of 0.051 m. The turbine operates at 275 rpm in a liquid having a density of 909 kg/m<sup>3</sup> and viscosity of 0.02 Pa s.

Calculate the kW power of the turbine and kW/m<sup>3</sup> of volume. Scale-up this system to a vessel whose volume is four times as large, for the case of equal mass transfer rate.

*Solution*

Reynolds number for mixing,  $N_{Re}$ , is

The number of revolutions per second is  $N = \frac{275}{60} = 4.58$  rps.

$$\begin{aligned} N_{Re} &= \frac{\rho ND_A^2}{\mu} \\ &= \frac{(909)(4.58)(0.203)^2}{0.02} \quad \left\{ \frac{\text{kg}}{\text{m}^3} \cdot \frac{1}{\text{s}} \cdot \text{m}^2 \cdot \frac{\text{ms}}{\text{kg}} \right\} \\ &= 8578.1 \end{aligned}$$

$$N_{Re} \approx 8600$$

Using curve 6 in Figure 7-21, the Power number  $N_p = 6.0$

The power of the turbine  $P = N_p \rho N^3 D_A^5$

$$\begin{aligned} P &= (6.0)(909)(4.58^3)(0.203^5) \\ &= 0.1806 \text{ kW}(0.24 \text{ hp}) \end{aligned}$$

The original tank volume  $V_1 = \frac{\pi D_{T1}^3}{4}$

The tank diameter  $D_{T1} = 0.61$

$$V_1 = \frac{(\pi)(0.61)^3}{4}$$

$$V_1 = 0.178 \text{ m}^3$$

The power per unit volume is  $P/V$

$$\begin{aligned} \frac{P}{V} &= \frac{0.1806}{0.178} \\ &= 1.014 \text{ kW/m}^3 \end{aligned}$$

*Scale-up of the system*

The scale-up ratio  $R$  is

$$R = \frac{V_2}{V_1} = \frac{\pi D_{T2}^3 / 4}{\pi D_{T1}^3 / 4} = \frac{D_{T2}^3}{D_{T1}^3}$$

By using Eq. (7-75)

$$R = \frac{D_{T2}}{D_{T1}}$$

(continued)

**EXAMPLE 7-4—(continued)**

where

$$V_2 = 4V_1$$

$$V_2 = 4(0.178)$$

$$= 0.712 \text{ m}^3$$

$$R = (4)^{1/3} = 1.587$$

The dimensions of the larger agitator and tank are as follows:

$$D_{A2} = RD_{A1} = 1.587 \times 0.203 = 0.322 \text{ m}$$

$$D_{T2} = RD_{T1} = 1.587 \times 0.61 = 0.968 \text{ m}$$

For equal mass transfer rate  $n = 2/3$ .

$$N_2 = N_1 \left( \frac{1}{R} \right)^{2/3} \quad (7-80)$$

$$N_2 = 4.58 \left( \frac{1}{1.587} \right)^{2/3}$$

$$= 3.37 \text{ rps}$$

The Reynolds number  $N_{Re}$  is

$$N_{Re} = \frac{\rho N_2 D_{A2}^2}{\mu}$$

$$= \frac{(909)(3.37)(0.322)^2}{0.02} \left( \frac{\text{kg}}{\text{m}^3} \cdot \frac{\text{rev}}{\text{s}} \cdot \text{m}^2 \cdot \frac{\text{ms}}{\text{kg}} \right)$$

$$= 15,880.9$$

$$N_{Re} \approx 16,000$$

Using curve 6 in Figure 7-21,  $N_p = 6.0$ Power required by the agitator is  $P_2 = N_p \rho N_2^3 D_{A2}^5$ 

$$P_2 = (6.0)(909)(3.37)^3(0.322)^5$$

$$P_2 = 722.57 \text{ W}$$

$$= 0.723 \text{ kW}(0.97 \text{ hp})$$

The power per unit volume  $P/V$  is given as:

$$\frac{P_2}{V_2} = \frac{0.723}{0.712}$$

$$= 1.015 \text{ kW/m}^3$$

A Microsoft Excel spreadsheet program from the companion website (Example 7-4.xls) was developed for this example.

**MIXING TIME SCALE-UP**

Predictions of the time for obtaining uniformity in concentration in a batch mixing operation can be based on model theory. In using the appropriate dimensionless groups of the pertinent variables, we can develop a relationship between mixing times in the model and large scale systems for geometrically similar equipment.

Consider the mixing in both small and large-scale systems to occur in the turbulent region, designated as  $S$  and  $L$  respectively. Using the Norwood and Metzner correlation [55], the mixing time for both systems is

$$\frac{t_S (N_S D_{AS}^2)^{2/3} g^{1/6} D_{AS}^{1/2}}{H_S^{1/2} \cdot D_{TS}^{3/2}} = \frac{t_L (N_L D_{AL}^2)^{2/3} g^{1/6} D_{AL}^{1/2}}{H_L^{1/2} \cdot D_{TL}^{3/2}} \quad (7-88)$$

Applying the scale-up rule of equal mixing times, and re-arranging Eq. (7-88), we have

$$\left( \frac{N_L}{N_S} \right)^{2/3} = \left( \frac{D_{TL}}{D_{TS}} \right)^{3/2} \left( \frac{D_{AS}}{D_{AL}} \right)^{4/3} \left( \frac{D_{AS}}{D_{AL}} \right)^{1/2} \left( \frac{H_L}{H_S} \right)^{1/2} \quad (7-89)$$

Assuming geometric similarity,

$$\frac{H_L}{H_S} = \frac{D_{AL}}{D_{AS}} \quad (7-90)$$

$$\frac{D_{TL}}{D_{TS}} = \frac{D_{AL}}{D_{AS}} \quad (7-91)$$

Substituting Eqs (7-90) and (7-91) into Eq. (7-89) gives

$$\left( \frac{N_L}{N_S} \right)^{2/3} = \left( \frac{D_{AL}}{D_{AS}} \right)^{3/2} \left( \frac{D_{AS}}{D_{AL}} \right)^{4/3} \left( \frac{D_{AS}}{D_{AL}} \right)^{1/2} \left( \frac{D_{AL}}{D_{AS}} \right)^{1/2} \quad (7-92)$$

$$\left( \frac{N_L}{N_S} \right)^{2/3} = \left( \frac{D_{AL}}{D_{AS}} \right)^{1/6}$$

or

$$\left( \frac{N_L}{N_S} \right) = \left( \frac{D_{AL}}{D_{AS}} \right)^{1/4} \quad (7-93)$$

The exponent  $n$  for the mixing time scale-up rule is 0.25.The power  $P$  of the agitator for both large and small systems is

$$\frac{P_L}{\rho N_L^3 D_{AL}^5} = \frac{P_S}{\rho N_S^3 D_{AS}^5} \quad (7-94)$$

where

$$\frac{P_L}{P_S} = \left( \frac{N_L}{N_S} \right)^3 \left( \frac{D_{AL}}{D_{AS}} \right)^5 \quad (7-95)$$

Substituting Eq. (7-93) into Eq. (7-95) yields

$$\frac{P_L}{P_S} = \left( \frac{D_{AL}}{D_{AS}} \right)^{0.75} \left( \frac{D_{AL}}{D_{AS}} \right)^5 \quad (7-96)$$

or

$$\frac{P_L}{P_S} = \left( \frac{D_{AL}}{D_{AS}} \right)^{5.75} \quad (7-97)$$

The power per unit volume P/V for both large and small-scale systems is

$$\begin{aligned} \frac{P_L/V_L}{P_S/V_S} &= \frac{P_L / \frac{\pi D_{TL}^3}{4}}{P_S / \frac{\pi D_{TS}^3}{4}} \\ &= \frac{P_L}{P_S} \cdot \left( \frac{D_{TS}}{D_{TL}} \right)^3 \end{aligned} \quad (7-98)$$

Substituting Eqs (7-91) and (7-97) into Eq. (7-98) gives

$$\begin{aligned} \frac{(P/V)_L}{(P/V)_S} &= \left( \frac{D_{AL}}{D_{AS}} \right)^{5.75} \left( \frac{D_{AS}}{D_{AL}} \right)^3 \\ &= \left( \frac{D_{AL}}{D_{AS}} \right)^{2.75} \end{aligned} \quad (7-99)$$

Table 7-11 summarizes the effects of equipment size on the rotational speed needed for the same mixing time by various investigators.

The relationships in Table 7-11 show that the rotational speed to obtain the same batch mixing time is changed by a small power of the increase in linear equipment dimension as equipment size is changed. Equation (7-94) shows that greater power will be required for a large-scale system compared to a smaller system. Very often, the power required for a larger system may be prohibitive, thus modification of the scale-up rule will be needed (e.g.  $t_L = 10t_S$  or  $t_L = 100t_S$ ), so as to obtain a lower power requirement. It is essential to note that relaxation of mixing time requirements does not pose other problem. For instance, if the mixing is accompanied by chemical reaction in a CFSTR, assuming that the Norwood-Metzner [55] correlation for mixing time ( $t$ ) is still applicable, then we must ensure that the mixing time in the larger scale ( $t_L = 10t_S$  or  $t_L = 100t_S$ ) is less than 5% of the average residence time of the liquids in the reactor, otherwise the conversion and product distribution will be affected [56]. Figure 7-40 summarizes the scale-up relationships for many of the important and controlling functions, depending upon the nature of the process equipment. The figure identifies which curves apply to laminar ( $L$ ) and turbulent ( $T$ ) flow patterns in the fluid being subjected to the mixing operation. The scale-up chart only applies to systems of similar geometry. When the geometry is different, special and specific analyses of the system are required, as the chart will not apply.

Samant and Ng [57] compared various scale-up rules for agitated reactors. They suggested that a scale-up rule of power per unit volume and constant average residence time (where the power per unit volume and average residence time cannot be increased) is the most suited in many operations. However, this may still

not improve or preserve the performance of the systems. Therefore, adequate consideration must be given to a trade-off between performance and operating constraints.

#### **Heat Transfer, Hydraulic Similarity [14]**

$$h = f(N, D, \rho, \mu, C_p, k, d) \quad (7-100)$$

$$\left( \frac{\text{result}}{\text{system conductivity}} \right) = f \left( \frac{\text{applied force}}{\text{resisting force}} \right) \quad (7-101)$$

$$\left( \frac{hD}{k} \right) = \left( \frac{ND^2\rho}{\mu} \right)^x \left( \frac{C_p\mu}{k} \right)^y \left( \frac{D}{d} \right)^z \quad (7-102)$$

$x, y, z$  are empirical exponents

$C_p$  = specific heat of fluid

$d$  = heat transfer tube diameter

$D$  = impeller diameter

$h$  = film coefficient of heat transfer

$k$  = thermal conductivity

$N$  = impeller speed

$\rho$  = density of fluid or specific gravity

$\mu$  = viscosity of fluid.

#### **Blending, Hydraulic Similarity**

$$\theta = f(N, D, \rho, \mu, T) \quad (7-103)$$

where

$\theta$  = time

$T$  = tank diameter.

$$\left( \frac{\text{Result}}{\text{system conductivity}} \right) = f \left( \frac{\text{applied force}}{\text{resistancy force}} \right)$$

$$\theta N \propto \left( \frac{ND^2\rho}{\mu} \right)^x \left( \frac{D}{T} \right)^z \quad (7-104)$$

$x$  and  $z$  are empirical coefficients.

## **7.17 BAFFLES**

Vertical sidewall baffles (Figures 7-19b, c and 7-46) projecting about 1/10–1/12 of the tank diameter into the vessel perform a helpful purpose in controlling vortex action. The baffles are set off from the tank wall a few inches to prevent build-up of particles.

The important dimensional features and/or ratios for a center-mounted mixer unit (vertical) are [14, 58] given below.

**TABLE 7-11 Effect of Equipment Size on Rotational Speed needed for the Same Mixing Time**

Relationship between $N$ and $D$	Equipment	$\Delta\rho$	Equation	Investigator
$N \propto D^{-1/6}$	Propellers, no baffles	Not zero	$\left( \frac{\theta ND^2\rho}{V} \right) \left( \frac{\rho D^2 N^2}{g Z_L \Delta\rho} \right) = 9$	van der Vusse [68]
$N \propto D^{-0.1}$ to $D^{-0.2}$	Paddles, turbines	Not zero	$\left( \frac{\theta Q}{V} \right) \propto \left( \frac{\rho D^2 N^2}{g Z_L \Delta\rho} \right)^{-0.3}$	van der Vusse [68]
$N = \text{constant}$	Propellers, paddles, turbines	Zero		van der Vusse [68]
$N \propto D^{-1/5}$	Propellers	Zero	$\theta = \frac{C_1 Z_L^{1/2} T}{N_{Re}^{1/6} (ND^2)^{4/6} g^{1/6}}$	Fox and Gex [69]
$N \propto D^{1/4}$	Turbines	Zero		Norwood and Metzner [65]

Source: Gray, J.B., Mixing I Theory and Practice, Uhl, V.W., and Gray, J.B., eds. Academix Press Inc., 1966.

**EXAMPLE 7-5**

Scale-up from Small Test Unit [58], (see Figure 7-46)

Follow the example of [53], using scale-up rules. A pilot plant test run has been conducted using a laboratory-equipped test vessel. Design equivalent process results for a 10,000 gal tank are shown in the table below.

Data from Test Unit	Proposal Vessel
Vessel dia. $T_1 = 12$ in.	$T_2 = 144$ in.
Vessel liquid level, $Z_1 = 12$ in.	$Z_2 = 144$ in.
Batch volume, $V_1 = 6$ gal.	$V_2 = 10,000$ gal
Impeller dia., $D_1 = 4$ in.	$D_2 = 48$ in.
Impeller shaft speed, $N_1 = 450$ rpm	$N_2 = 90$ (calc.)
Impeller width, $W_1 = 1$ in.	$W_2 = 12$ in.
Baffle width, (4) $B_1 = 1$ in.	$B_2 = 12$ in.
Distance to impeller, $C_1 = 4$ in.	$C_2 = 48$ in.
HP input to shaft, $HP_1 = 0.0098$ (calc.)	HP input $HP_2 = 19.5$ (calc.)

*Solution*

From scale-up ratio:

$$R = \left( \frac{V_2}{V_1} \right)^{1/3} = \left( \frac{10,000}{6} \right)^{1/3} = 11.85, \text{ round to } 12$$

Then, using relationships of volume ratios to geometric similarity,

$$\frac{T_1}{T_2} = \frac{12}{R}$$

$$T_2 = (12 \text{ in.})(12) = 144 \text{ in.}$$

$$Z_2 = (12 \text{ in.})(12) = 144 \text{ in.}$$

$$D_1 = (4 \text{ in.})(12) = 48 \text{ in.}$$

$$W_2 = (1 \text{ in.})(12) = 12 \text{ in.}$$

$$B_2 = (1 \text{ in.})(12) = 12 \text{ in.}$$

$$C_2 = (4 \text{ in.})(12) = 48 \text{ in.}$$

$$N_2 = ?$$

For example, blend time as the criterion, the scale-up exponent = 0 and no change in speed is required for the larger-scale equipment, requiring that the longer 48-in. diameter turbine operates at 450 rpm,

$$\begin{aligned} HP &= \left( \frac{D_T}{394} \right)^5 (S_g)(N)^3 \\ &= \left( \frac{48}{394} \right)^5 (1)(450)^3 \\ &= 2445.5 \text{ hp input to shaft of impeller, impossible to use} \end{aligned}$$

Alternating, scaling-up for equal mass transfer, with  $n = 0.667$ , reading Figure 7-44 speed ratio,

$$\frac{N_1}{N_2} = 0.2, \text{ at } \frac{V_2}{V_1} = \frac{10,000}{6} = 1666$$

Then,

$$N_2 = 0.2(450) = 90 \text{ rpm shaft speed}$$

and

$$HP = \left( \frac{D_2}{394} \right)^5 (S_g)(N)^3$$

$$HP = \left( \frac{48}{394} \right)^5 (1)(90)^3$$

$$= 19.5 \text{ hp input to shaft}$$

$$\text{Actual motor hp: } 19.5/0.85 = 22.9$$

Using the grid in Figure 7-45, calculated shaft hp = 19.5 and agitator speed = 90 rpm.

Read closest motor hp = 20. However, the 0.5 hp difference between 19.5 and 20 may not be sufficient to handle the power loss in the gear box. Most industrial practice is to take the closest standard motor hp to the 22.9 hp determined above, which is 25 hp. The gear box must have an output speed of 90 rpm and will use only the hp determined by the impeller shaft even if the motor is larger, that is, 25 hp. It will only put out the net hp required, that is, the sum of impeller shaft and losses through the gear box.

Referring to Table 7-9 for turbulent, baffled systems, if power is held constant and the system has too large a shear characteristic and apparently too small a volume or flow, the impeller can be increased 20% and the new speed at constant Power,  $P$ , will be:

$$N_r = D_r^{-5/3} \quad (7-105)$$

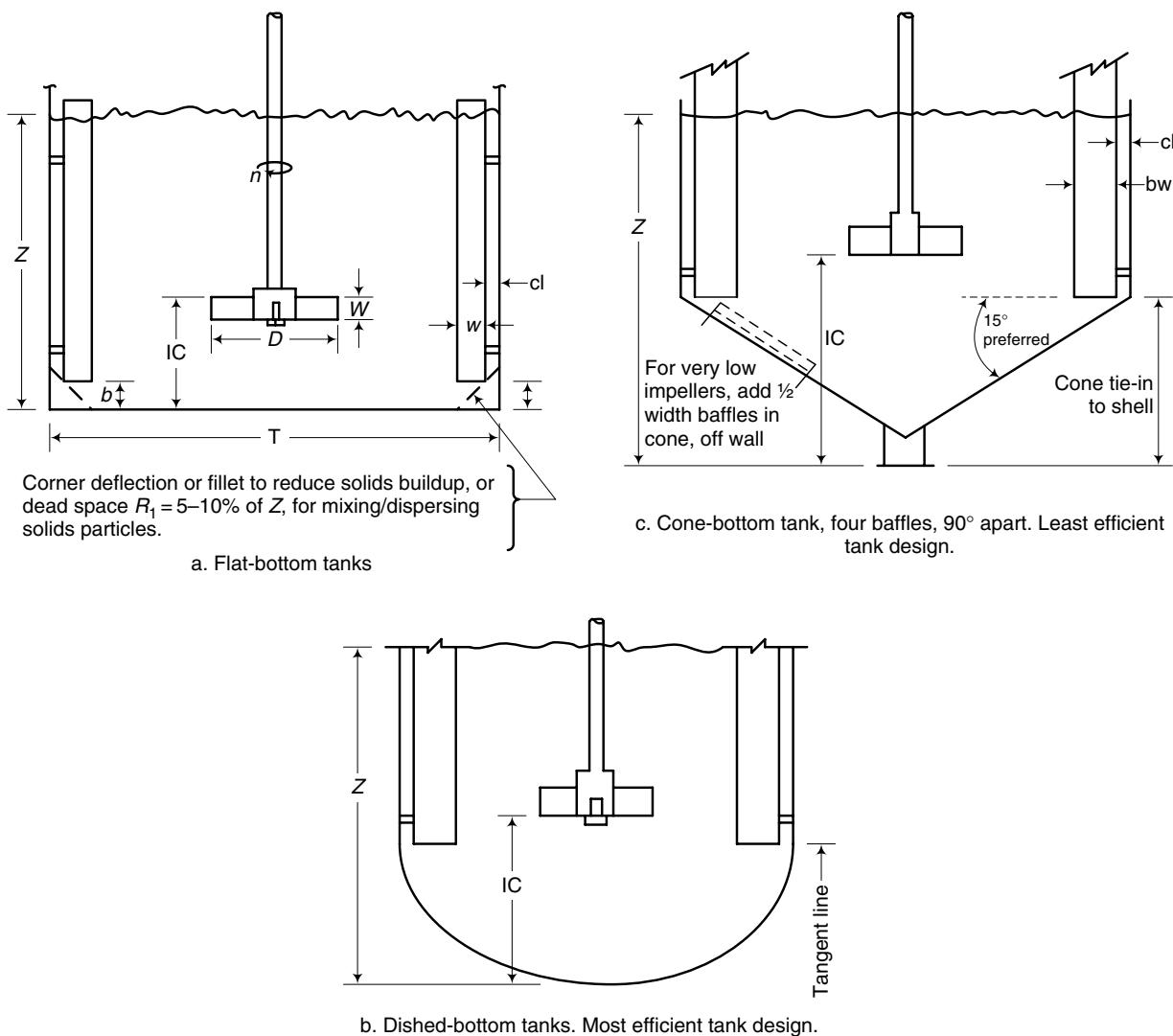
$$\frac{N_1}{N_2} = \left( \frac{D_1}{D_2} \right)^{-5/3} \quad (7-106)$$

$$N_2 = N_1 \left( \frac{D_2}{D_1} \right)^{-5/3} = N_1 \left( \frac{1.2}{1.0} \right)^{-5/3} = 0.738(N_1) \quad (7-107)$$

Use Table 7-10 for viscous systems, or the new speed will be 73.8% of the original, using the 20% larger diameter impeller. This is true for geometrically similar systems. However, there is little power change over a wide ratio of impeller-to-tank diameter.

For a constant amount of power available to a system, the flow and turbulence effects and ratios can be changed by replacing one impeller by another dimensionally similar. Figures 7-41 and 7-42 illustrate the type of studies which should be made in evaluating a system. If the density or viscosity of a fluid changes during scale-up, then in the turbulent range the horsepower is directly proportional to density, and thus viscosity has very little effect. In viscous flow the density has no effect while the horsepower is proportional to viscosity [54]. The effect is small in the range from 1 to 1000 cP, but amounts to a factor of 1.4 when changing from 1000 to 10,000 cP. Above this point the change is quite large and should not be handled by proportion.

Figure 7-40 summarizes the scale-up relationships for many of the important and controlling functions, depending upon the nature of the process equipment. The figure identifies which curves apply to turbulent ( $T$ ) and laminar ( $L$ ) flow patterns in the fluids being subjected to the mixing operation. This scale-up chart applies to systems of similar geometry. When the geometry is different, special and specific analysis of the system must be made, as the chart will not apply.



**Figure 7-46** Typical vessel baffles to improve mixing performance. (Adapted/modified by permission from Casto, L.V., *Chem. Eng.*, 1972, p. 97 [58].)

### 1. Flat bottom tank

- Number of vertical baffles in a vertical vessel: four (more than four provides little, if any, benefit)
- Width of baffles:  $1/10\text{--}1/12$  tank diameter,  $w$
- Distance baffles off wall,  $cl$ : 3–6 in.
- Baffle spacing: on 90° around tank
- Distance of baffles off flat vessel bottom: 4–6 in.,  $b$
- Height of liquid in tank:  $Z$
- Height of impeller off flat bottom: IC, equal to impeller (turbine) diameter, or  $IC = D$ ; sometimes  $IC = 2/3D$  is suggested
- Liquid depth over top of impeller or turbine:  $Z-C$  should be  $2D$
- Baffles extend above liquid level.

### 2. Dished bottom tank, center-mounted mixer, four baffles

- Essentially same criteria as for flat bottom tank
- Impeller distance off bottom: essentially same referenced to vessel tangent line as for flat bottom vessel.

### 3. Cone bottom tank. Handle as for dished bottom above.

- For fluids with viscosity up to 5000 cP, and even up to 30,000 cP for some situations, use standard baffles described above. The baffle widths can be reduced as the viscosity increases from 5000 to 12,000 cP [58], and may be eliminated completely for viscosities over 12,000 cP. There are exceptions, such as mixing wide range of fluids of low and high viscosities.
- Special baffles. For certain mixing problems, various baffling arrangements have been found to be advantageous [58].
- Baffles can be omitted when propeller mixers are top mounted at an angular off-center position (see Figure 7-19d) and vortex swirling is prevented. This is not recommended for large power systems on large tanks, due to shaft fatigue.

Baffles that extend from the liquid level down, but not, to the tank bottom allow heavy swirling action in the bottom of the tank, but no vortex at the top. When baffles extend from the bottom up, but not to the liquid level, some vortex and swirling action will take place at the top.

In this case foam can be re-entered into the mixture by this action, and solids or liquids added will enter the impeller rather

rapidly. The deeper the liquid above the baffle the greater the rotating action. Bottom swirling action allows the segregation of heavy solid particles [23].

Generally, some sidewall baffles are desirable in most mixing operations. Baffles allow the system to absorb relatively large amounts of power which is needed for development of mixing turbulence, and still avoid vortex and swirling action, that is, the tank fluid stays under control. This is indicated typically by Figure 7-23 for the flat-blade impeller in area CD (or the similar region on other figures). Use four vertical baffles, see Figures 7-19b, c and 7-46.

A large number of mixing problems operates in this region and can be easily interpreted. Here large amounts of power can be added to the system for greater volume and/or shear forces by simply increasing the speed. This is one reason for variable speed drives. However, in the portion of Figure 7-23 lettered EF as well as AB, BC, and BE, the power changes exponentially with the speed. The advantage of using baffles is that the flow pattern is fixed to follow the portion CD of the curves.

When fluid swirling is prominent, it is difficult or impossible to reproduce the same mass transfer effect in any other size vessel, and hence cannot be reproduced by geometric similarity [23].

Mixing operations are not limited to a flat bottom tank but, in general, for each system there is a tank configuration which is optimum. The difference between them is small in many instances.

A tank bottom of dished or spherical shapes is usually better than a flat bottom as it requires less horsepower for the system. Figure 7-46b shows baffle types that have enhanced mixing effectiveness over a period of time. The latest baffle type is the concave baffle which has increased mixing effectiveness by 60%.

#### IMPELLER LOCATION AND SPACING: TOP CENTER ENTERING

For base tank dimensions,  $Z/T$  (height/tank diameter) equal to 1.0 [14]:

- A. For blending and solid suspension, use  $Z/T$  for minimum power at about 0.6–0.7, but recognize that this may not be the most economical.
- B. One or two impellers can operate on one shaft at liquid coverage ( $Z-IC$ ) over the impeller of at least  $3D$ , see Figure 7-46A.
- C. Draw-off of the mixed liquids influences the location of the impeller. For blending, the preferred impeller location in a  $Z/T = 1.0$ . Vessel is at the mid-point of liquid depth for a continuous flow process, but may not be for a batch system.
- D. It is important to avoid zone mixing, when each impeller on a shaft (of more than one impeller) mixes its own zone, and then a stagnant or less-mixed region develops between the impellers. Multiple axial flow impellers have less tendency in this regard than do multiple radial flow impellers. As viscosity of one fluid increases, the flow pattern becomes more radial, thus the tendency for zone mixing increases. When axial flow impellers are too close together on one shaft, they tend to behave as a single larger impeller, with decrease in power draw but with a decrease in pumping capacity also.
- E. For radial flow turbine, locate  $1.5D$ – $2.0D$  apart, with liquid coverage over the top impeller of minimum  $0.5D$ – $3.0D$ , depending on surface motion desired.
- F. Polyethylene polymer autoclave-type reactors usually contain 8–120 impellers of the same or different circulation designs on a single shaft to ensure rapid total homogeneous mixing in the reactor, which contains a gas at about 30,000 psi and, hence, the fluid is neither a gas nor a liquid because the densities are about the same.

Casto [58] suggests the depth distances in a tank over which the turbines are effective mixers:

Liquid Viscosity (cP)	Turbine Diameters (Vertical Spacing)
<5,000	3.0–4.0
5,000	2.5
15,000	2.0
25,000	1.7
50,000	1.4

The turbine should be positioned to give 35% of the liquid below, and 65% above the turbine. For depths exceeding these values, multiple impellers should be used. For total liquid depths less than the effective height of a turbine impeller, the turbine should be located  $0.35Z$  from the tank bottom, unless there is an overriding restraint.

The top-entering mixer units are better for producing flow at constant power than the side entering units [14]. The radial flow impellers require more horsepower when compared to the marine impellers.

If propeller is located quite close to the bottom of a tank, the flow becomes radial like that of the flat-blade turbine. In a properly baffled system the propeller flow is axial. When dynamic similarity is accomplished, the systems are similar [23]. For a first approximation, placing the impeller at 1/6 of liquid height off the bottom is good.

When possible it is best to withdraw the fluid from a propeller-mixed system directly below the propeller. This allows removal of all solids and mixed liquids.

For a turbine the preferred location for withdrawal of mixed fluid is at the side opposite the turbine impeller. A study of the flow pattern of the system should be made to ascertain that the proper fluid mixture is sampled or removed.

Side-entering mixers (usually propellers) as shown in Figure 7-19g are placed 18–24 in. above a flat tank floor with shaft horizontal, and at a  $10^\circ$  horizontal angle with a vertical plane through the tank centerline. This equipment is used for fluids up to 500 cP [23]. For fluids from 500 to 5000 cP, the mixer is usually top-entering.

Viewing from behind the mixer, for a clockwise rotation of the impeller, the mixer must be angled to the left at the  $7^\circ$ – $10^\circ$  angle to the vertical plane noted above. Side-entering mixers usually run at 280–420 rpm, compared to runs at 30–100 rpm for top-entering units. The side-entering units are primarily used for large blending tanks of gasoline, oils, chemicals (watch the shaft seal through the tank here), paper stock blending, and similar large systems, but they do not necessarily have to be limited to large systems.

The proper placement of the impeller for specific applications is necessary for good mixing performance; therefore, a thorough discussion with a mixing company specialist is useful.

In gas dispersion systems, the gas inlet should normally be directly below the impeller inlet, or on a circular pattern at the periphery of the impeller.

In order to achieve uniform solid suspension or pickup of solid particles off the bottom, the upward velocities of the fluid streams in all portions of the vessel must exceed the terminal settling velocity of the particular particles. This can be determined by small-scale tests.

For solids which float, or which are added from the top, a vortex action helps to draw the material down into the impeller. Often a draft tube is used to serve as a suction entrance for the impeller. Uniform suspensions are difficult to maintain when the tank liquid height is much greater than the tank diameter. The impeller is normally placed at 1/6th of liquid depth off the bottom [23].

## PROCESS RESULTS

The effect on the process of a change in operation of the mixer system (impeller, baffles, etc.) is the final measurement of performance. Thus, operations such as blending, uniform particle suspension, reaction, gas absorption, and so on, may be acceptable under one physical system and not so to the same degree under a slightly modified one. The ratio per unit volume on scale-up must be determined experimentally.

Generally, as system size increases, the impeller flow per fixed power input will increase faster than will the turbulence of the system. Even the same degree of turbulence there is no guarantee that the rates of mixing, mass transfer, and so on, will be the same in the larger system. Once it can be determined whether the process result is controlled, more or less, by flow or turbulence, the scale-up can be more intelligently analyzed with the use of the relations previously presented; noting that head,  $H$ , is the turbulence indicator. Generally, for fixed power input the relative proportions of flow and turbulence in a given system vary as indicated in Table 7-12.

Operations such as blending, solids-suspension, dissolving, heat transfer, and liquid-liquid extraction are typical of systems requiring high flow relative to turbulence, while gas-liquid reactions and some liquid-liquid contacting require high turbulence relative to flow. The case of (1) 100% of suspension requires head to keep particles suspended and (2) 100% uniformity of distribution of particles requires head for suspension plus flow for distribution.

In the case of heat transfer using coils in the tank, it is generally necessary to increase the horsepower per unit volume from small scale to full size equipment.

For some scale-up situations, particularly when the change in size of the system is not great and the fluid properties remain unchanged, the use of horsepower per unit of liquid volume is an acceptable scale-up tool.

The horsepower per unit volume is fairly constant with increasing tank volume, actually falling off slightly at large volumes. Therefore, a usually safe scale-up is to maintain a constant HP per volume.

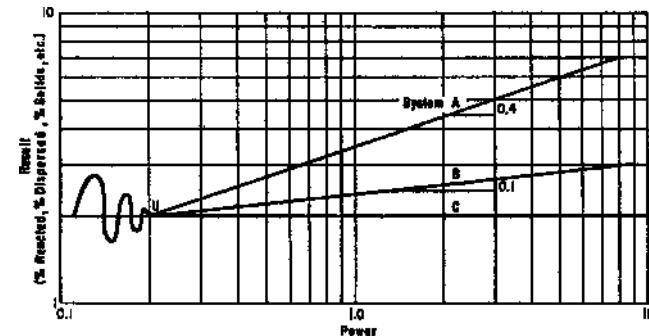
Bates [40] describes the handling of process results for reaction completion, gas absorption, and phase distribution when related to power, as shown in the log-log plot of Figure 7-47.

The slope of the line is significant and serves as a guide to the type of mixing mechanism required. The irregular line to the left of point U indicates non-uniformity of tank contents. At U, homogeneity is accomplished. The System C line indicates no change is required in horsepower in order to achieve better or different results. This would be typical of a blending operation. For System B or any system between C and B where the slope of the line is between 0 and 0.1, the action is only slightly influenced by mixing, and for scale-up the system will be more controlled by the process and fluid properties than by increased mixing action. A change in  $D/T$  will only slightly influence the process result.

For system A and generally those in between systems with slope 0.1–0.4, the process result is primarily dependent upon

**TABLE 7-12 Expected Proportions of Flow and Turbulence in a Mixing System**

Relative Impeller Diameter	Percent Flow	Percent Turbulence	Relative Speed
Large	High, >50	Low, <50	Low
Medium	About 50	About 50	Medium
Small	Low, <50	High, >50	High



**Figure 7-47** Process result as function of power. (By permission from *Fluid Agitation Handbook*, Chemineer, Inc.)

impeller flow and will be influenced by a change in  $D/T$ . For those systems with slopes greater than 0.4, the process result is significantly influenced by fluid head and shear and by the ratio of impeller flow to shear. The determination of the proper  $D/T$  is worth close study [54].

## 7.18 BLENDING

Blending of two or more fluids into a uniform mixture is quite common in the finishing of many chemical and petroleum products. This includes the addition of additives as well as upgrading off-specifications with above-specification material to yield a salable product. Rushton [33] describes blending in large tanks and Oldshue et al. [59] evaluate factors for effective blending. The 7°–12° angle with a flat bottom tank plan centerline as shown in Figure 7-19g as been shown to be optimum for efficient blending in small and large tanks. The angle should be to the left of the centerline and the propeller should rotate clockwise when viewed from the shaft or driver end. The results for low viscosity fluids (0.3–1.0 cP) relate time for a complete blend using a side-entering, 3-blade propeller mixer starting with a full stratified tank (no baffles) of two liquids [23].

$$\theta = k' \left( \frac{\rho_h - \rho_l}{\rho_h} \right)^{0.9} \left( \frac{D}{T} \right)^{-2.3} (\text{HP})^{-1} \quad (7-108)$$

This time is considerably longer than for the arrangement with the second fluid entering the suction of the operating propeller.

For side-entering horizontal mixers not limited to blending operations, there are some differences in recommendations concerning the physical location of the impeller:

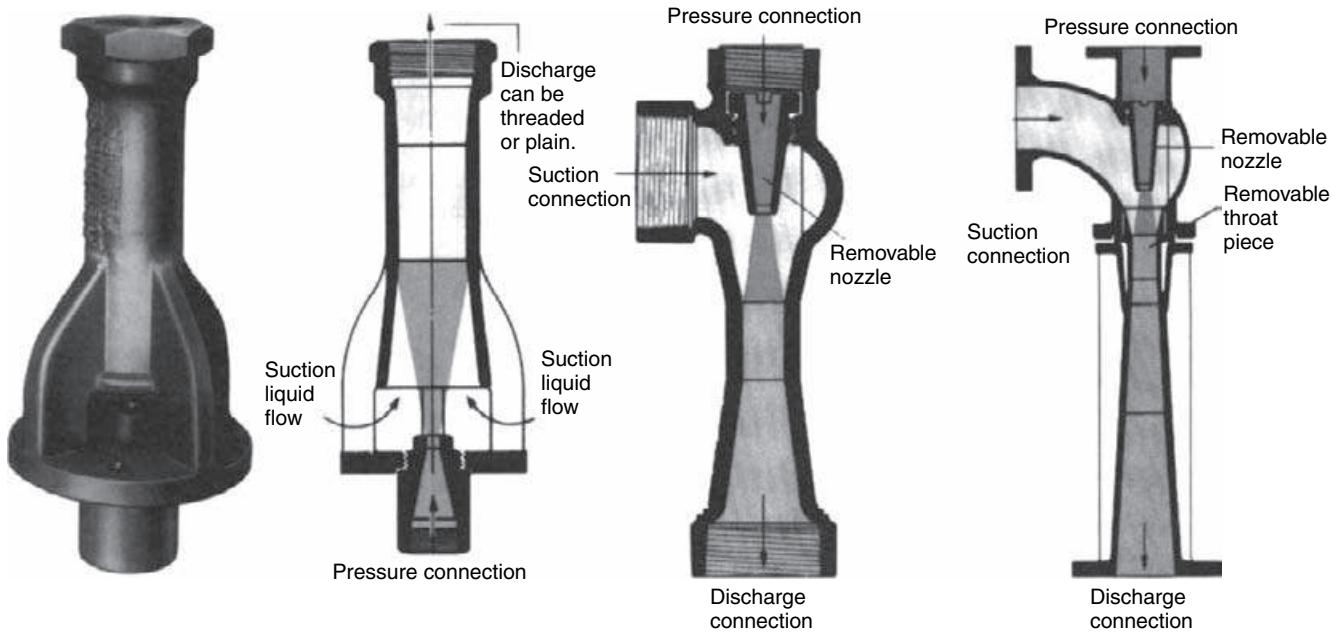
1. The impeller should be located 1/2 to 1 1/2 times the impeller diameter away from the tank wall in plan.
2. The impeller centerline should be 3/4 to 2D off the tank bottom.
3. The impeller shaft should make a plan angle of 8°–30° (10° optimum) to the left of the centerline of the tank.

For blending design and selection of mixing impellers, the fluids are divided into those below and above 50,000 cP [14]. Different impellers must be selected for the various ranges of viscosities, even within the 50,000 cP limits.

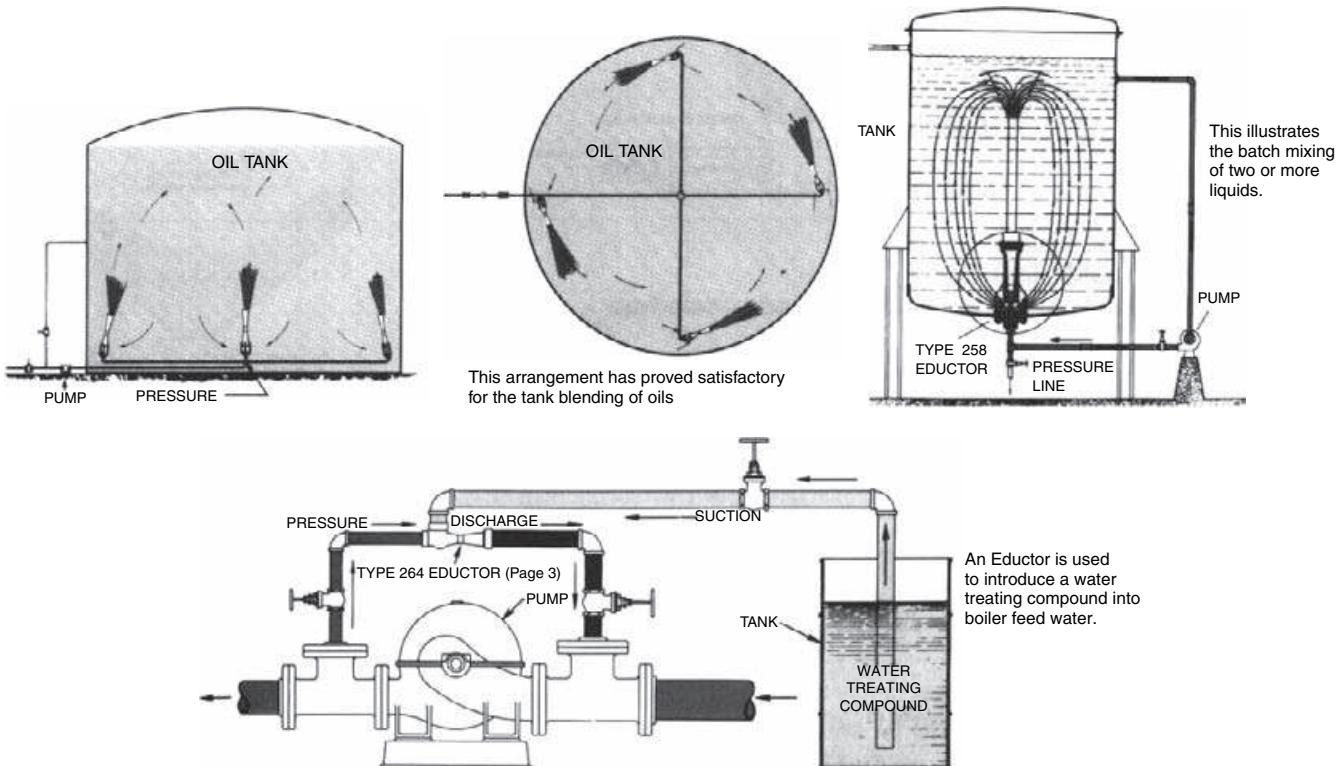
Blending is usually involved in developing uniform viscosities, densities, and temperatures. For best blending performance, the mixer should be operating while a second fluid is added to an initial tank of original fluid. This aids in preventing the tank contents from stratifying. To aid the manufacturer of the mixing equipment, the owner's engineer should provide viscosity and shear rate data for each of the fluids to be blended.

Jet mixing using liquid (recirculating or direct feed, Figures 7-48a and b) through single or a multiple jet nozzle arrangement (Figure 7-6w) have been studied for tank blending operations. Oldshue [14] points out that a mixer can be considerably more

efficient than jets producing the same flow of liquids. The jets are reported to be useful for mixing thick slurries, where settling with the agitator, not running, can pose a real problem for start-up [60, 61].



**Figure 7-48a** Liquid mixing jets. (By permission from Ketema, Schutte & Koerting Div.)



**Figure 7-48b** Illustration of jet mixing for blending of oils by circulation within the tank. Oil from the top is drawn down and entrains the oil in the bottom of the tank through the eductor nozzle (jet). (By permission from Ketema, Schutte & Koerting Div.)

## 7.19 EMULSIONS

Emulsions require high shear in the mixing operation with high speed and low  $D/T$  ratio.

## 7.20 EXTRACTION

The mass transfer in extraction equipment using mixers requires careful study before scale-up.

## 7.21 GAS-LIQUID CONTACTING

This is an important system in chemical processing. The effect of apparent density (liquid plus gas) as the fluid mixture enters the impeller is quite pronounced on the system horsepower. The horsepower falls off with increased gas flow which may lead to the danger of under powering the unit. The absorption coefficient is a function of the impeller size, its speed, and the inlet gas flow rate. For scale-up they should be handled in the form of the Sherwood number  $k_1(D/D_v)$  which can be related to power [23].

## 7.22 GAS-LIQUID MIXING OR DISPERSION

This is another common processing operation, usually for chemical reactions and neutralizations or other mass transfer functions. Pilot plant or research data are needed to accomplish a proper design or scale-up. Therefore, generalizations can only assist in alerting the designer as to what type of mixing system to expect.

This dispersion of the gas passes through several stages depending on the gas feed rate to the underside of the impeller and the horsepower to the impeller, varying from inadequate dispersion at low flow to total gas bubble dispersion throughout the vessel. The open, without disk, radial flow type impeller is the preferred dispersing unit because it requires lower horsepower than the axial flow impeller. The impeller determines the bubble size and interfacial area.

The gas dispersion ring or sparger can be a special design with holes or a single pipe entering the underside of the impeller, and there will be very little differences in mass transfer performance. Oldshue [10, 14] provides valuable detail for considering design for gas dispersion/mass transfer.

## 7.23 HEAT TRANSFER: COILS IN TANK, LIQUID AGITATED

Heat transfer during mixing of fluids in a tank depends to some extent on the degree of mixing, turbulence, and so on, affecting the heat transfer coefficient on the process side of the system and flowing against the coils, plates, or other surfaces for transfer. However, sizing an impeller or selecting an impeller to achieve a particular heat transfer coefficient has been proven to be impractical, because the coefficient is relatively independent of impeller speed [14, 62]. The heat transfer in a mixing vessel is by forced convection, and its heat transfer coefficient is usually one of the controlling factors to heat transfer. The other factors are cooling/heating side film coefficient, except when condensing steam, the scaling or fouling factors on the process side, and coolant/heating medium on the opposite side.

Despite the technical study and examination of this subject, it is important to recognize that because of the variety of factors noted earlier, the designer should not expect precise results and should allow considerable flexibility in the physical/mechanical design in order to adjust the system to achieve the required results.

## 7.24 EFFECTS OF VISCOSITY ON PROCESS FLUID HEAT TRANSFER FILM COEFFICIENT

Figure 7-49 presents a typical heating and cooling chart for the changes in process side film coefficients,  $h_o$ , as a function of bulk viscosity for organic chemicals.

The common arrangements for transferring heat to liquids in a tank are as follows:

1. vertical tube banks
2. helical coils
3. external jackets.

The range of chemical, petrochemical, and refining processes requires some type of heat transfer involving the mixing operation at the same time that some heat transfer is required. Typically, these include:

- reactions in mixing vessels with heat transfer to add or remove heat
- hydrogenation
- fermentation
- polymerizations
- general cooling
- general heating
- wide variety of processes including pharmaceuticals.

When heat transfer occurs in the presence of suspended solid particles, the chemical reaction may require mass transfer, and so on. However, the mixing part of the operation must be designed to accomplish the process results which may be those named above (and others) plus the required heat transfer affecting the outside film coefficient,  $h_o$ , across/around any coils, flat surfaces, or wall jackets.

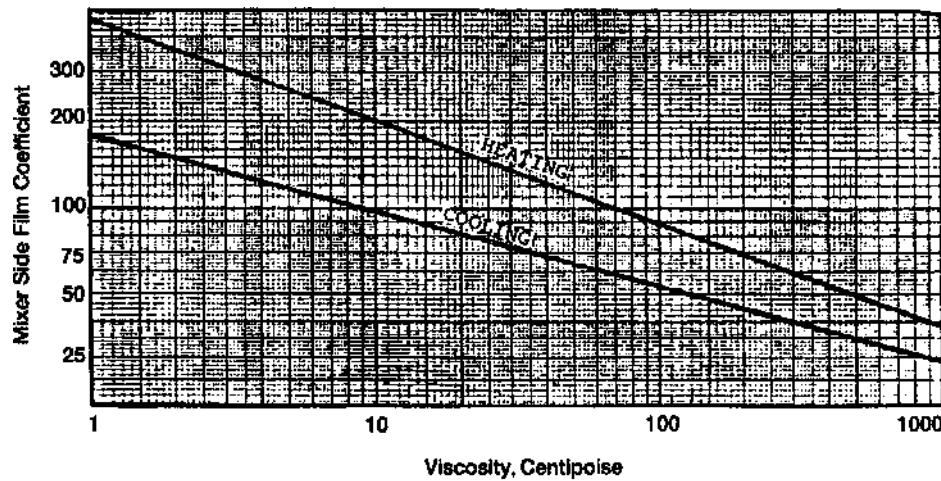
Heat transfer in these systems conforms to the general laws of other types of heat transfer, because heat is still being transferred through a barrier wall between two fluids in motion to determine the effective individual film coefficients.

$$q = U_o A \Delta T \quad (7-109)$$

where

- $U_o$  = overall heat-transfer coefficient referenced to outside heat transfer surface area,  $\text{Btu}/\text{h}/\text{ft}^2/\text{°F}$   
 $h_o$  = outside, process side in tank, film coefficient,  $\text{Btu}/\text{h}/\text{ft}^2/\text{°F}$   
 $k_w$  = thermal conductivity of material of heat transfer wall,  $\text{Btu}/\text{h}/\text{ft}^2/\text{°F ft}$   
 $A_n$  = area of outside coil or heat transfer barrier,  $\text{ft}^2/\text{ft}$   
 $A_i$  = area of inside of surface for heat transfer, such as coil, flat surface, or other barrier,  $\text{ft}^2/\text{ft}$   
 $h_i$  = inside heat transfer fluid side coefficient, in coil, flat plate, or other barrier,  $\text{Btu}/\text{h}/\text{ft}^2/\text{°F}$   
 $r_o$  = fouling resistance (factor) associated with fluid on outside (tank process side) of heat transfer barrier such as tube or flat plate/wall of vessel,  $\text{h ft}^2/\text{°F Btu}$   
 $r_i$  = fouling resistance (factor) associated with fluid on inside (heat transfer fluid) of heat transfer barrier such as tube or flat plate/wall of vessel,  $\text{h ft}^2/\text{°F Btu}$   
 $A_{avg}$  = average of inside and outside tube surface area,  $\text{ft}^2/\text{ft}$

$$\frac{1}{U_o} = \frac{1}{h_o} + r_o + \left( \frac{L_w}{k_w} \right) \left( \frac{A_n}{A_{avg}} \right) + r_i + \frac{1}{h_i} \left( \frac{A_i}{A_o} \right) \quad (7-110)$$



#### Thermal Effectiveness of Heat Transfer Surfaces

Tank Jacket .....	1.0
Vertical Tubes.....	1.54
Helical Coils (First Bank) .....	1.54
Helical Coils (Second Bank).....	1.31
Helical Coils (Third Bank) .....	1.08
Plate Coils (or Panel Coils).....	1.12

**Figure 7-49** Typical mixer-side film coefficients for heating and cooling outside vertical or single helical coils with organic chemicals being mixed and heated/cooled (thermal effectiveness = 1.54 compared to tank jacket = 1.0). (By permission from Lightnin Technology, Lightnin Technology Seminar, 3rd ed. 1982, p. 5, Section 2D, Lightnin (Formerly Mixing Equipment Co.), a unit of General Signal.)

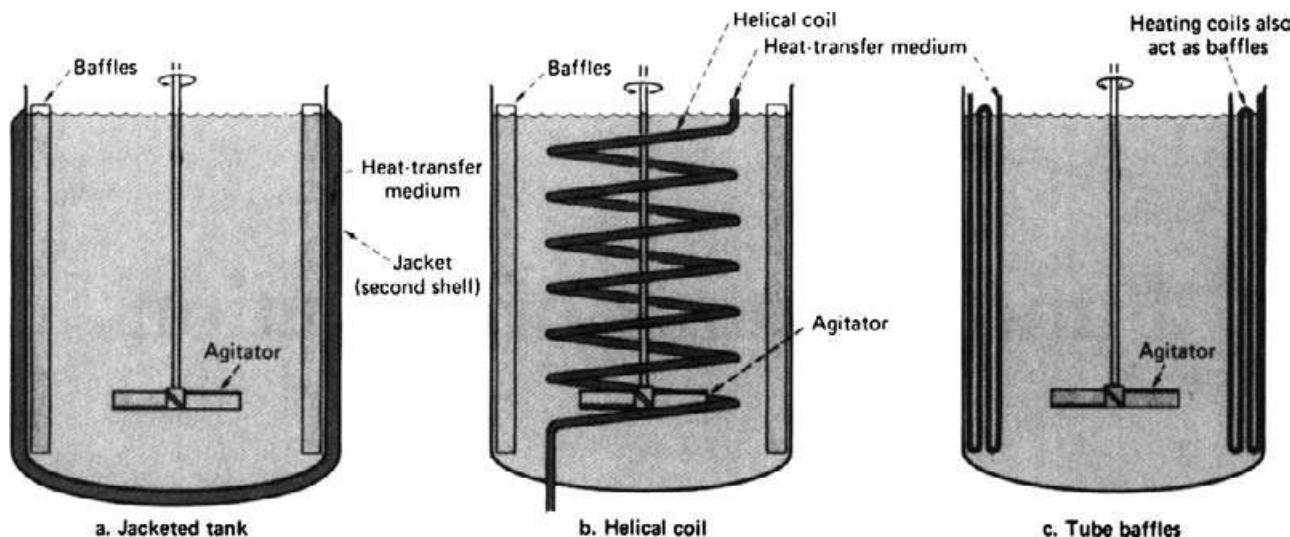
$r_w$  = resistance of tube wall,  $L_w/k_w$ ; h ft<sup>2</sup> F/Btu

$L_w$  = thickness of tube wall, ft

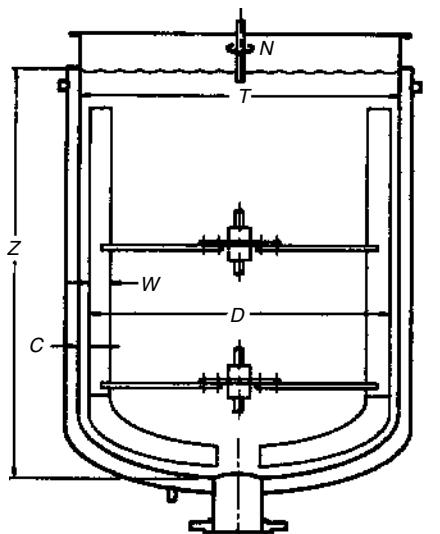
For estimating or even practical purposes, some of the components of the equation can be simplified. The ratio,  $A_i/A_o$ , can be used as  $(D_i/D_o)$ . Note that consistent units must be used for  $k_w$  and  $L_w$ .

In mixing, the moving liquid in a vessel establishes a heat transfer film coefficient on the surface of the heat transfer barrier such as tube coils, or the internal vessel shell wall with a jacket

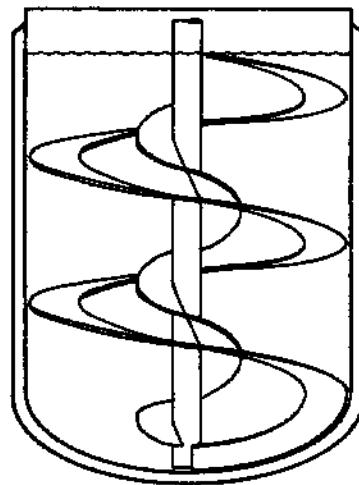
exterior to this wall for circulating heating and cooling fluid. This film becomes a function of the movement of the fluid against and/or next to the heat transfer barrier surface. Thus, the thinner the film, the better is the heat transfer. Therefore, the selection of the type of impeller, its rotational speed, and the fluid properties all influence the actual flow of heat through the film (Figures 7-50 and 7-51). Oldshue [14] identifies features of the system (vessel, impeller, and fluid properties) that influence the resulting heat transfer coefficient.



**Figure 7-50** Heat transfer surfaces in agitated tanks may be the actual wall of the vessel or immersed tubes. (By permission from Dickey, D.S. and R.W. Hicks, *Chem. Eng.*, Feb 2, 1976, p. 93 [62].)



JACKETED VESSEL WITH ANCHOR IMPELLER



JACKETED VESSEL WITH HELICAL IMPELLER

**Figure 7-51** Close-clearance anchor and helical impellers. (By permission from Oldshue, J.Y., "Fluid Mixing Technology", *Chem. Eng.*, McGraw-Hill Publications Co. Inc., 1983 [14].)

The generalized representation is given below:

$$N_{Nu} = \frac{h_o d}{k} = 0.17 N_{Re}^{0.67} N_{Pr}^{0.37} \left( \frac{D}{T} \right)^{0.1} \left( \frac{d}{T} \right)^{0.5} \left( \frac{\mu_w}{\mu} \right)^m \quad (7-111)$$

$N_{Nu}$  = Nusselt number

$$N_{Re} = \text{Reynolds number} = \left( \frac{D^2 N p}{\mu} \right) \cong \frac{10.754 D_i^2 N_m S_g}{\mu} \quad (7-112)$$

$$N_{Pr} = \text{Prandtl number} = \left( \frac{C_p \mu}{k} \right)$$

$$h_o \alpha \mu^{-3} D^{1.44} N^{0.67} T^{-0.6} k_w^{0.63} d^{-0.6} \rho^{0.67} C_p^{0.37}$$

where

$D$  = impeller diameter, ft

$N$  = impeller speed, rpm

$\rho$  = density, lb/ft<sup>3</sup>

$\mu$  = bulk viscosity of fluid

$\mu_w$  = wall viscosity at film process fluid temperature at heat transfer surface

$m$  = experimentally determined exponent, depending on bulk viscosity

$n_o$  = mean mixer side film coefficient of tank temperature

$k$  = thermal conductivity of fluid, Btu/h/ft<sup>2</sup>/°F/ft

$d$  =  $d_t$  = tube OD, ft

$T$  = tank diameter, ft

The vertical tubes serve as baffles to a certain extent, but not enough to prevent some vortex formation. The helical coil installations may have sidewall baffles (usually four 1/10 or 1/12 dia.), or baffles assembled with the coil itself (see Figures 7-19h, i, 7-50, and 7-51).

Figure 7-52 gives the heat transfer relations for a flat-paddle turbine and for anchors in a jacketed vessel and also in coil-tank arrangements. The data of Cummings and West [63] are based on large equipment and give results 16% higher for the coil and 11% higher for the jacket than the results of Chilton et al. [26].

Heat transfer data appear to be no better than ±20% when trying to compare several investigators and the basic fundamentals of their systems.

The exponents applying to each system are given on the figure.

The work of Uhl [64] gave particular emphasis to viscous materials in jacketed vessels, and the correlating equations are as follows:

1. Paddles, with or without baffles;  $N_{Re} = 20-4000$

$$\frac{hT}{k} = 0.415 \left( \frac{D^2 N p}{\mu} \right)^{0.67} \left( \frac{C_p \mu}{k} \right)^{0.33} \left( \frac{\mu_w}{\mu} \right)^{-0.24} \quad (7-113)$$

2. Turbine, no baffles;  $N_{Re} = 20-200$

Relation same as for paddle, coefficient changes from 0.415 to 0.535 when the turbine is about two-thirds of the distance down from top to bottom of the vessel. When the turbine is very near the bottom the coefficient is 0.44.

3. Anchor, no baffles

(a)  $N_{Re} = 300-4000$ , coefficient is 0.38 and  $\mu_w/\mu$  is raised to -0.18 power.

(b)  $N_{Re} = 30-300$ , coefficient is 1.00 and  $(\mu_w/\mu)^{-0.18}$  and Reynolds number term is raised to 0.50 power.

The correlations of Cummings and West [14, 63] for turbine mixers in vessels with jackets and coils are given below:

1. Vertical helical coils, multiple coils [14]

$$h_o(\text{coil}) \frac{d}{k} t = 0.17 \left( \frac{D^2 N p}{\mu} \right)^{0.67} \left( \frac{C_p \mu}{k} \right)^{0.37} \left( \frac{D}{T} \right)^{0.1} \left( \frac{d_t}{T} \right)^{0.5} \left( \frac{\mu}{\mu_w} \right)^m \quad (7-114)$$

$m$  can vary from 0.1 to 1.0

$$m = 0.1 (\mu 8.621 \times 10^{-5})^{-0.21}$$

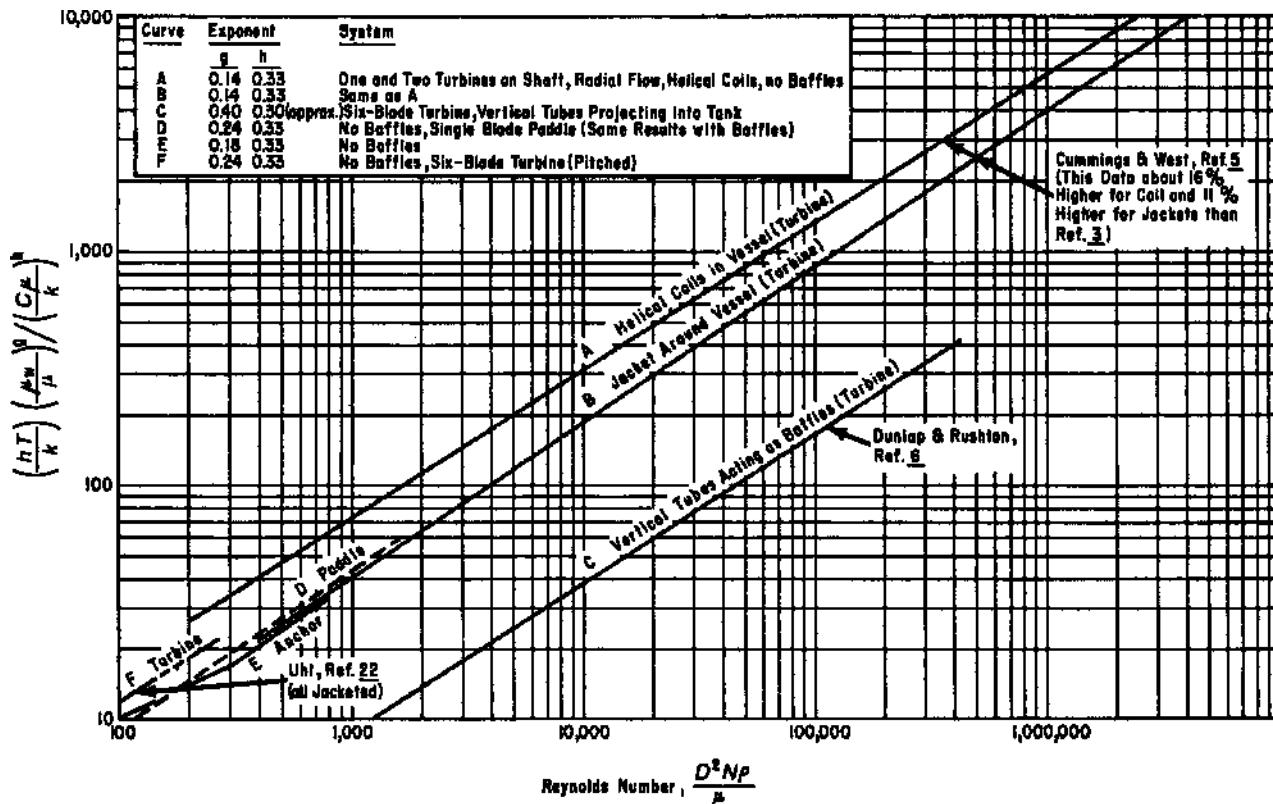


Figure 7-52 Liquid mixing heat transfer. (Compiled by permission from [26, 34, 63], and [64].)

The tube diameter,  $d_t$ , and tube spacing influence the film coefficient,  $h_o$ . The work of Oldshue [14] covers a practical range about as large and as small as is industrially used. The tube spacing of 2–4 tube diameters produced consistent results.

where

$D$  = impeller diameter, ft

$T$  = tank diameter, ft

$\mu$  = bulk viscosity

$\mu_w$  = wall viscosity at bulk process fluid temperature,  $d_t$

$d_t$  = tube diameter, ft, OD.

The helical coil relationship was developed using four vertical baffles of  $1/12T$ , located either outside the coil or inside the coil [14].

The addition of a second bank of similar coils for additional heat transfer area either outside or inside the first bank of coils will not provide twice the transfer of heat; but the effectiveness is estimated to be 70–90% of the first bank of duplicate coils (assuming the same heat transfer area per bank) [14]. The additional coils will provide additional baffling; therefore, the need for the dimension of the vertical baffles is reduced, or they might be removed entirely.

The value of,  $m$  (an experimental exponent) [62], will run between 0.1 and 1.0 for  $\mu$  ranging from 600 to 0.3 cP.

The particular work of Oldshue and Gretton [65] on heat transfer outside helical coils is of considerable general application. Refer to Eq. (7-114).

This is considered applicable to all sizes of tanks using flat, six-blade turbines, even with baffles on wall or inside single helical coil, providing that tube diameters are of  $0.018 \leq d_t/T \leq 0.036$ , with viscosity range probably to 10,000 cP. Tube spacing of coil wraps

of 2–4 tube diameters indicated no appreciable effect on the coefficient. The wider spacing gives a lower coefficient for materials above 50 cP. For example, the ratio of coefficient at  $4d$  and  $2d$  for water of 0.4 cP and oil of 50 cP were 0.96 and 0.88 respectively. The placement of baffles directly adjacent to the wall gives only a 5% better coefficient than if placed 1 in. off the wall or inside the helical coil. However, the power is about 10% lower when the baffles are off the wall.

#### 2. Vertical tubes [14]

Data of Dunlap and Rushton are given in [34]. This tube design can prevent the need for vertical baffles in a tank, and the heat transfer is good.

$$h_o(\text{tubes})d_t/k = 0.99 \left( \frac{D^2 Np}{\mu} \right)^{0.65} \left( \frac{C_p \mu}{k} \right)^{0.3} \left( \frac{D}{T} \right)^{0.33} \left( \frac{2}{n_b} \right)^{0.2} \left( \frac{\mu}{\mu_w} \right)^{0.14} \quad (7-115)$$

where

$n_b$  = number of tube baffles (vertical), that is, four or six banks of vertical tubes with three tubes per bank

$k$  = liquid thermal conductivity.

Although the outside coefficient of a vertical coil is some 13% higher than for a helical coil, the inside coefficient is quite often lower due to the physical arrangement and the lower coefficient if gases are evolved and venting is required. The overall coefficient may end up about the same as the helical coil. The outside film

**TABLE 7-13 Order of Magnitude of Outside Film Coefficients,  $h$ , in Turbulent Range**

Fluid	Helical Coils Btu/h ft <sup>2</sup> °F (between tank fluid and coil fluid)	Vertical Coils Btu/h ft <sup>2</sup> °F (between tank fluid and coil fluid)
Water, 0.38 cp	500–1100	500–1100
Oil, 10 cp	—	20–70
Oil, 52 cp	40–70	10–30

coefficient for a system varies with  $(HP)^{0.22}$  in the turbulent region. Thus

$$\frac{h_2}{h_1} = \left[ \frac{HP_2}{HP_1} \right]^{0.22} \quad (7-116)$$

The power required for vertical tubes in a vessel is 75% of that for standard wall baffles [65]. It is sometimes difficult to physically place as much vertical coil surface in a tank as helical coil surface. Dunlap studied vertical coils and the results are correlated for dimensionally similar systems by [14, 34].

This is shown in Figure 7-52 with certain simplifications to facilitate plotting with the other data. The four-blade turbine mixer was centered in the tank at about 1/2 of the fluid depth from the flat bottom. The vertical coils extended out into the tank in groups of three. The liquid depth was equal to the tank diameter.

Table 7-13 gives the order of magnitude for coil-in-tank heat transfer.

### 3. Vertical plate coils

The results of Petree and Small are summarized in [14]. These coils present a solid vertical face, with the "coils" vertical but impressed in the plates for flow of the heating or cooling medium. They take the place of vertical baffles, and are more solid obstructions to "through flow" in the vessel than individual vertical coils. Usually four or six banks are used.

For  $N_{Re} < 1.4 \times 10^3$  in fluid bulk in tank:

$$h_o(\text{plate coil}) (P_{pcw}/k) = 0.1788 \left( \frac{ND^2 \rho}{\mu} \right)^{0.448} \left( \frac{C_p \mu}{k_w} \right)^{0.33} \left( \frac{\mu}{\mu_f} \right)^{0.50} \quad (7-117)$$

$\mu_f$  = viscosity of fluid film at mean film temperature

$\mu$  = viscosity of fluid bulk at bulk temperature

$P_{pcw}$  = plate coil width, one plate, ft

For  $N_{Re} > 4 \times 10^3$  in fluid bulk in tank

$$h_o(\text{plate coil}) (P_{pcw}/k) = 0.0317 \left( \frac{ND^2 \rho}{\mu} \right)^{0.658} \left( \frac{C_p \mu}{k_w} \right)^{0.33} \left( \frac{\mu}{\mu_f} \right)^{0.50} \quad (7-118)$$

Some scale-up heat transfer relationships [14]:

- a. Laminar flow,  $N_{Re} < 100$ :  $h_o \propto HP$  (horsepower at the impeller shaft).
- b. For a constant impeller diameter, turbulent flow:  $h_o \propto (HP)^{0.22}$ .
- c. For constant speed, same impeller type (family):  $h_o \propto (HP)^{0.29}$ .

**TABLE 7-14 Constant ( $C$ ) for Various Impellers**

Agitator	Surface	C
Turbine	Jacket	0.62
Turbine	Coil	1.50
Paddle	Jacket	0.36
Paddle	Coil	0.87
Anchor	Jacket	0.46
Propeller	Jacket	0.54
Propeller	Coil	0.83

(Source: Chopey, N.P. and T.G. Hicks, *Handbook of Chemical Engineering Calculations*, McGraw-Hill Book Co., 1984.)

### 4. Jacket [14, 62]

$$\frac{h_o T}{k} = 0.85 \left( \frac{D^2 N \rho}{\mu} \right)^{0.66} \left( \frac{C_p \mu}{k_w} \right)^{0.33} \left( \frac{\mu}{\mu_f} \right)^{0.14} \left( \frac{Z}{T} \right)^{-0.56} \left( \frac{D}{T} \right)^{0.13} \quad (7-119)$$

The following equation can be used to predict heat transfer coefficients from coils to tank walls in agitated tanks.

$$\frac{h D}{k_f} = C \left( \frac{\rho N D_A^2}{\mu} \right)^{2/3} \left( \frac{C_p \mu}{k} \right)^{1/3} \left( \frac{\mu_b}{\mu_w} \right)^{1/4} \quad (7-120)$$

where  $C$  is a constant. Table 7-14 gives values of  $C$  for various agitator types and surfaces.

We have developed a software package (MIXER) to determine the heat transfer coefficient for any type of agitator and surface using the value in Table 7-14, fluid physical properties, agitator speed, and diameter. The Excel spreadsheet can also be used to determine  $h$  from Eq. (120) and Table 7-14.

## 7.25 HEAT TRANSFER AREA

Surface area for heating or cooling agitated vessels can be provided by either external jacketing or internal coils (or tubular baffles). Jacketing is usually preferred because of:

- cheaper materials of construction
- less tendency to foul
- easier cleaning and maintenance
- fewer problems in circulating catalysts and viscous fluids
- larger heat transfer surface.

Coils should be considered only if jacketing alone would not provide sufficient heat transfer area, if jacketing pressure exceeds 150 psig or high temperature vacuum processing is required. The coil offers the advantage of a higher overall film coefficient because of thinner walls with the latter conditions.

## DESIGN APPLICATION

A few typical overall heat transfer coefficients,  $U_o$ , are represented in Table 7-15 [12, 14].

Figure 7-53 indicates the mixing correlation exponent,  $x$ , as related to power per unit volume ratio for heat transfer scale-up. The exponent  $x$  is given in Table 7-16 for the systems shown, and is the exponent of the Reynolds number term, or the slope of the lines as given in Figure 7-53. It should be determined on any system not considered dimensionally similar or otherwise applicable to cases shown in Table 7-16 or Table 7-17.

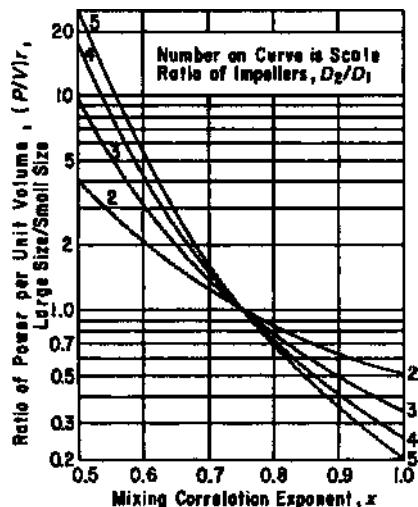
Maintaining  $(h_o)_1 = (h_o)_2$  on scale-up – for dimensionally similar systems [38]

**TABLE 7-15 Approximate Overall Heat Transfer Coefficients for Jacketed Mixing Vessels**

Fluid Inside Jacket	Fluid in Vessel	<i>U</i> Expressed in Btu/h/ft <sup>2</sup> /°F Wall Material	Agitation	<i>U</i>
Steam	Water	Enameled C.I.*	0–400 rpm	96–120[25]
Steam	Milk	Enameled C.I.*	None	200
Steam	Milk	Enameled C.I.*	Stirring	300
Steam	Milk boiling	Enameled C.I.*	None	500
Steam	Boiling water	Steel	None	187
Hot water	Warm water	Enameled C.I.	None	70
Cold water	Cold water	Enameled C.I.	None	43
Steam	Water	Copper	None	148
Steam	Water	Copper	Simple stirring	244
Steam	Boiling water	Copper	None	250
Steam	Paraffin wax	Copper	None	27.4
Steam	Paraffin wax	Cast iron	Scraper	107
Water	Paraffin wax	Copper	None	24.4
Water	Paraffin wax	Cast iron	Scraper	72.3

(Source: Adapted by permission from Lightnin/Unit of General Signal [12].)

\* C.I. = cast iron

**Figure 7-53** Scale-up relation for power, volume, and mixing slope. (By permission from Rushton, J.H. and J.Y. Oldshue, *Chem. Eng. Prog.*, Vol. 49, No. 4, 1953, p. 161 [8].)

- Determine from experimental data the Reynolds number exponent for the system, or use information in Figure 7-52 if the systems of Table 7-13 can be considered similar, use proper coefficients and solve for outside film coefficient,  $h_o$ .
- Referring to Figure 7-53, read the new approximate horsepower per unit volume ratio for the exponent  $x$  and the impeller ratio,  $D_2/D_1$ , where  $D_2$  is the larger impeller diameter. Calculate:

$$\left(\frac{P}{V}\right)_2 \Big/ \left(\frac{P}{V}\right)_1 \quad \text{Ratio from the curve, Figure 7-53} \quad (7-121)$$

- Determine impeller speed:

$$N_{2s} = N_{1s} \left(\frac{D_1}{D_2}\right)^{\frac{(2x-1)}{x}} \quad (7-122)$$

- Determine power using  $x$  and compare with step 5 below:

$$P_2 = P_1 \left(\frac{D_2}{D_1}\right)^{\frac{(3-x)}{x}} \quad (7-123)$$

- Determine tank volume:

$$V_2 = V_1 \left(\frac{D_2}{D_1}\right)^3 \quad (7-124)$$

and substitute in step 2, solving for  $P_2$ .

For a scale-up of impeller diameters of 2,  $D_2/D_1 = 2$ . Table 7-17 illustrates the changes to be expected. Such scale-up has been found to be quite reliable.

## 7.26 IN-LINE, STATIC, OR MOTIONLESS MIXING

This mixing device contains no moving parts, is relatively simple, and its cost can be quite reasonable when compared to mechanically driven mixers. Static or motionless mixers are a relatively new development and have proven to be effective for many specific and valuable process applications. Although useful for a wider range of viscosity fluids, some of the units have performed exceptionally well in the mixing of molten polymers. This type of unit is particularly useful in liquid-liquid mixing, although some units can be designed for solid-liquid dispersion and for gas-gas and gas-liquid mixing/dispersion. Some units are also reported suitable for solid-solid blending. The three commonly applied units are from Chemineer (Kenics U.S. Patent 3,286,992), Charles Ross and Son (Interfacial Surface Generator [Dow Chemical, U.S. Patent 3,168,390]), and Koch Engineering Co. (Sulzer Bros. of Winterthur, Switzerland, US Patent pending). Some designs work well on mixing powders. There are over 30 different models of static mixers worldwide [66].

The concept of the motionless mixer is to achieve a uniform composition and temperature distribution in fluids flowing through the device. Originally, this objective dealt with molten polymer mixing as an alternative to the dynamic mixing of an extruder screw [66]. The units are now used widely for just about all fluid mixing, blending, dispersion, and so on, by multiple splitting of the flowing streams. The concept is built around a stationary rigid element placed in a pipe or cylinder that uses the energy of the fluid flowing

**TABLE 7-16 Mixing Correlation Exponents for Various Systems**

Tank Configuration	Slope $x$ of Correlation Line	Reference
Jacketed cast iron hemispherical bottom vessel	0.67	1
Propeller – no baffles, $T/D = 2.5$		
U-Type impeller – no baffles, $T/D = 1.05$		
Helical coil, 9.6 in. diameter	0.62	3
1/2 in. tubing in 1 ft diameter tank		
Flat paddle, $T/D = 1.66$ close to bottom		
Liquid depth equal to tank diameter	0.67	3
No baffles $T/D = 24$		
Helical coil, 18 in. diameter	0.62	5
1 in. tubing in a 24 in. diameter tank		
Two curved-blade turbines	0.67	5
No baffles, $T/D = 2.5$ ; $T/D = 30$		
Four vertical tube baffles, 1 1/2 in. tubes	0.65	6
one flat-blade turbine.		
$T/D = 3$		
Turbine position one-half liquid depth $T/D = 25.3$		
Four vertical tube baffles, 1 in. tubes	0.90	19
One flat-blade turbine		
$T/D = 3$ ; $T/D = 37$		
Helical coil 34 in. diameter	0.67	13
1 3/4 in. tube in 4 ft diameter Tank		
One flat-blade turbine		
Four baffles each $1/12T$ , $T/D = 3$		
Turbine position 1/3 liquid depth, $T/D = 27.5$		
Paddle 14 1/8 in. $\times$ 2 3/8 in. close to bottom	0.67	22
$T = 141/8$ in.		
No baffles and 4 at 1/8 $T$		
Oils from 100 to 46,000 cSt		
Fan turbine 6 blades 12 in. 45° Pitch	0.67	22
Tank and liquids like above		
No baffles		
Same as above	0.67	22
Anchor impeller	0.5 up to $N_{Re} = 300$	22
2 1/2 in. diameter		
Tank and liquids like above		
	0.67 above $N_{Re} = 300$	

(Source: Extracted in part from Rushton J.H. and J.Y. Oldshue, *Chem. Eng. Prog.*, Vol. 49, No. 273, 1953 and the same presented at Philadelphia meeting AIChE, 1958, [18, 23] resp., by permission)

Note: Reference numbers refer to published article cited.

**TABLE 7-17 Example Scale-up of Two,  $D_2/D_1$ , for Varying Mixing Slopes**

Mixing Slope, $x$	$N_2/N_1$	$P_2/P_1$	$V_2/V_1$	$(P/V)_2/(P/V)_1$
0.5	1.0	32.0	8	4.0
0.6	0.8	16.0	8	2.0
0.75	0.63	8.0	8	1.0
0.9	0.54	5.0	8	0.62
1.0	0.50	4.0	8	0.50

(Source: By permission from Rushton, J.H., *Chem. Eng. Prog.*, Vol. 47, 1951, p. 485 [38].)

past to produce the mixing. The unit becomes more efficient by adding additional static elements (Figures 7-54a–p, 7-55, 7-56 and 7-57), causing the flowing fluid elements to split, rearrange, recombine and split again, and repeat the process ' $x$ ' times until a uniform or homogeneous flowing stream is produced at the discharge. The pressure drop varies with the mixer element's design, number per

unit length, and fluid flowing (density and viscosity). The shear on the fluid carries forward the mixing process.

Better mixing occurs under turbulent flow conditions.

Residence time in the units can be varied or adjusted, which makes them suitable to serve in certain types of reactions.

Quite often the mixing units or elements are installed in a circular pipe; however, they can be adapted to rectangular or other arrangements. Pressure drop through the units varies depending on design and whether flow is laminar or turbulent. Because of the special data required, pressure drops should be determined with the assistance of the manufacturer. Of course, pressure drops must be expected to be several multiples of conventional pipe pressure drop.

#### PRINCIPLES OF OPERATION

The Kenics static mixer unit is a series of fixed, helical elements enclosed within a tubular housing. The fixed geometric design of the unit produces the following unique patterns of flow division and radial mixing simultaneously. In laminar flow, a processed material divides at the leading edge of each element and follows the

**EXAMPLE 7-6**

Determine the heat transfer coefficient from a coil immersed in an agitated vessel with a diameter of 10 ft (3.048 m). The agitator is a paddle 3.5 ft (1.01 m) in diameter and turns at 200 rpm. The fluid properties are as follows:

$$\rho = \text{density} = 720 \text{ kg/m}^3$$

$$\mu_b = \text{viscosity} = 4.13 \text{ cP} = 4.13 \times 10^{-3} \text{ (Pa s)}$$

$$C_p = \text{specific heat} = 2.9 \text{ kJ/kg K}$$

$$k = \text{thermal conductivity} = 0.17 \text{ W/m K.}$$

$$\text{Assume } (\mu_b/\mu_w)^{0.14} = 1.0$$

*Solution*

From Table 7-14, for a paddle type agitator,  $C = 0.87$ . The heat transfer coefficient from Eq. (120) becomes

$$\frac{hD}{k_f} = 0.87 \left( \frac{\rho N D_A^2}{\mu} \right)^{2/3} \left( \frac{C_p \mu}{k} \right)^{1/3} \left( \frac{\mu_b}{\mu_w} \right)^{1/4}$$

$N$  = number of revolutions per second is  $200/60 = 3.3 \text{ rps}$ .

The Reynolds number,  $N_{Re} = \frac{\rho N D_A^2}{\mu}$  is:

$$N_{Re} = \frac{(720)(3.3333)(1.01)^2}{4.13 \times 10^{-3}} \left( \frac{\text{kg}}{\text{m}^3} \cdot \frac{1}{\text{s}} \cdot \frac{\text{m}^2}{\text{kg s}} \right)$$

$$= 592,794$$

The Prandtl number,  $N_{Pr} = \frac{C_p \mu}{k}$  is

$$N_{Pr} = \frac{(2.9 \times 10^3)(4.13 \times 10^{-3})}{0.17} \left( \frac{\text{J}}{\text{kg K}} \cdot \frac{\text{kg}}{\text{m s}} \cdot \frac{\text{s m K}}{\text{J}} \right)$$

$$= 70.45$$

The heat transfer coefficient is:

$$h_v = 0.87 \left( \frac{0.17}{3.048} \right) (592,794)^{2/3} (70.45)^{1/3}$$

$$h_v = 1414 \text{ W/m}^2 \text{ K}$$

Table 7-18 shows the computer results from the MIXER software.

**TABLE 7-18 Heat Transfer Coefficient for Fluids in a Vessel using Mechanical-Agitated Coils or Jacket**

Agitator:	Paddle
Surface:	Coil
Value of A:	0.870
Diameter of vessel, m:	3.048
Thermal Conductivity, W/m K:	0.170
Diameter of Agitator, m:	1.010
Speed of Agitator, rpm:	200.000
Density of Fluid, kg/m <sup>3</sup> :	720.800
Viscosity of Fluid, cP:	4.130
Specific Heat Capacity, kJ/kg K:	2.900
Viscosity at Bulk Fluid Temperature, cP:	1.000
Viscosity at Surface Temperature, cP:	1.000
Reynolds Number:	593453
Prandtl Number:	70.5
Heat Transfer Coefficient, W/m <sup>2</sup> K:	1415

channels created by the element shape. At each succeeding element, the two channels are further divided, resulting in an exponential increase in stratification. The number of striations produced is  $2^n$  where  $n$  is the number of elements (Figure 7-55).

### RADIAL MIXING

In either laminar or turbulent flow, rotational circulation of a processed material around its own hydraulic center in each channel of the mixer causes radial mixing of the material. All processed material is continuously and completely intermixed, resulting in virtual elimination of radial gradients in temperature, velocity and material composition.

The HEV Static mixer produces complete stream uniformity through controlled vortex structures generated by the mixing elements. The element geometry takes advantage of the naturally occurring vortices induced by the element edges. These principles

result in a reliable scale-up. In general, Kenics static mixer maximizes efficiency without the wasted energy and material blockage that occur in other restrictive motionless mixers.

The pressure drop in the Kenics mixer of the same length and diameter as an empty pipe can be determined from

$$\Delta P_{SM} = K \cdot \Delta P \quad (7-125)$$



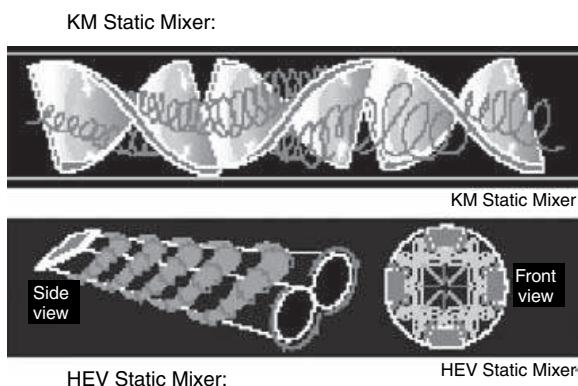
Application: Covers all ranges of Reynolds for both blending and dispersion. Heat transfer, extrusion, plug flow reactors, and solids dispersion.

**Figure 7-54a** KM Static mixer. (By permission from Kenics Corp., Div. Chemineer, Inc.)



Application: Used with low flow rates in full scale and pilot plant operations. Edge-sealed mixing elements attached to housing walls.

**Figure 7-54b** Tube mixer. (By permission from Kenics Corp., Div. Chemineer, Inc.)

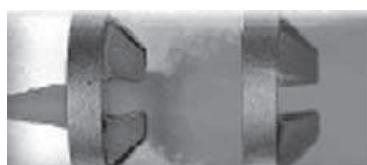


#### Principles of operation

In the KM Static mixer, the helical mixing element directs the flow of material radially toward the pipe walls and back to the center. Additionally, velocity reversal and flow division result from combining alternating right- and left-hand elements, increasing mixing efficiency. All material is continuously and completely mixed, eliminating radial gradients in temperature, velocity, and material composition.

In the HEV Static mixer, the tab geometry maximizes the conversion of turbulent energy into efficient mixing. The HEV produces complete stream uniformity through controlled vortex structures generated by the mixing elements. The element geometry takes advantage of the naturally occurring vortices induced by the element edges. These mixing principles result in an application technology that can be easily produced and reliably scaled.

**Figure 7-54c** KM and HEV Static mixers. (By permission from Kenics Corp., Div. Chemineer, Inc.)



Application: Turbulent blending static mixer. Each tab of the HEV static mixer generates a pair of stream-wise counter rotating vortices that produce vigorous cross-stream mixing and rapid uniformity.

**Figure 7-54d** HEV Static Mixer. (By permission from Kenics Corp., Div. Chemineer, Inc.)



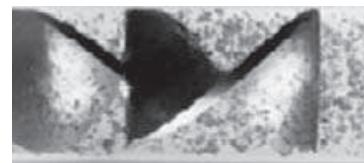
Application: Turbulent blending mixer. Produces rapid mixing by inducing circular patterns that reverse direction at each element intersection.

**Figure 7-54e** KM Static Mixer. (By permission from Kenics Corp., Div. Chemineer, Inc.)



Application: Laminar blending. The alternating helical elements of the mixer continually divide, stretch, and reorient the flow stream to produce complete mixing with minimum pressure drop.

**Figure 7-54f** KM Static mixer. (By permission from Kenics Corp., Div. Chemineer, Inc.)



Application: Liquid–liquid dispersion. Disperses immiscible liquids and produces a narrow drop size distribution.

**Figure 7-54g** KM Static mixer. (By permission from Kenics Corp., Div. Chemineer, Inc.)



Application: Gas–liquid dispersion. Gases can be incorporated into turbulent liquids. Mass transfer rates are dramatically enhanced to maximize absorption or reaction.

**Figure 7-54h** KM Static mixer. (By permission from Kenics Corp., Div. Chemineer, Inc.)



Application: Used for laminar, transitional, and turbulent flow. Suitable for most blending or dispersion problems involving liquids or gases. Mixing elements are attached to the housing wall.

**Figure 7-54i** Fixed Element KMS. (By permission from Kenics Corp., Div. Chemineer, Inc.)



Application: Mixing elements are inserted in the existing housing.

**Figure 7-54j** KMA-Element Assembly. (By permission from Kenics Corp., Div. Chemineer, Inc.)

**Application:** Used for maximum heat transfer, polymer reactors, certain fibrous applications, and mixing liquids with wide viscosity ratios. Mixing element edges are furnace-brazed to the housing walls eliminating dead areas.



Continuous joining of elements to the housing eliminates wall clearance to maximize heat conduction and minimize thermal degradation or fouling.

**Figure 7-54k** KME-Edge-Sealed element. (By permission from Kenics Corp., Div. Chemineer, Inc.)

**Application:** Used for laminar, transitional, and turbulent flow applications where periodic cleaning or inspection is required.



Suitable for most blending or dispersion involving liquids or gases.

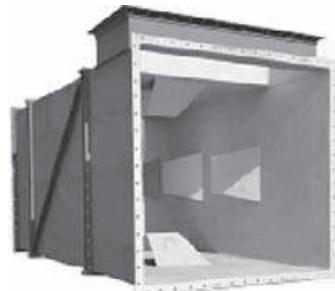
**Figure 7-54l** KMR-Removable element. (By permission from Kenics Corp., Div. Chemineer, Inc.)

**Application:** HEV high efficiency static mixers handle all turbulent flow mixing applications regardless of line size or shape. Mixing is accomplished by controlled vortex structures generated by the low profile tab geometry. This provides uniform blending while limiting mixer length to less than  $\frac{1}{2}$  pipe diameters. Complete mixing is achieved with pressure losses 75% less than conventional static mixers.



Low viscosity liquid-liquid blending processes, as well as gas-gas mixing. The HEV is ideal for processes where pressure loss and length are critical.

**Figure 7-54m** HEV Static mixer. (By permission from Kenics Corp., Div. Chemineer, Inc.)



**Figure 7-54n** HEV Static Mixer. (By permission from Kenics Corp., Div. Chemineer, Inc.)



**Application:** Used for process applications, for example polymers, plastics, adhesives, hydrocarbon processing, and food industries. Effective for both heating and cooling of viscous fluids.

**Figure 7-54o** Kenics heat exchanger. (By permission from Kenics Corp., Div. Chemineer, Inc.)



**Figure 7-54p** Static Mixers and mixing elements. (By permission from Kenics Corp., Div. Chemineer, Inc.)

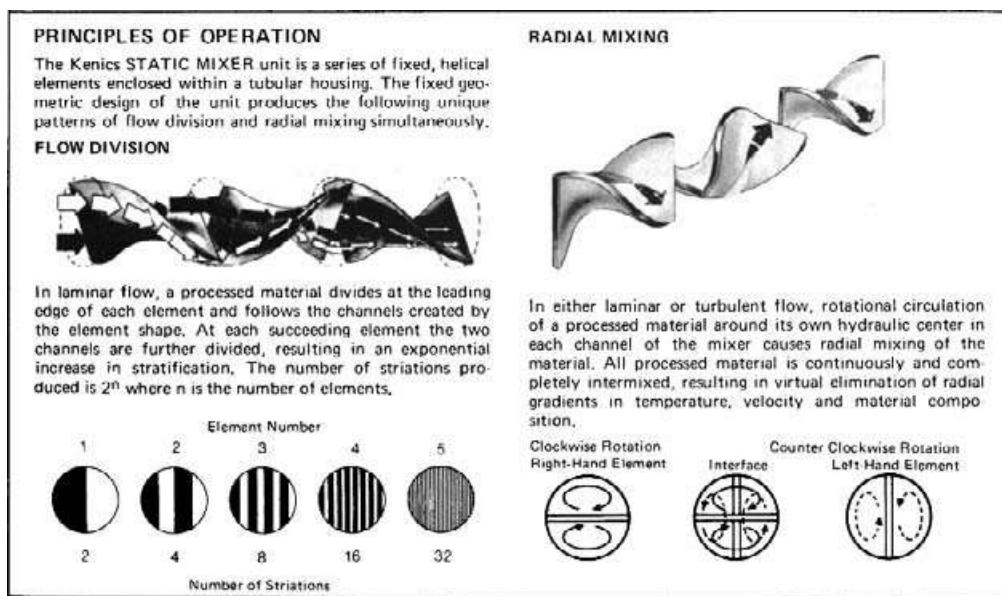


Figure 7-55 Principles of operation of static mixer modules. (By permission from Kenics Corp., Div. Chemineer, Inc.)

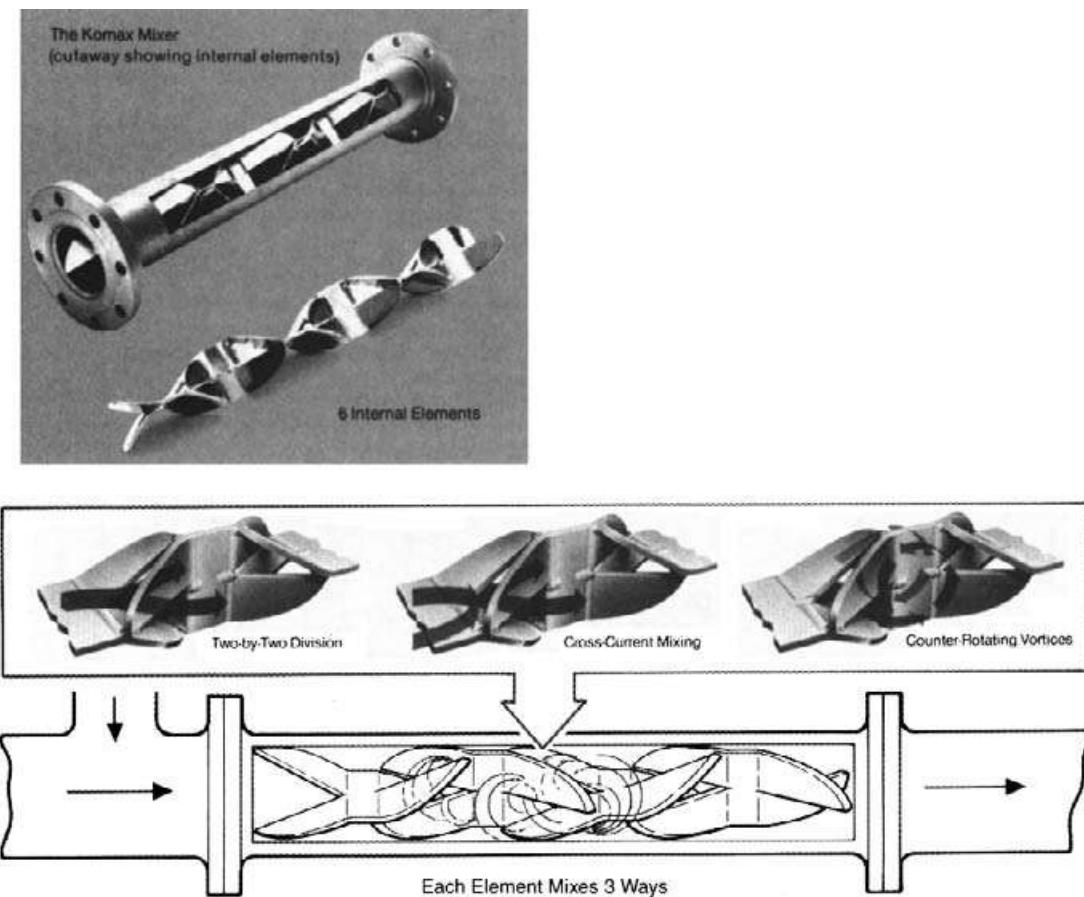
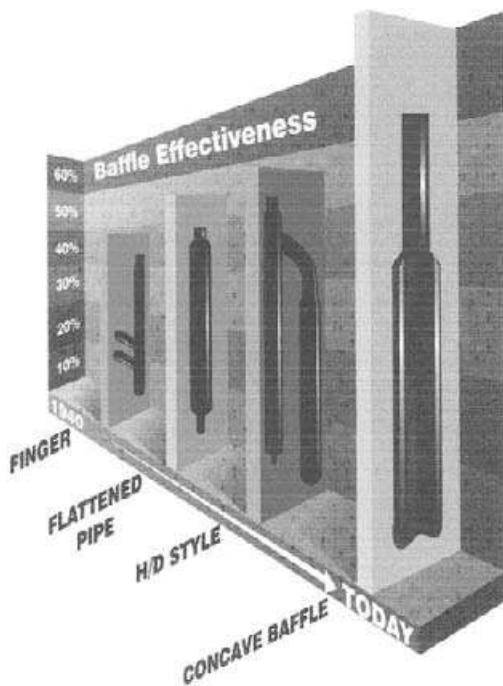


Figure 7-56a Komax™ motionless mixer. (By permission from Komax Systems, Inc.)



**Figure 7-56b** Improved baffle types. (Courtesy of Pfaudler-Balfour Inc.)

where

$\Delta P_{SM}$  is the pressure drop in the Kenics mixer and  $K$  is a function of the geometry of the mixer elements and the Reynolds number. The value for  $K$  is determined from Figures 7-58 and 7-59 depending on the type of flow regimes. Table 7-19 shows the Kenics mixer specification parameters for varying pipe sizes.

Since there are no moving parts in the Kenics mixer, only the processed materials are in motion. Therefore, the only energy required for the mixer is the energy required to overcome the pressure drop ( $\Delta P$ ). The general equation for calculating the pressure drop in an empty pipe for isothermal incompressible fluids is given by

$$\Delta P = 4f_F \left( \frac{L}{D_{SM}} \right) \frac{\rho v^2}{2g_c} \quad (7-126)$$

For laminar flow (i.e.  $N_{Re} < 2000$ ),  $\Delta P$  is given by

$$\Delta P = 3.4 \times 10^{-5} \left( \frac{L}{D_{SM}^4} \right) \left( \frac{\mu W}{\rho} \right) = 2.73 \times 10^{-4} \left( \frac{L}{D_{SM}^4} \right) (\mu Q) \quad (7-127)$$

$\Delta P$  in S.I. unit is

$$\Delta P = 113.2 \left( \frac{L}{D_{SM}^4} \right) \left( \frac{\mu W}{\rho} \right) = 6.79 \left( \frac{L}{D_{SM}^4} \right) (\mu \rho) \quad (7-128)$$

For turbulent flow,  $\Delta P$  is given by

$$\Delta P = 2.16 \times 10^{-4} f_D \left( \frac{L}{D_{SM}^5} \right) \rho Q^2 = 3.36 \times 10^{-6} f_D \left( \frac{L}{D_{SM}^5} \right) \left( \frac{W^2}{\rho} \right) \quad (7-129)$$

$\Delta P$  in S.I. unit is

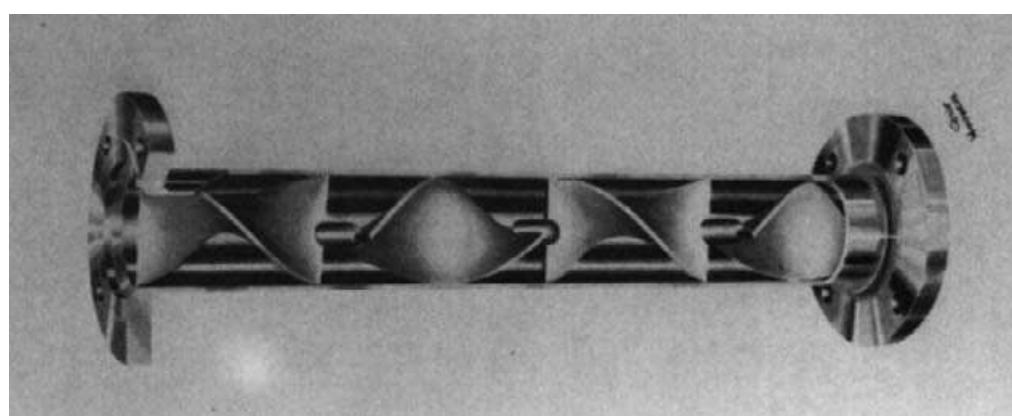
$$\Delta P = 2.252 f_D \left( \frac{L}{D_{SM}^5} \right) \rho Q^2 = 625.3 f_D \left( \frac{L}{D_{SM}^5} \right) \left( \frac{W^2}{\rho} \right) \quad (7-130)$$

where

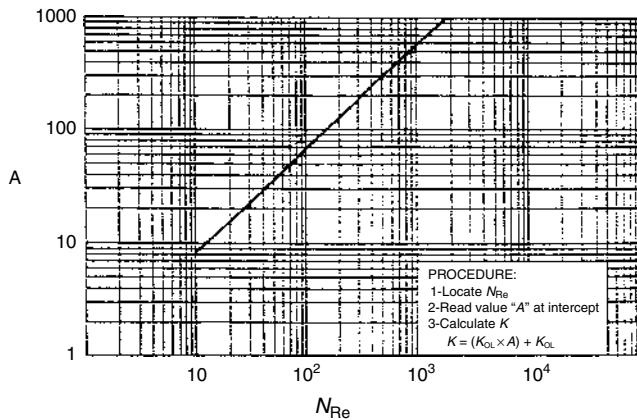
$\Delta P$	= pressure drop in a pipe, (psi, bar)
$\rho$	= fluid density, (lb/ft <sup>3</sup> , Kg/m <sup>3</sup> )
$f_D$	= Darcy friction factor, ( $f_D = 4f_F$ )
$L$	= length of a static mixer, (ft., m)
$v$	= fluid velocity, (ft/sec., m/sec)
$g_c$	= conversion factor (32.714 lb <sub>m</sub> /lb <sub>f</sub> . ft/sec <sup>2</sup> , 1 kg.m/N.sec <sup>2</sup> )
$D_{SM}$	= inside diameter of a static mixer, (inch., mm)
$Q$	= volumetric flow rate (U.S. gal/min, l/min)
$W$	= mass flow rate, (lb/hr, kg/hr).

The Reynolds number  $N_{Re}$  is determined by

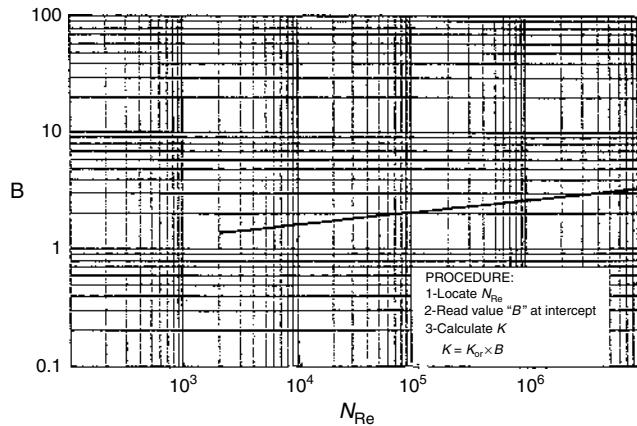
$$N_{Re} = 50.6 \frac{Q \cdot \rho}{\mu \cdot D_{SM}} = 6.31 \frac{W}{\mu \cdot D_{SM}} \quad (7-131)$$



**Figure 7-57** Luwa Blendrex™ motionless mixer. (By permission from Luwa Corp.)



**Figure 7-58** A-factor versus Reynolds number in the laminar flow region  $10 < N_{Re} < 2,000$ . (Source: Chen, S.J. *Kenics technical data KTEK-2*, 1978.)



**Figure 7-59** B-factor versus Reynolds number in the turbulent region  $10 < N_{Re} < 2,000$ . (Source: Chen, S.J. *Kenics technical data KTEK-2*, 1978.)

For S.I. unit,  $N_{Re}$  is determined by

$$N_{Re} = 21.22 \frac{Q \cdot \rho}{\mu \cdot D_{SM}} = 354 \frac{W}{\mu \cdot D_{SM}} \quad (7-132)$$

where

$\mu$  = fluid viscosity, cP.

For  $0 < N_{Re} < 2000$ , the friction factor is

$$f_D = \frac{64}{N_{Re}} \quad (7-133)$$

At higher Reynolds numbers, the friction factor is affected by the roughness of the surface, measured as the ratio  $\varepsilon/D$  of projections on the surface to the diameter of the pipe. Glass and plastic pipe essentially have  $\varepsilon = 0$ . Table 7-20 gives the pipe roughness of various materials. Figure 7-60 shows plots of Darcy friction factor versus Reynolds number for various pipe sizes. Alternatively, an explicit equation for the friction factor is given by [67].

$$\frac{1}{\sqrt{f_C}} = -4 \log \left\{ \frac{\varepsilon/D_{SM}}{3.7} - \frac{5.02}{N_{Re}} \log A \right\} \quad (7-134)$$

where

$$A = \frac{\varepsilon/D_{SM}}{3.7} + \left( \frac{6.7}{N_{Re}} \right)^{0.9}$$

$f_C$  = Chen friction factor.  
 $\varepsilon$  = pipe roughness.

A simpler explicit equation is given by [68]:

$$f = 1.6364 \left[ \ln \left( \frac{0.135\varepsilon}{D_{SM}} + \frac{6.5}{N_{Re}} \right) \right]^{-2} \quad (7-135)$$

The Darcy friction factor  $f_D$  is four times the Fanning ( $f_F$ ) friction factor (i.e.  $f_D = 4f_F = 4f_C$ ).

#### TWO-PHASE FLOW PRESSURE DROP (GAS-LIQUID) ESTIMATION

The Lockhart and Martinelli [69] correlation is employed to estimate the two-phase pressure drop in the Kenics mixer. We calculate the pressure drop for each phase assuming that each phase is flowing alone in the unit. The pressure drop for each phase is related to the X-factor defined by

$$X = \left( \frac{\Delta P_L}{\Delta P_G} \right)^{0.5} \quad (7-136)$$

where

$X$  = a factor (dimensionless)

$\Delta P_L$  = Pressure drop of liquid phase only (psi/element).

$\Delta P_G$  = Pressure drop of gas phase only (psi/element).

The two-phase pressure drop is obtained by multiplying either the liquid-phase drop by  $\phi_L^2$  or the gas-phase pressure drop by  $\phi_G^2$ . Figure 7-61 gives the Lockhart-Martinelli correlation between X and  $\phi$ 's.

where

$\phi_{LTT}$  = liquid-phase pressure drop correction factor with both phases in turbulent flow region.

$\phi_{LVV}$  = liquid-phase pressure drop correction factor with both phases in viscous (or laminar) flow region.

$\phi_{GTT}$  = gas-phase pressure drop correction factor with both phases in turbulent flow region.

$\phi_{GVV}$  = gas-phase pressure drop correction factor with both phases in viscous (or laminar) flow region.

The values midway between the TT and VV curves in Figure 7-61 are used for flow systems with either turbulent gas phase and viscous liquid phase or vice versa.

The total pressure drop is

$$\Delta P = \Delta P_L \cdot \phi_{LTT}^2$$

or

$$\Delta P = \Delta P_G \cdot \phi_{GTT}^2 \quad (7-137)$$

Coker [70] has developed a computer program that uses the Lockhart-Martinelli correlation to determine the total pressure drop of the two-phase flow based on the vapor phase pressure drop. The program also determines the gas-liquid phase regime employing a modified Baker's map. Static mixers generate smaller gas bubbles than stirred tank reactors (STR) and bubble columns, resulting in improved mass transfer rates 10–100 times those of a STR.

**TABLE 7-19 Kenics Mixer Specification**

Nom. Pipe Size	Housing Schedule.	Outside. Inch.	Dia mm	Inside Inch.	Dia mm	*Mod. Feet	Length <sup>†</sup> m	K <sub>OL</sub>	K' <sub>OL</sub>	K <sub>OT</sub>
1/2	40	0.84	21.34	0.62	15.75	0.51	0.16	6.00	0.075	40.7
3/4	40	1.06	26.92	0.82	20.83	0.65	0.20	5.23	0.050	23.5
1	40	1.32	33.53	1.05	26.67	0.90	0.27	5.79	0.069	36.3
1	80	1.32	33.53	0.96	24.38	0.85	0.26	5.57	0.062	31.4
1 1/2	40	1.90	48.26	1.61	40.89	1.27	0.39	5.72	0.071	36.8
1 1/2	80	1.90	48.26	1.50	38.10	1.27	0.39	5.53	0.065	32.6
2	40	2.38	60.45	2.07	52.58	1.71	0.52	5.70	0.068	35.1
2	80	2.38	60.45	1.94	49.28	1.67	0.51	5.54	0.062	31.6
2 1/2	40	2.88	73.15	2.47	62.74	2.30	0.70	5.04	0.053	24.3
2 1/2	80	2.88	73.15	2.32	58.93	1.92	0.59	5.58	0.066	33.8
3	40	3.50	88.90	3.07	77.98	2.82	0.86	4.94	0.052	23.6
3	80	3.50	88.90	2.90	73.66	2.82	0.86	4.82	0.049	21.4
4	40	4.50	114.3	4.03	102.36	3.37	1.03	5.08	0.058	26.9
4	80	4.50	114.3	3.83	97.28	3.18	0.97	5.16	0.060	28.2
6	40	6.63	168.4	6.07	154.18	4.88	1.49	5.19	0.060	28.6
6	80	6.63	168.4	5.76	146.30	4.88	1.49	5.08	0.057	26.2
8	40	8.63	219.2	7.98	202.69	6.26	1.91	5.14	0.061	28.4
10	40	10.75	273.05	10.02	254.51	7.79	2.37	5.07	0.060	27.8
12	40	12.75	323.85	11.94	303.28	9.66	2.94	4.88	0.056	24.8

<sup>†</sup> Add (T + 1/8) in. per flange up to a maximum of 3/8 in. per flange when ordering with flanges (T = wall thickness of housing) (1/8 in. = 3.18 mm, 3/8 in. = 9.53 mm).

\* Mod. Length is based on Model 6.

(Source: Chen, S.J., *Pressure Drop in the Kenics Mixer*, Kenics Technical data KTEK-2, 1978.)

**TABLE 7-20 Pipe Roughness of Materials**

Material	Pipe roughness ε(ft)	Pipe roughness ε(mm)
Riveted steel	0.003–0.03	0.9–9.0
Cast iron	0.00085	0.25
Galvanized iron	0.0005	0.15
Asphalted cast iron	0.0004	0.12
Commercial steel or wrought iron	0.00015	0.046
Drawn tubing	0.000005	0.0015

Recently, a static mixer has successfully been employed to enhance gas-liquid mixing and mass transfer in a process intensification of a packed-bed reactor. This resulted in increased productivity, elimination of a gel by-product in the reactor and thus avoiding the need for frequent shutdown [71].

In most process piping applications, the Kenics mixer diameter is the same as the existing process line diameter.

The K-factor for a specific process application is determined by the Reynolds number as follows [72]:

- If  $N_{Re}$  is less than 10,  $K = K_{OL}$ .
- If  $N_{Re}$  is between 10 and 2000, use Figure 7-58.
- If  $N_{Re}$  is greater than 2000, use Figure 7-59.

Multiply  $K$  by empty-pipe pressure drop to obtain the pressure drop caused by Kenics mixer module installation.

The theoretical horsepower (hp) required by the Kenics mixer is determined by

$$hp = (0.262)(\Delta P_{SM})(q) \quad (7-138)$$

where

$\Delta P_{SM}$  = pressure drop in the Kenics Mixer (psi).

$q$  = volumetric flow rate ( $\text{ft}^3/\text{sec}$ ).

To design the Kenics Mixer for a blending application, the procedure is as follows:

- calculate the Reynolds number ( $N_{Re}$ ) for the existing process line;
- choose the appropriate number of standard modules from Table 7-21;
- determine the pressure drop ( $\Delta P$ ) in the existing process line as a result of Standard Module installation.

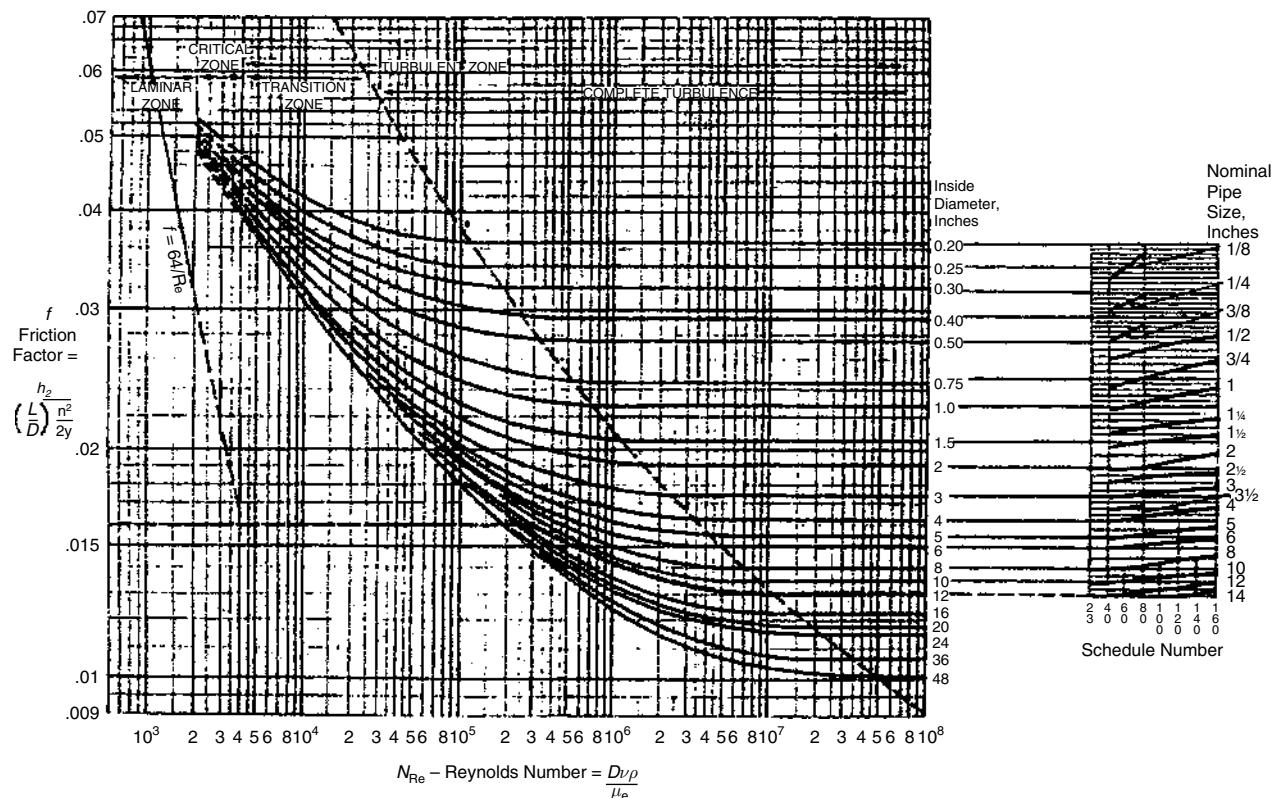
If  $\Delta P$  is high, repeat steps 1 through 3 using a larger mixer diameter.

Li et al. [73] performed an extensive study on  $\Delta P$  in a Sulzer SMX static mixer with both Newtonian and non-Newtonian fluids. They showed that  $\Delta P$  increased by a factor of 23 in a SMX static mixer in the laminar flow regime. Figures 7-62 shows their correlation between the Fanning friction factor and the Reynolds number for experimental points under various operating conditions.

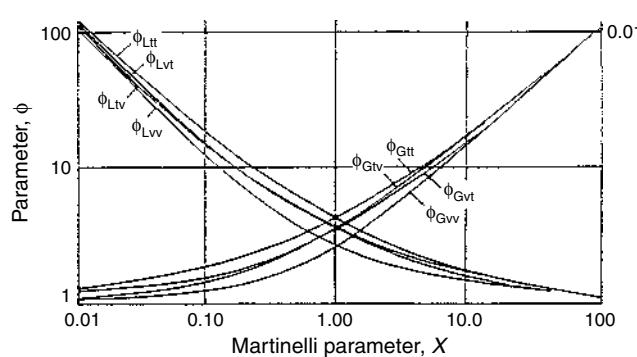
## APPLICATIONS

In addition to the general classification of applications previously mentioned, Tables 7-22a–c give typical applications. Although the number of modules or elements referred to is somewhat specific to the manufacturer, the tables give a general description of similar systems from other manufacturers.

Computational fluid dynamics (CFD) simulations are a valuable tool in studying the flow and mixing in static mixers [74]. Static mixer technology has been employed to solve many process mixing problems. Additionally, static mixers are used in tubular laminar flow reactors. Laminar open-pipe chemical reactors give a broad residence time distribution, which can be narrowed by the addition of helical elements thus approaching the ideal plug flow system. The design data and calculations of static mixers can be incorporated into knowledge-based computer programs that produce optimal and cost effective designs for a wide range of applications [75]. Figure 7-63 shows a flowsheet that illustrates the pertinent issues to consider when selecting the proper mixer.



**Figure 7-60** Friction factor chart for various pipe sizes.

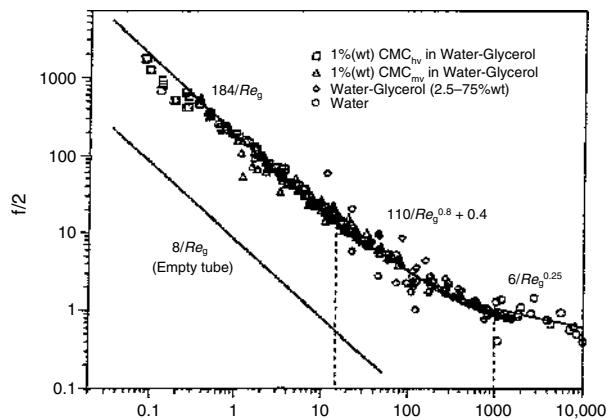


**Figure 7-61** Parameters for pressure drop in liquid/gas flow through horizontal pipes. (Source: Lockhead and Martinelli, *Chem. Eng., Prog.*, 45, 39, 1949.)

**TABLE 7-21** The Correct Number of Kenics Mixer Modules for Blending Applications

$N_{Re}$ (Based on empty pipe)	Standard Modules Required
< 10	4
10 to 1,000	3
1,000 to 2,000	2
> 2,000	1

One module consists of six mixer elements.  
Source: Chen, S.J., *Pressure drop in the Kenics mixer: Kenics Technical data KTEK-2*, 1978.



**Figure 7-62** Friction factor versus Reynolds number in Sulzer SMX Static mixer. (Source: Li et. al., *Trans. IChemE*, vol. 75, Part A, pp 792-798, 1997, Reproduced with permission of the Trans. IchemE.)

#### MATERIALS OF CONSTRUCTION

These elements that are inserted in a pipe or specially fabricated cylindrical holder can usually be fabricated from any workable and weldable metal or alloy. Additionally, most plastic that can be fabricated by molding, cutting, heat welding, or even bolting can be used. This wide array of fabrication materials allows the units to fit an extremely wide range of corrosive applications.

## 516 MIXING OF LIQUIDS

**TABLE 7-22a Typical Applications for the Motionless Mixers**

Application	Type of Flow	Number of Modules
1. Blend one grade of oil (or gasoline) into another oil	Turbulent	1 six-element module
2. Generate liquid-liquid dispersions (droplets)	Highly turbulent $N_{Re} > 100,000$	2 six-element modules
	Less turbulent $N_{Re} < 100,000$	3 six-element modules
3. Blend out thermal gradient in a viscous stream	Laminar	1 six-element module
4. Blend two resins to form a homogeneous mixture	Laminar	4 six-element modules
5. Dilution of molasses stream with water	Low turbulence	1 or 2 six-element modules, (number of modules depend on the flow rate and viscosity ratio)
6. pH control. Neutralization and treatment of waste water streams	Turbulent	1 four- or six-element module depending on reaction conditions
7. BOD treatment of water	Turbulent	1 six-element module
8. Gas-liquid dispersions	Turbulent	1 or 2 six-element modules
9. Solvent dilution	Turbulent	1 six-element module
10. Pipeline reactor for gaseous or liquid phase reactions	Turbulent	1 or 2 six-element modules
11. Gas dispersion in a liquid	Turbulent	Varies
12. Gas-gas mixing and/or dispersion	Turbulent	Varies

### MIXER DESIGN AND SOLUTION

For some fluid systems, the motionless mixer may not practically achieve total homogeneity. In some situations of widely diverse fluid densities, the centrifugal motion created may ‘throw’ some of the fluid to the outside of the flow path when it emerges from the unit. These are concepts to examine with the manufacturer, as only the manufacturer’s data can properly predict performance, and the design engineer should not attempt to actually physically design a unit.

Mixing fluids with viscosity ratios of over 1000:1 is one of the most difficult applications [76]. Mixing highly viscous fluids is also quite difficult.

The statistical measure of homogeneity is expressed as a function of the element geometry and the length of the unit (i.e., the number of elements in the mixer assembly).

$$\sigma = \sqrt{\frac{1}{n-1} \sum_{i=1}^n (x_i - \bar{x})^2} \quad (7-139)$$

**TABLE 7-22b Typical Applications for the Motionless Mixers**

Application	Type of Flow	Number of Modules
1. Blending catalyst, dye, or additive into a viscous fluid	Laminar	Usually 10–14. The exact number of elements will depend on the viscosity ratio
2. Delustering of polymer dope	Laminar	10
3. Blend out thermal gradients of polymer melt stream from extruder or heat exchanger	Laminar	4–6
4. Disperse solid particles in a viscous fluid	Laminar	10
5. Liquid-liquid blending to a homogeneous product	Turbulent flow	4–6
6. Solid-solid Blending of food products, explosives	Up to 2500 lb/h	6 elements placed in a vertical position
7. Waste water neutralization	Turbulent	4–6
8. Pipeline reactor to provide selectivity of product	Laminar	10
9. Multi-component epoxy dispensing systems	Laminar	10
10. Concrete or clay mixing	Laminar	10
11. Manufacture of powder coating	Laminar	4–6 (Mixer located downstream of extruder)
12. Thermal and color dispersion in blow-molding machinery	Laminar	4–6 (Mixer located downstream of extruder)
13. Manufacture of liquid or solid foods requiring sanitary construction	Laminar or Turbulent	4–10

where

$$\bar{x} = \frac{\sum_{i=1}^n x_i}{n}$$

$x_i$  = temperature, concentration, or some other measurable variable

$\frac{\sigma}{\bar{x}}$  = the lower this value, the more homogeneous the mixture; generally,  $\sigma/\bar{x}$  does not need to be lower than 0.05, that is, 5% standard deviation from the arithmetic mean

$\sigma$  = standard deviation

**TABLE 7-22c Application of Static Mixers in the Chemical Process Industry**

Industry	Applications
Chemicals	Chlorination and oxidation Steam injection Acid and base dilution Fast reactions
Food Processing	Acid washing of fats and oils Constituent blending Starch slurry and cooking
Mineral Processing	Slurry dilution Metal recovery by solvent extraction
Paints and Resins	Coloring and tinting Solvent blending
Petrochemicals and Refining	Emissions monitoring and control Gaseous reactant blending Gasoline blending Emissions monitoring and control
Pharmaceuticals	Nutrient blending Sterilization pH control
Polymers and Plastics	Reactant/catalyst blending Thermal homogenization Plug-flow finishing reactors
Pulp and Paper	Chemical and coatings preparation Stock dilution and consistency control Addition of bleaching chemicals
Water and Waste Treatment	Polymer dilution Disinfection, aeration and dechlorination

Source: Myers, K.J. et al., *Chem. Eng. Prog.*, June 1997, Reproduced with permission of AIChEJ. © Copyright 1997. All rights reserved.

$\bar{x}$  = arithmetic mean

n = number of samples.

The results of these calculations depend entirely on the manufacturer's design.

#### PRESSURE DROP

The energy to drive the fluid through a static mixer comes from the fluid pressure itself, creating a loss in pressure (usually small) as the fluid flows through the unit.

For laminar flow

$$\Delta P_l = 8.9 \times 10^{-8} (N_e Re_D) \frac{\mu M (L/D')}{\rho (D')^3} \quad (7-140)$$

where

$N_e$  = Newton number, depends on unit design. Can range from 0.8 to 1.9, for example.

$M$  = mass flow rate, lb/h

$\mu$  = absolute viscosity, cP

$\rho$  = density, lb/ft<sup>3</sup>

L = mixer length, in.

D' = inside pipe diameter, in.

$Re_D$  = Reynolds number related to inside diameter of pipe

$Q_g$  = flow rate, gpm.

For turbulent flow

$$\Delta P_t = (3.6 \times 10^{-5}) (N_e) \left( \frac{\rho Q_g^2}{(D')^4} (L/D') \right) \quad (7-141)$$

#### HORSEPOWER REQUIREMENT

The power requirements for the mixer element or mixing unit is expressed:

$$\text{Theoretical HP} = (5.83 \times 10^{-4})(Q_g)(\Delta P) \quad (7-142)$$

$\Delta P$  = pressure drop, psi

$Q_g$  = flow rate, gpm.

Reynolds number [10]

Compute the open  $N_{Re}$ :

$$N_{Re} = 3157 \frac{Q_g S_g}{\mu' D'} \quad (7-143)$$

where

$S_g$  = sp gr, dimensionless

$\mu'$  = viscosity, cP

$D'$  = inside pipe diameter, in.

$\Delta P$  = pressure drop, psi

k = viscosity correction factor for turbulent flow.

For laminar flow:  $N_{Re} < 2000$

$\Delta P_o$  for equivalent length of open pipe.

$$\Delta P_o = 2.73 \times 10^{-4} \frac{\mu' L Q_g}{(D')^4}, \text{ for one module} \quad (7-144)$$

L = length, ft

$S_g$  = sp gr

$Q_g$  = gpm

$\mu'$  = viscosity, cP.

For a laminar static mixer:

$$\Delta P = (7.4 + 0.07 N_{Re})(\Delta P_o) = \Delta P_L \quad (7-145)$$

for  $\Delta P$  (total), multiply ( $\Delta P_L$ ) (number of modules)

For turbulent flow:  $N_{Re} > 2000$

$\Delta P_o$  for open pipe, and  $f = N_{Re}$  vs  $f$  chart (see Chapter 4) for pipe

$$\Delta P_o = 0.0135 \left[ \frac{(f)(L)(S_g)(Q_g)^2}{D'^5} \right] \quad (7-146)$$

$$\Delta P_t = 66.5 N_{Re}^{0.086} \mu'^{0.064} (\Delta P_o) \quad (7-147)$$

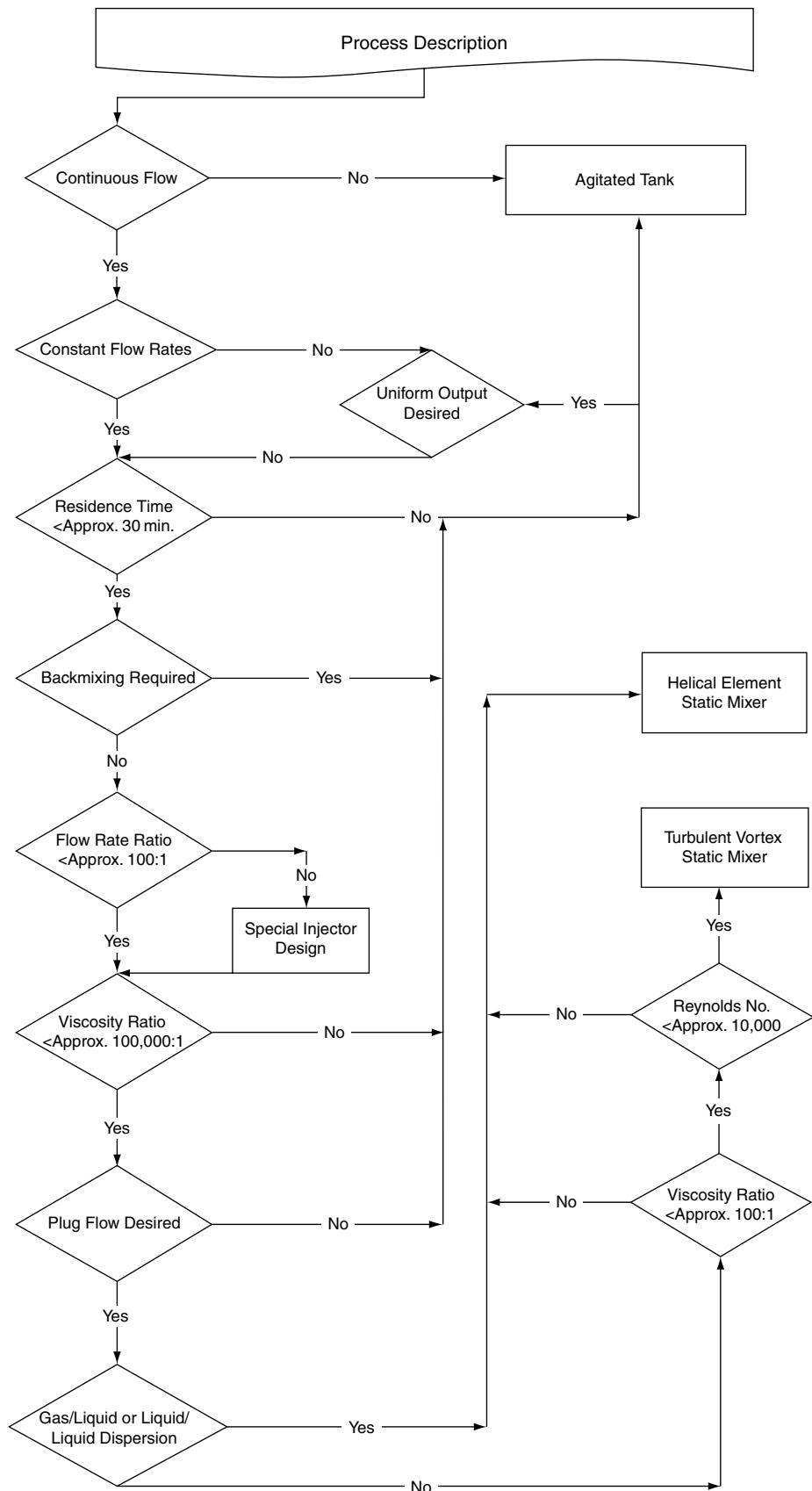
$\Delta P$  = open pipe pressure drop

$\Delta P_t$  = static mixer pressure drop in turbulent flow, psi.

For fluid velocities > 2.5 fps, and when lower volume fraction to be mixed is greater than 25% of total flow, use a dual turbulent module, that is,  $\Delta P_t$  would be 2x for one module.

The static mixer is also useful for direct contact heat transfer between fluids, two-phase contacting, and other useful applications such as mass transfer (see Figure 7-54h).

For final design details and selection of mixer elements, refer directly to the manufacturers, as each design is different and may not perform like a competitor's.



**Figure 7-63** Flow chart shows the main issues in deciding the use of either a static mixer or a agitated tank. (Source: Myers, K., et. al. *Chem. Eng. Prog.*, June 1997. With permission of AICHE. ©1997. All right reserved.)

**EXAMPLE 7-7**

How many Kenics Mixer modules of 3 in. Sch 40 are required to process a Newtonian fluid with a viscosity of 1,50,000 cP, a density of 60 lb/ft<sup>3</sup>, and a flow rate of 650 lb/hr? What are the pressure drop ( $\Delta P$ ) and the theoretical horsepower?

*Solution*

From Table 7-19

$$D_{SM} = 3.07 \text{ in.}$$

$$K_{OL} = 4.94$$

$$K'_{OL} = 0.052$$

$$K_{OT} = 23.6$$

By using Eq. (7-131), the Reynolds number is

$$\begin{aligned} &= 6.31 \cdot \frac{650}{(1.5 \times 10^5)(3.07)} \\ &= 0.0089(8.9 \times 10^{-3}) \end{aligned}$$

Since  $N_{Re} < 10$ , from Table 7-21 four modules are required.

By using Eq. (7-133) the Darcy friction factor  $f_D$  is

$$\begin{aligned} f_D &= \frac{64}{N_{Re}} = \frac{64}{(8.9 \times 10^{-3})} \\ &= 7185 \end{aligned}$$

From Table 7-19, the length of the Kenics mixer module is

$$L = 2.82 \text{ ft.}$$

Therefore, the length of four modules is

$$\begin{aligned} L &= 4 \times 2.82 \\ &= 11.28 \text{ ft.} \end{aligned}$$

The pressure drop  $\Delta P$  in the empty tube is

$$\begin{aligned} \Delta P &= 3.4 \times 10^{-5} \left( \frac{L}{D_{SM}^4} \right) \left( \frac{\mu W}{\rho} \right) \\ &= (3.4 \times 10^{-5}) \left( \frac{11.28}{3.07^4} \right) \left( \frac{1.5 \times 10^5 \times 650}{60} \right) \\ &= 7.02 \text{ psi.} \end{aligned} \quad (7-127)$$

Since  $N_{Re} < 10$ , Table 7-19 gives  $K = K_{OL} = 4.94$

By using Eq. (7-125) the pressure drop in four mixer modules is

$$\begin{aligned} \Delta P_{SM} &= K \cdot \Delta P \\ &= 4.94 \times 7.02 \\ &= 34.66 \text{ psi.} \end{aligned}$$

The required theoretical horsepower from Eq. (7-138) is

$$\begin{aligned} hp &= (0.262)(34.66) \left( \frac{650}{60 \times 3600} \right) \\ &= 0.027 \text{ hp.} \end{aligned}$$

**EXAMPLE 7-8**

Determine the number of mixer modules of a 2-in. Sch 40 required to process water-like fluids at a flow rate of 10 gpm. What is the required theoretical horsepower?

Data: viscosity = 1 cP  
Density = 62.4 lb<sub>m</sub>/ft<sup>3</sup>.

*Solution*

$$\varepsilon = 0.00015 \text{ ft.}$$

From Table 7-19,

$$D_{SM} = 2.07 \text{ in.}$$

$$K_{OL} = 5.70$$

$$K'_{OL} = 0.068$$

$$K_{OT} = 35.1$$

By using Eq. (7-131), the Reynolds number is

$$\begin{aligned} N_{Re} &= 50.6 \frac{Q \cdot \rho}{\mu \cdot D_{SM}} \\ &= 50.6 \frac{(10)(62.4)}{(1)(2.07)} \\ &= 15253.3(1.5 \times 10^4) \end{aligned}$$

Since  $N_{Re} > 2000$ , one module is required (Table 7-21). The Darcy friction factor  $f_D$  from Eq. (7-134) is

$$f_D = 0.0293$$

Length of the static mixer,  $L = 1.71 \text{ ft.}$

By using Eq. (7-129) the pressure drop ( $\Delta P$ ) in the empty pipe is

$$\begin{aligned} \Delta P &= 2.16 \times 10^{-4} (0.0293) \left( \frac{1.71}{2.07^5} \right) (62.4)(10^2) \\ &= 0.00177 \\ &= 1.8 \times 10^{-3} \text{ psi.} \end{aligned}$$

Since  $N_{Re} > 2000$ , from Figure 7-59,  $B = 1.85$

$$\begin{aligned} K &= K_{OT} \times B \\ &= 35.1 \times 1.85 \\ &= 64.94 \end{aligned}$$

The pressure drop of the Kenics mixer is

$$\begin{aligned} \Delta P_{SM} &= K \cdot \Delta P \\ &= 64.94 \times 1.8 \times 10^{-3} \\ &= 0.12 \text{ psi.} \end{aligned} \quad (7-125)$$

The required theoretical horsepower is

$$\begin{aligned} hp &= (0.262)(\Delta P_{SM})(q) \\ &= (0.262)(0.12) \left( \frac{10 \times 0.134}{60} \right) \\ &= 7.02 \times 10^{-4} \text{ hp.} \end{aligned} \quad (7-138)$$

**EXAMPLE 7-9**

A liquid-gas mixture is to flow in a 3-in. Sch 40 Kenics mixer. Estimate the pressure drop of the unit. The system conditions and physical properties are

	<b>Liquid</b>	<b>Gas</b>
Flow rate, lb/hr.	1000	3000
Density, lb/ft <sup>3</sup> .	62.4	0.077
Viscosity, cP.	1.0	0.00127
Surface tension, dyne/cm.	15.0	

Length of 3in.-Sch 40 pipe from Table 7-19 is 2.82 ft.

Assuming  $\varepsilon = 0.00015$  ft.

*Solution*

A computer program TWOPHASE has been developed that uses the Lockhart-Martinelli correlation and determines the total pressure drop based on the vapor phase pressure drop. The total length of the unit depends on the nature of the Reynolds number. The program also calculates the gas-liquid phase regime employing a modified Baker's map (see Chapter 4).

Table 7-23 gives the results of the two-phase pressure drop.

**TABLE 7-23 Two-phase pressure drop calculation of Example 7-9**

Pipe internal diameter, in.	3.070
Equivalent length of pipe, ft.	0.00
Actual length of pipe, ft.	2.82
Total length of pipe, ft.	2.82
Liquid density, lb/ft <sup>3</sup>	62.4
Liquid viscosity, cP	1.00
Surface tension, dyne/cm.	15.0
Liquid flowrate, lb/hr	1000.0
Liquid reynolds number	2055
Liquid friction factor	0.0311
Pressure drop of liquid per 100 ft. psi/100 ft.	0.0007
Gas flow rate, lb/hr.	3000.0
Gas density, lb/ft <sup>3</sup>	0.077
Gas viscosity, cP	0.001
Gas reynolds number	4743172
Gas friction factor	0.0174
Pressure drop of gas per 100 ft, psi/100 ft.	2.7313
Flow regime is	ANNULAR
Lockhart-martinelli two phase Modulus	0.0157
Velocity of Fluid in Pipe, ft/sec.	210.4991
Baker parameter in the liquid phase	1.644
Baker parameter in the gas phase	57508.66
Two-phase flow modulus	1.4561
Pressure drop of the two-phase Mixture, psi/100 ft.	3.9769
Overall pressure drop of the two-phase, psi	0.1121
Index 4553. is less than 10000. pipe erosion is unlikely	

**NOMENCLATURE**

<i>A</i>	Heat transfer area, referenced to inside, i, or outside, o, ft <sup>2</sup>
<i>A<sub>n</sub></i>	Area of outside coil or heat transfer barrier, ft <sup>2</sup> /ft
<i>A<sub>i</sub></i>	Area of inside surface for heat transfer, such as coils, flat surfaces, or other barrier, ft <sup>2</sup> /ft
<i>A<sub>avg</sub></i>	Average of inside and outside tube surface area, ft <sup>2</sup> /ft
<i>a</i>	Constant in Froude number exponent equation
<i>a</i>	Constant in Power number equation
<i>B</i>	Number of vertical wall baffles
<i>b</i>	Constant in Power number equation
<i>b</i>	Constant in Froude number exponent equation
<i>b</i>	Distance baffle off flat vessel bottom
<i>b<sub>a</sub></i>	Baffle clearance off tank bottom
<i>b<sub>w</sub></i>	Baffle width, in.
<i>C<sub>p</sub></i>	Specific heat Btu/lb °F
<i>cP</i>	Viscosity, cP
<i>c<sub>l</sub></i>	Distance baffles off wall
<i>C<sub>F</sub></i>	Correction factor for Reynolds number for viscosity effects
<i>D<sub>A</sub></i>	Impeller diameter, ft
<i>D<sub>i</sub></i>	Impeller diameter, in.
<i>D</i>	Impeller diameter, ft or <i>L</i>
<i>D'</i>	Inside diameter of pipe, in.
<i>D<sub>o</sub></i>	Diameter of jet from propeller mixer at origin, ft
<i>D<sub>T</sub></i>	Diameter of single pitch-blade turbine, under turbulent conditions, in.
<i>D<sub>T</sub></i>	Diameter of tank, ft
<i>D<sub>cor</sub></i>	Impeller diameter for turbulent regime, corrected for viscosity effect, in.

<i>D<sub>v</sub></i>	Diffusivity ft <sup>2</sup> /s
<i>d</i>	Tube outside diameter, ft
<i>d'</i>	Pipe inside diameter, ft
<i>d<sub>t</sub></i>	Tube diameter, OD, ft
<i>F</i>	Force (fluid) on turbine, perpendicular to shaft, ML/t <sup>2</sup>
<i>N<sub>Fr</sub></i>	Froude number
<i>f</i>	Function of _____
<i>g</i>	Acceleration of gravity, 32.2 ft/s <sup>2</sup>
<i>g<sub>c</sub></i>	Gravitational conversional factor, 32.174 lb (mass) ft/lb (force) s <sup>2</sup>
<i>H</i>	Total potential head during flow, ft of liquid
<i>H<sub>p</sub></i>	Motor, horsepower
<i>HP</i>	Impeller horsepower used by the system, ft lb/s, or HP
<i>h</i>	Film coefficient of heat transfer, kettle liquid to jacket wall or to coil, Btu/h ft <sup>2</sup> °F
<i>IC</i>	Clearance of impeller off tank bottom, in. equal to impeller (turbine) diameter, or <i>IC</i> = <i>D</i> ; sometimes <i>IC</i> = 2/3 <i>D</i> is suggested
<i>K</i>	Absorption coefficient
<i>K<sub>1</sub></i>	Proportionality constant, a function of the impeller shape, and is equal to 0.4 for three-blade propeller in water
<i>K<sub>2</sub></i>	Correlating factor for viscous flow power
<i>K<sub>3</sub></i>	Mixing factors, turbulent flow power
<i>k</i>	Viscosity correction factor for turbulent flow (static mixer), or
<i>k</i>	Thermal conductivity of heat transfer fluid (liquid), Btu/h ft <sup>2</sup> °F/ft
<i>k'</i>	Proportionality constant depending upon system, for blending
<i>k<sub>w</sub></i>	Thermal conductivity heat transfer wall, Btu/h ft <sup>2</sup> °F/ft

$k_1$	Liquid film mass transfer coefficient (lb moles) $\text{ft}^3/\text{s ft}$ (lb mole)	Z	Overall liquid vertical height of mixing vessel, from top liquid level to bottom (flat or dished or elliptical), ft or in., consistent with other components of equations
$L$	Static mixer length, in., or length of pipe in ft	$z$	Empirical constant
$L_w$	Thickness of heat transfer wall, ft		
$M^*$	Mass flow rate, lb/h		
$m$	Function of fluid properties, such as $\mu$ , $k$ , and $C_p$		
$N_s$	Shaft speed of rotation, rps		
$N_e$	Newton number, depends on design		
$N_F$	Force number, consistent units, dimensionless $= F/(PN^2 D^4)$		
$N_{Fr}$	Froude number		
$N = N_m$	Impeller speed of rotation, rpm		
$P_o = N_p$	Power number, dimensionless		
$N_{Pr}$	Prandtl number (heat transfer)		
$N_Q$	Flow number		
$N_{Re}$	Reynolds number, dimensionless		
$N_{nu}$	Nusselt number (heat transfer)		
$N_3$	Correlating factor for turbulent flow power		
$N_{We}$	Weber number		
$n$	Exponent in scale-up equation, describing type/degree of mixing required or number of samples in statistics, or		
$n$	Number of impellers		
$n_b$	Number of tube baffles (vertical)		
$P$	Power input to impeller, ft lb/s or if in a ratio can be as horsepower		
$P_{hp} = P'$	Power, horsepower used by impeller mixing system		
$P_o = N_p$	Power number, dimensionless		
$P_{pcw}$	Plate coil width, one plate, ft		
$\Delta p$	Pressure drop, psi		
$\Delta P_o$	Pressure drop for open pipe, psi		
$\Delta P_t$	Static mixer pressure drop in turbulent flow, psi		
$Q$	Flow rate or pumping capacity from impeller, $\text{ft}^3/\text{s}$ , or $L^3/t$		
$Q'$	Flow rate or pumping capacity from impeller, $\text{ft}^3/\text{min}$		
$Q_e$	Volume entrained into circular jet from propeller mixer, $\text{ft}^3/\text{s}$		
$Q_g$	Flow, gpm		
$R$	Scale-up ratio		
$r_w$	Resistance of heat transfer barrier wall $\text{h}/\text{ft}^2 \text{ }^\circ\text{F Btu} = L'_w/k_w$		
$S_g$	Fluid specific gravity (not density), referenced to water = 1.0		
$s$	Exponent of Schmidt group		
$T$	Tank diameter, ft, or $L$ (consistent units)		
$t$	Residence or holding time, s, or time of mixing		
$U_o$	Overall heat transfer coefficient, bulk mixing liquid to transfer fluid on opposite side of heat transfer wall (coil, plate, jacket), $\text{Btu}/\text{h}/\text{ft}^2/\text{ }^\circ\text{F}$		
$u$	Velocity of mixed fluids through mixer, ft/s		
$V$	Volume, consistent units		
$W$	Physical depth or height of turbine mixer, ft or in., consistent with other dimensions or		
$W$	Impeller blade width, ft		
$w$	Width of baffles in vertical tank		
$X$	Distance from impeller source, not to exceed 100 jet diameters, ft		
$x$	Mixing correlation exponent, or empirical constant		
$\bar{x}$	Arithmetic mean (statistics)		
$x_i$	Concentration of measurable variable		
$x_m$	Dimension of model		
$x_p$	Dimension of scale-up unit		
$x_R$	Ratio of dimensions on scale-up		
			<b>SUBSCRIPTS</b>
		r	Ratio of values of two conditions
		1	Initial condition
		2	Second condition
		f	Film
		i	Inside surface (heat transfer)
		o	Outside surface (heat transfer)
		G	Gravity
		I	Inertia
		R	Ratio
		v	Viscosity
		t	tube
			<b>GREEK SYMBOLS</b>
		$\theta$	Blending time, min
		$\mu_w$	Viscosity of liquids at wall surface, $\text{lb}/\text{s ft}$
		$\mu$	Viscosity in body liquid, $\text{lb}/\text{s ft}$
		$\mu'$	Fluid viscosity, cP
		$\rho$	Density, $\text{lb}/\text{ft}^3$
		$\sigma$	Standard deviation (statistics), or interfacial tension
		$\tau$	Torque on shaft, consistent units, $\text{FL}$ or $\text{ML}^2/\text{t}^2$
		$\phi$	$N_p = P_o$ = Power number, dimensionless
		$\Phi$	Power number, $P_o$ , or ratio of Power number to Froude number, $N_{Fr}$ , to exponential power, $n$
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[www.sulzerchemtech.com](http://www.sulzerchemtech.com)

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# 8

## EJECTORS AND MECHANICAL VACUUM SYSTEMS

### 8.1 EJECTORS

Industry has grouped pressures below atmospheric level into [1]:

Range of Pressure	
Rough vacuum	760–1 torr
Medium vacuum	1–0.001 torr
High vacuum	0.001–10 <sup>-7</sup> torr
Ultra-high vacuum	10 <sup>-7</sup> torr and below

Rough vacuum is used in about 90% of the chemical, petrochemical, and other processing industries. This range generally includes vacuum distillation, filtration, crystallization, drying, reaction, and others. Medium vacuum is most applicable to molten metals degassing, molecules distillation, freeze drying, and others. High and ultra-high ranges are most useful for thin films, research, and space simulation.

For reference, note that the unit of vacuum measurement is torr.

1.00 torr	= 1 mm mercury (mmHg), abs
25.4 torr	= 1 in. mercury (in. Hg)
750.1 torr	= 1 bar
1.868 torr	= 1 in. water at 4° C (in. H <sub>2</sub> O)
760 torr	= 1 standard atmosphere (atm) (at sea level)
Absolute pressure	= barometric pressure at location – vacuum

Statements about vacuum can be misleading when a clarification is not included. Vacuum refers to the “degree of emptiness” of a process system. A perfect vacuum represents an absolute zero of pressure, which is technically unmaintainable. A vacuum system indicates a system that can be a matter of the degree to which the system approaches absolute zero pressure. To create a vacuum in a fixed system, it is necessary to draw out or pump out the air in the volume. When part of the air is removed, the system has a partial vacuum. For example, when a 15-in. Hg vacuum is referenced to a 29-in. Hg barometer, then the absolute pressure is 29 – 15 = 14-in. Hg absolute. The 15-in. Hg vacuum can be considered a negative gauge reading.

See Chapter 4 for a diagrammatic relationship of pressure.

For quick reference, the listing below presents the most commonly used types of vacuum pumping equipment:

- liquid piston/ring
- centrifugal
- axial
- two-impeller straight lobe
- helical lobe
- reciprocating
- sliding-vane rotary
- ejector

- rotary oil-sealed
  - rotary piston type
  - vane type
- diffusion (not used for industrial/commercial application).

### 8.2 VACUUM SAFETY

Safety around mechanical vacuum pumps is possibly no different than that for other process mechanical rotating machinery. However, there is a decided danger of an implosion (collapse) of a tank, reactor, and other process equipments operating below atmospheric pressure, if:

1. It is not designed to satisfy the ASME codes for total or “full” vacuum, regardless of the expected actual operating vacuum on the equipment, vessel, and so on.
2. There are none or inadequate vacuum relief devices on the equipment or system being evacuated.
3. Block valves are installed to allow the blocking off of equipment (vessels, tanks, etc.) thereby pulling a higher vacuum than design, if not for “full” vacuum.

The implosion or collapse danger is real even for a tank, for example, that is not designed for vacuum (such as an API large storage tank), and liquid is pumped out of the tank thereby creating a negative pressure, or vacuum, which collapses the roof and/or sidewalls, because no or inadequate vacuum relief was installed to allow inflow of air as the liquid is removed (see Chapter 9).

4. Air inleakage, depending on the quantity, can create an explosive mixture in some process reaction systems; therefore, the system should be tested for air leaks and kept as tight as practical.
5. Also see Wintner [2].

### 8.3 TYPICAL RANGE PERFORMANCE OF VACUUM PRODUCERS

A useful summary of the typical capacities and operating ranges for vacuum equipment is presented in Table 8-1 (also see Birgenheier [4]). The positive displacement type vacuum pumps can handle an overload in capacity and still maintain essentially the same pressure (vacuum), while the ejectors are much more limited in this performance and cannot maintain the vacuum. The liquid ring unit is more like the positive displacement pump, but it does develop increased suction pressure (higher vacuum) when the inlet load is increased at the lower end of the pressure performance curve. The shapes of these performance curves are important in evaluating the system flexibility. See later discussion.

A simplified alternate to the previously cited procedures is suggested by Gomez [5] for calculating air inleakage, but it is not presented in detail here.

The two most common ejectors are operated by water (or process liquid) or steam. The liquid ejectors are used for creating a modest vacuum or for mixing liquids. The steam ejector is important in creating and holding a vacuum in a system. Ejectors

**TABLE 8-1 Typical Capacities and Operating Ranges for Vacuum Equipment**

Type	Lowest Recommended Suction Pressure	Capacity Range (ft <sup>3</sup> /min)
<i>Steam ejectors</i>		
One-stage	75 torr	10–1,000,000
Two-stage	12 torr	
Three-stage	1 torr	
Four-stage	200 micron*	
Five-stage	20 micron	
Six-stage	3 micron	
<i>Liquid ring pumps</i>		
60° F water-sealed		
One-stage	75 torr	3–10,000
Two-stage	40 torr	
Oil-sealed	10 torr	
Air-ejector first stage	10 torr	
<i>Rotary piston pumps</i>		
One-stage	20 micron	3–800
Two-stage	1 micron	
<i>Rotary vane pumps</i>		
Operated as a dry compressor	50 torr	20–6,000
Oil-sealed	1 torr	50–800
Oil-sealed, spring-loaded vanes		
One-stage	20 micron	3–50
Two-stage	1 micron	
<i>Rotary blowers</i>		
One-stage	300 torr	30–30,000
Two-stage	60 torr	
<i>Integrated pumping systems</i>		
Ejector-liquid-ring pump	150 micron	100–100,000
Rotary blower-liquid ring pump	1 micron	100–10,000
Rotary blower-rotary piston pump	0.001 micron	100–30,000
Rotary blower-rotary vane pump	100 micron <sup>†</sup>	100–30,000

By permission from Ryans and Croll [3].

\* 1 micron = 0.001 torr

† Based on two-stage, oil-sealed rotary-vane design that relies on centrifugal force to throw the vanes against the casing wall.

have no moving parts and operate by the action of one high pressure stream entraining air and other vapors (or liquids) at a lower pressure into the moving stream and thereby removing them from the process system at an intermediate pressure. Figure 8-1 illustrates the major components and the principle of operation. Since the steam jet ejector is the unit most commonly used for many process applications, it will be discussed in the greatest detail.

Referring to Figure 8-1, the high pressure steam enters the steam chest and expands in passing through the steam nozzle, leaving the nozzle at high velocity. Air, gas or vapor, or liquid mixture enters the ejector through the suction nozzle or vapor inlet, passing into the suction chamber. Here the air or other mixture is entrained into the high velocity steam. This new mixture enters the upper (or inlet) portion of the diffuser, passes through the diffuser throat (center narrow portion), and exits through the outlet end of the diffuser. In the diffuser, the velocity head of the mixture is converted back to a pressure which is higher than the air-mixture suction, but considerably less than the inlet steam pressure.

#### 8.4 FEATURES

Ejectors have the following features that make them good choices for continuously producing economical vacuum conditions:

1. They handle wet, dry, or corrosive vapor mixtures.
2. They develop any reasonable vacuum needed for industrial operations.

3. All sizes are available to match any small or large capacity requirements.
4. Their efficiencies vary from reasonable to good.
5. They have no moving parts; hence, maintenance is low and operation is fairly constant when corrosion is not a factor.
6. Quiet operation.
7. Stable operation within design range.
8. Installation costs are relatively low when compared to mechanical vacuum pumps. Space requirements are small.
9. Simple operation.

Ejectors are rather versatile when applied to a wide variety of processing movement, compression, mixing, operations, and so on. A brief listing of some useful functions are (by permission of [6]):

##### 1. Pumping and Lifting Liquids

- Using steam as the motive fluid
  - Steam jet syphons
  - Steam jet exhausters
  - Single-stage vacuum pumps
- Using air as the motive fluid
  - Air jet syphons
  - Air jet exhausters
- Using liquids as the motive fluid
  - Water jet eductors
  - Water jet exhausters

### Ejectors and Mechanical Vacuum Systems

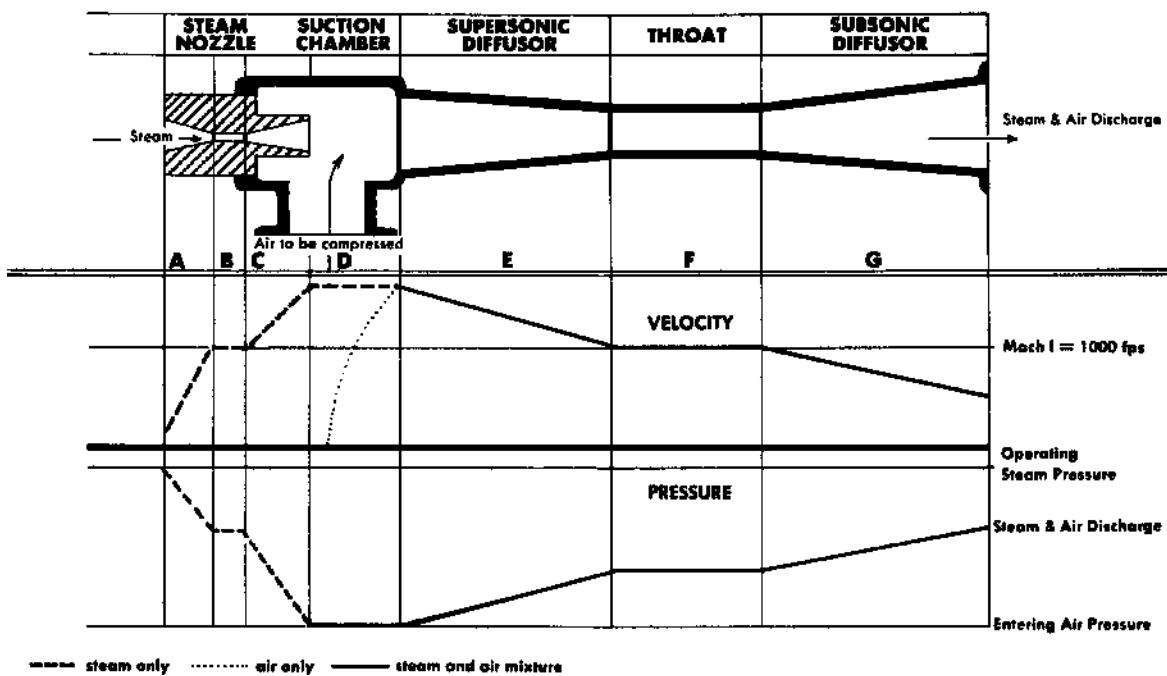


Figure 8-1 Basic ejector components and diagram of energy conversion in nozzle and diffuser. (By permission from Ingersoll-Rand Co.)

#### 2. Heating Liquids (by Direct Contact)

- Tank-type heaters
  - Steam jet heaters
- Pipeline-type heaters
  - Steam jet heaters
  - Steam jet heaters (large capacity)
  - Steam jet siphons
- Open-type heaters
  - Steam jet heaters (large capacity)

#### 3. Moving Air and Gases (and Pump Priming)

- Using steam as the motive fluid
  - Steam jet blowers
  - Steam jet exhausters
  - Steam jet thermocompressors
  - Single-stage vacuum pumps
  - Multi-stage vacuum pumps
- Using air as the motive fluid
  - Air jet blowers
  - Air jet exhausters
  - Single-stage vacuum pumps
  - Air jet compressors
- Using gas as the motive fluid
  - Gas jet compressors
- Using liquid as the motive fluid
  - Water jet exhausters
  - Barometric condensers
  - Low level condensers
  - Water jet eductors (small capacities)

#### 4. Handling Slurries and Granular Solids

- Using steam as the motive fluid
  - Steam jet siphons
  - Steam jet slurry heater
  - Single-stage vacuum pumps

#### • Using air as the motive fluid

- Air jet exhausters

#### • Using liquid as the motive fluid

- Water jet eductors

#### 8.5 TYPES

Ejectors may be single or multi-stage and also multijet inside a single housing or stage. The extra stages, with or without interstage condensing of steam, allow the system to operate at lower absolute pressures than a single-stage unit. Various combinations of series of jets with no intercondensing can be connected to jets with intercondensers or aftercondensers to obtain various types of operation and steam economy. The condensers may be barometric or surface type.

Figure 8-2 suggests a few of the many uses for which ejector-type units are used in industry.

Figure 8-3 illustrates a single-stage non-condensing ejector. In this type of installation, the steam outlet from the ejector is either exhausted to atmosphere or on top of water in a sump.

Figure 8-4 shows two individual single-stage ejectors discharging into a common surface aftercondenser. The steam condensate can be reused from this installation.

Figures 8-5 and 8-6 illustrate two-stage ejector installations with barometric and surface type inter/after condensers respectively. The discharge of the steam non-condensables from the second-stage jet of Figure 8-5 is exhausted to the atmosphere, while in Figure 8-6 the steam is condensed in the aftercondenser and, essentially, only non-condensables leave the vent of the aftercondenser. Figure 8-7a indicates a diagram of a three-stage barometric type installation.

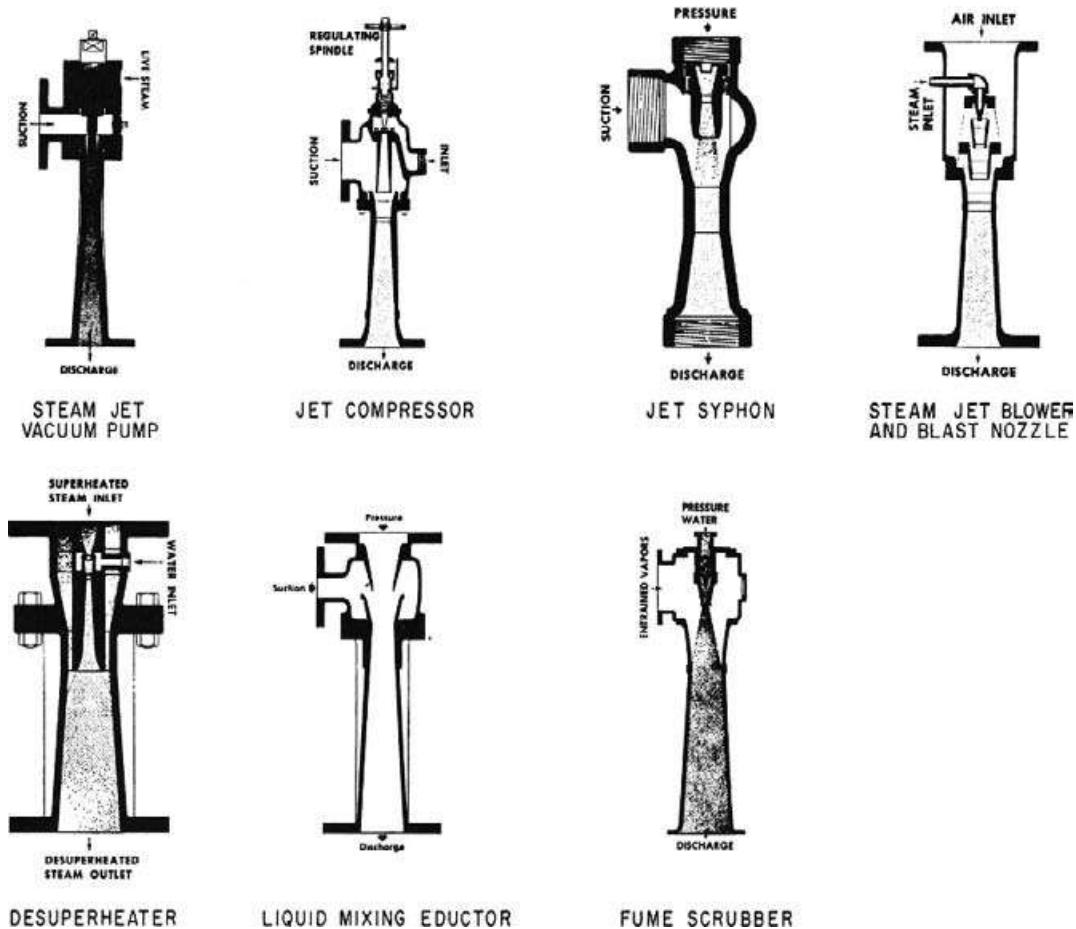


Figure 8-2 Steam, air, gas, and liquid ejectors. (By permission from Ketema, Schutte & Koerting Division.)

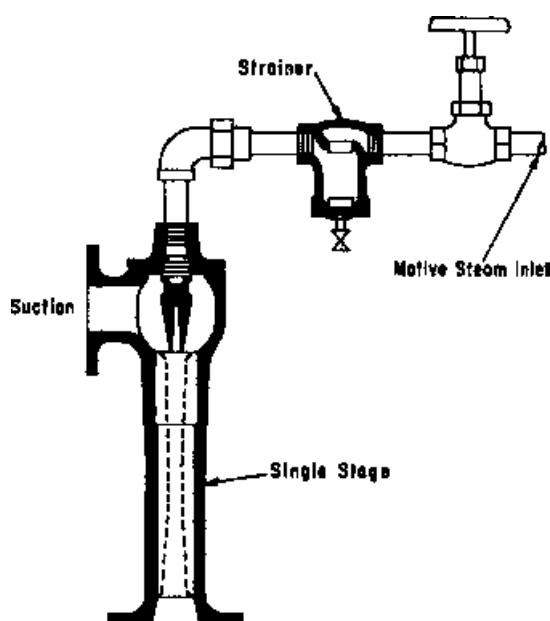


Figure 8-3 Single-stage non-condensing ejector. (By permission from C.H. Wheeler Mfg. Co.)

Figure 8-7b illustrates a barometric refrigeration unit, generating chilled water in the range of 34–55°F for process cooling. It uses steam ejectors to lower the chill tank's vapor pressure to establish boiling/evaporation of the water in the tanks and condenses the vapors released by plant cooling water in the barometric condensing unit, which is sealed through the vacuum leg into a "hot" well.

Figure 8-8 illustrates various arrangements of ejectors with inter- and aftercondensers. The condensers can be barometric or non-barometric types.

Note that in Figure 8-8 and Table 8-2 the letter designations of the stages conform to the latest Standards of the Heat Exchange Institute for Steam Jet Vacuum Systems [7]. The letter designates the jet's stage position in the system.

Precondensers are recommended for any ejector system when the pressure conditions and coolant temperature will allow condensation of vapors, thus reducing the required design and operating load on the ejectors. This is usually the situation when operating a distillation column under vacuum. The overhead vapors are condensed in a unit designed to operate at top column pressure, with only the non-condensables and vapors remaining after condensation passing to the ejector system.

Intercondensers are used to condense the steam from a preceding ejector stage, thus reducing the inlet quantity of vapor mixture to the following stage. This is a means of increasing steam economy.

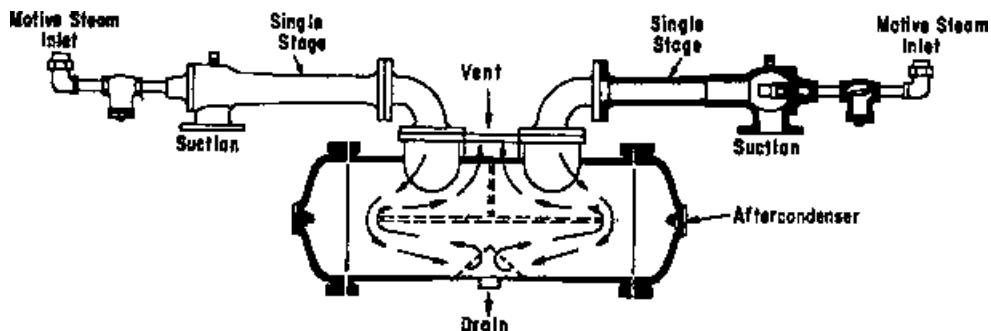


Figure 8-4 Twin single-stage ejectors with surface aftercondenser. (By permission from C.H. Wheeler Mfg. Co.)

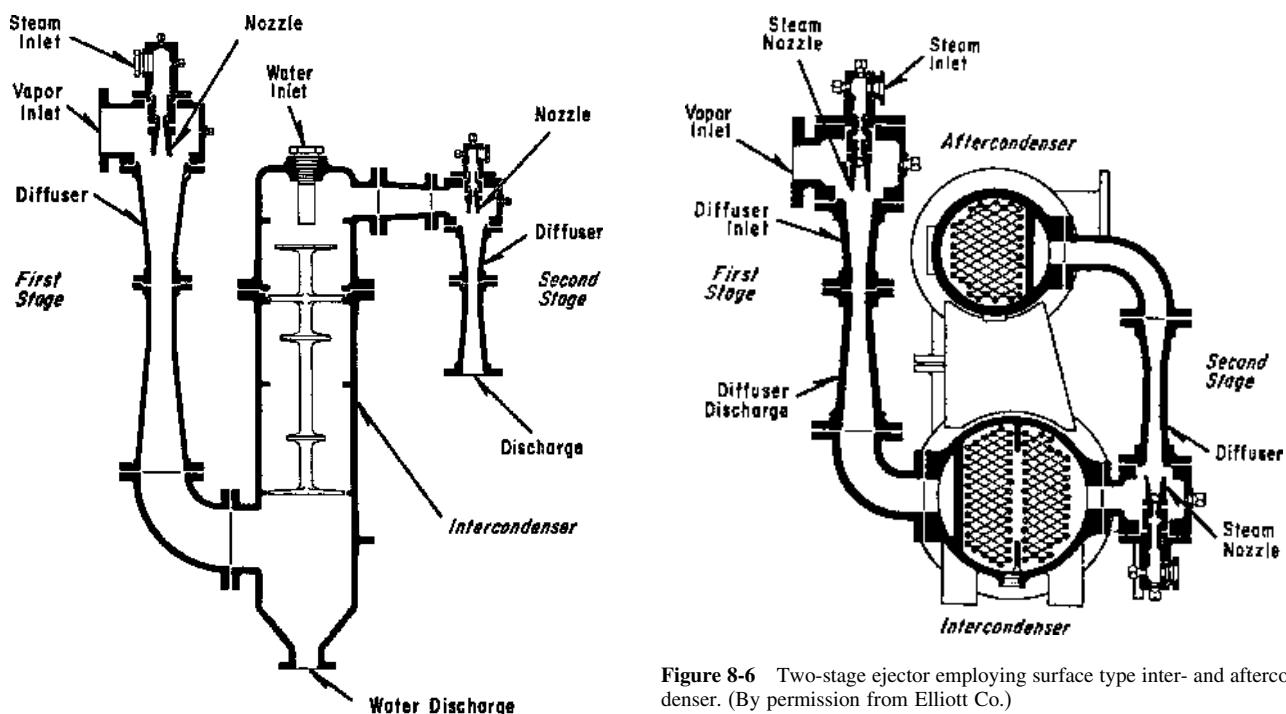


Figure 8-5 Two-stage ejector using barometric-type intercondenser. (By permission from Elliott Co.)

Aftercondensers operate at atmospheric pressure. They do not affect the steam economy or ejector performance, but they do avoid the nuisance of exhausting steam to the atmosphere, thus allowing steam to be recovered. They also serve as silencers on the ejectors, and with barometric types they can absorb odors and corrosive vapors.

Condenser tail pipes, used with any condenser, are sealed with a 34-ft leg into a sump, or with a condensate pump operating under vacuum on suction. With surface-type condensers, the level may be sealed in a receiver with a float or other type of level control.

Thermocompressors are steam jet ejectors used to boost low pressure or waste steam to a higher intermediate pressure. Single-stage units are usually not used for compression ratios (ratio of absolute discharge to suction pressures) greater than three [8]. This type of pressure increase for low pressure steam is usually uneconomical when the final discharge pressure exceeds one-third of the high pressure motive steam [8]. These units are usually limited to single-stage installations based on steam economy.

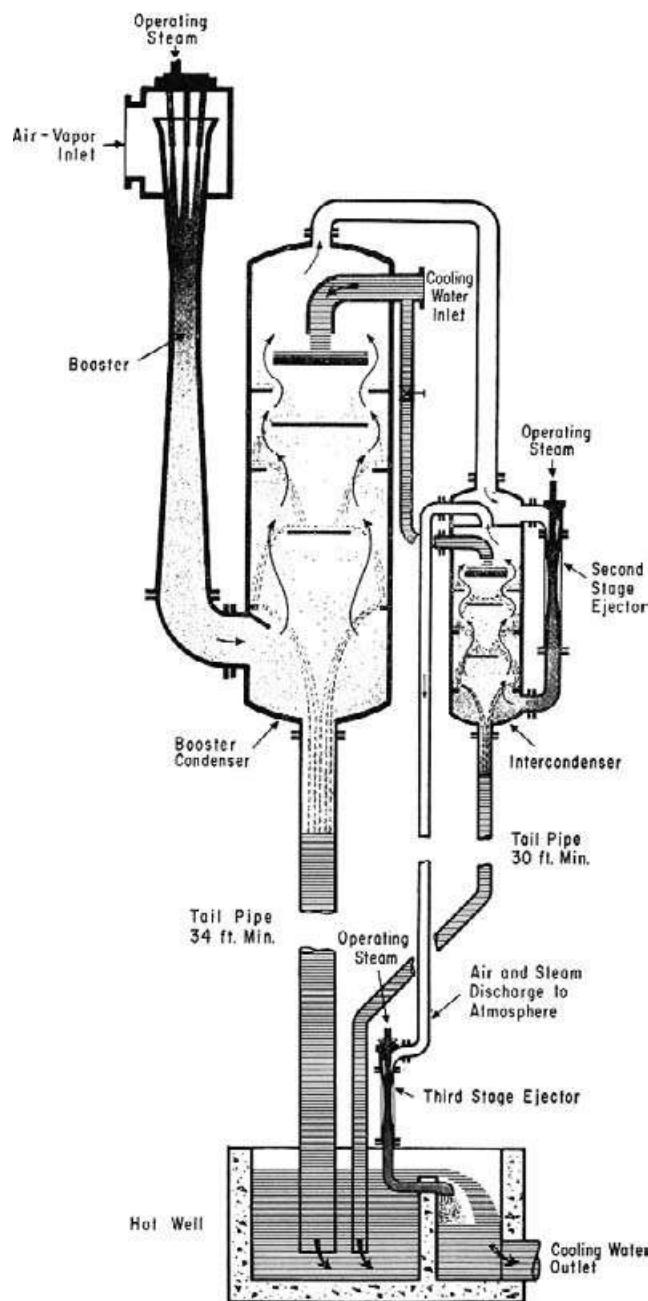
Figure 8-6 Two-stage ejector employing surface type inter- and aftercondenser. (By permission from Elliott Co.)

## 8.6 MATERIALS OF CONSTRUCTION

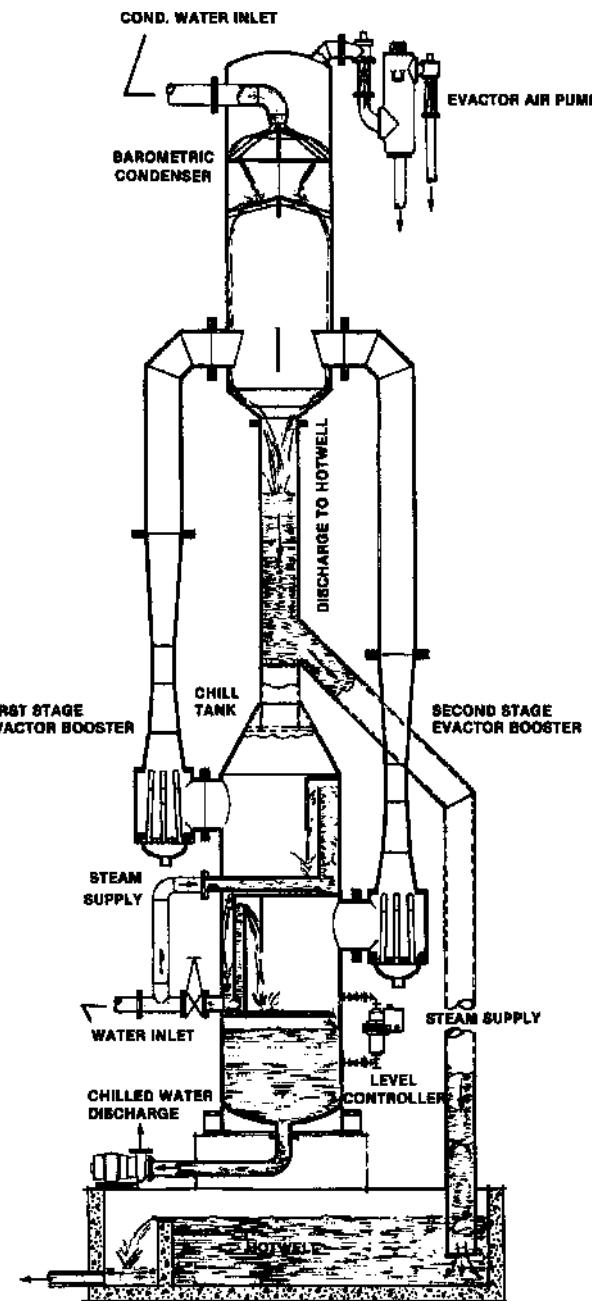
Because the ejector is basically simple in construction, it is available in many materials suitable for handling corrosive vapors. Standard materials include cast iron, Meehanite, cast steel, stainless steel, Monel®, Hastelloy®, titanium, Teflon®, Haveg®, rubber-lined steel, graphite-lined, polyvinyl chloride (PVC), fiberglass reinforced plastic (FRP), and bronze for the body and diffuser depending on the pressure and temperature rating. The nozzle is usually stainless steel or monel. Other materials of construction include porcelain, carbon, graphite, impregnated graphite, synthetic resins, glass, and special metals of all types. Intercondensers and aftercondensers are sometimes made of these same materials and may include random-packed surface-type, graphite or glass tubes, and so on.

## 8.7 VACUUM RANGE GUIDE

It is necessary to consult manufacturers for final and specific selections. However, the following guide data is reliable and should serve to check recommendations or to specify a system. It is advisable to try to accomplish the specific operation with as few ejectors



**Figure 8-7a** Flow diagram of three-stage ejector with counter-current barometric booster condenser and intercondenser. (By permission from Ingersoll-Rand Co.)



**Figure 8-7b** Chilled water refrigeration unit using steam jet ejectors. (By permission from Croll-Reynolds Co., Inc.)

as possible, because this leads to the most economical operation and lowest first cost in the majority of cases. Figures 8-9a–e are a basic comparison guide for vacuum systems. Figure 8-9f represents different views of a single-stage ejector.

The ranges shown for various numbers of ejector stages provide a reasonable operational guide with understandable variations between various manufacturers, even in combinations of each manufacturer's specific ejectors used to attain lower or upper range of the chart. For example, at zero load or shutoff pressure, Reference [9] indicates the approximate values for evacuation pressures (lowest):

Stage No. in System	Lowest Absolute Pressure (mmHg abs)
Single	50
Two	4–10
Three	0.8–1.5
Four	0.1–0.2
Five	0.01–0.02
Six	0.001–0.003

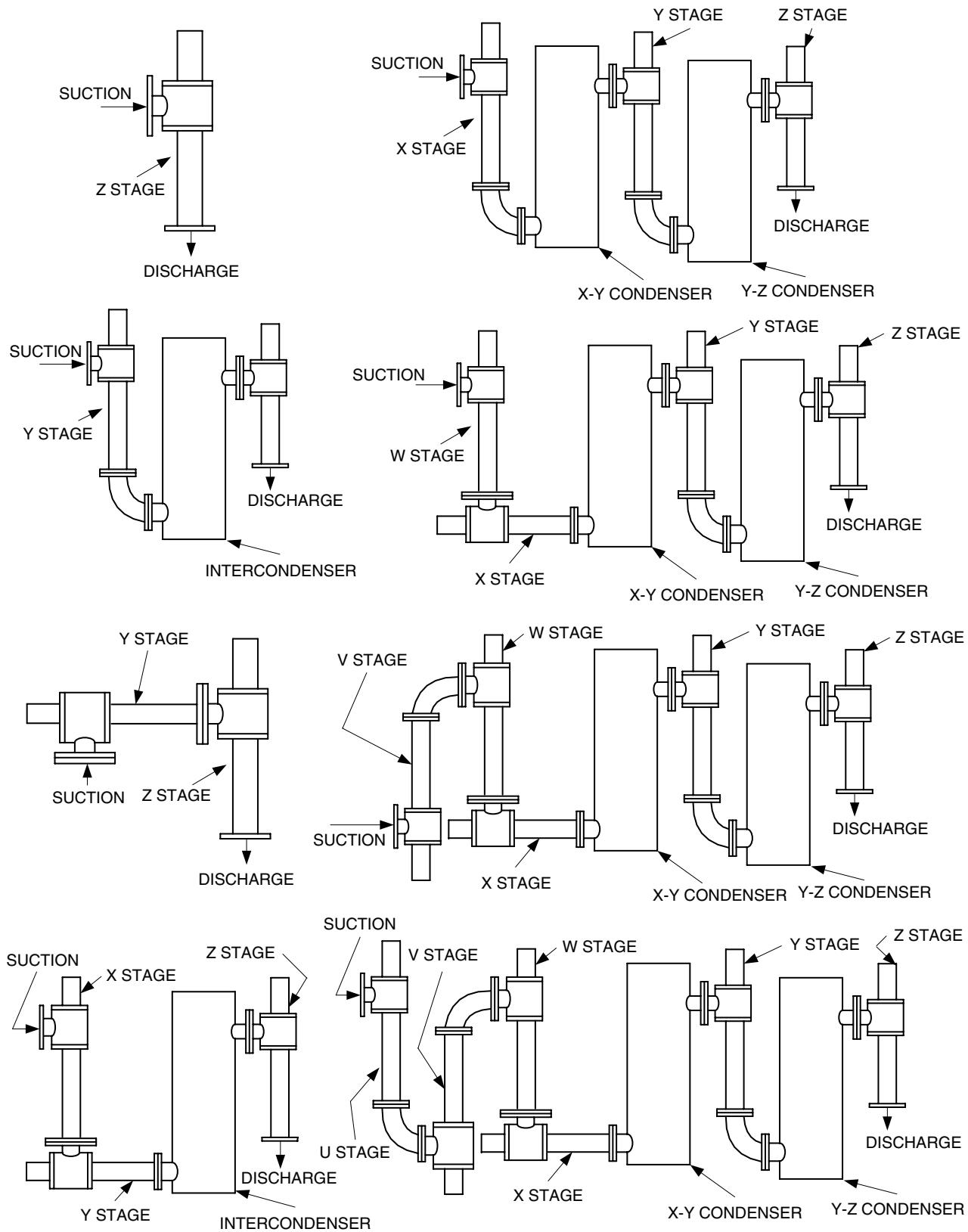


Figure 8-8 Steam jet arrangements with inter- and aftercondensers. (By permission from Croll-Reynolds Co., Inc.)

**TABLE 8-2 Standard Ejector Units Designations Conforming to Heat Exchange Institute**

Letter No.	Position in Series	Normal Range of Suction Pressures (Hg abs)	Normal Range of Disch. Pressures (Hg Abs)
Z	Atmospheric stages	3-12"	30-32"
Y	1st of two stages	0.5-4"	4-10"
X	1st of three stages	0.1-1"	1-3"
W	1st of four stages	0.2-4 mm	2-20 mm
V	1st of five stages	0.02-0.4 mm	0.4-3 mm
U	1st of six stages	0.01-0.08 mm	0.08-0.4 mm

The different types of condensing equipment used with the various series are identified by the following letters:

B – Barometric Counter-Flow Condenser, Intercondenser, and Aftercondenser

S – Surface-Type Condenser, Intercondenser, and Aftercondenser

J – Atmospheric Jet Condenser, Intercondenser, and Aftercondenser

C – Surface Coil-Type Condenser, Intercondenser, and Aftercondenser

N – Signifies no condenser in the series

The operating range of the condensing equipment determines the nomenclature. Here are the basic divisions.

Condenser 1.5-4" Hg abs

Intercondenser 4-10" Hg abs

Aftercondenser 30-32" Hg abs

By permission from Croll-Reynolds Co., Inc.

Figure 8-10 is a summary of operating pressure ranges for a variety of processes and vapor mixtures.

Tables 8-1 and 8-3 give the usual industrial application ranges for ejector stages.

Figures 8-11a-c indicate the capacity of various ejector-condenser combinations for variable suction pressures when using the same quantity of 100 psig motive steam. Each point on these curves represents a point of maximum efficiency, and thus any one curve may represent the performance of many different size ejectors each operating at maximum efficiency [11]. Good efficiency may be expected from 50 to 115% of a design capacity. Note that the performance range for the same type of ejector may vary widely depending upon design conditions.

One kilogram per square = 14.22 pound per square inch abs centimeter

One kilogram per square = 735.6 millimeter centimeter

#### EXAMPLE 8-1

##### Conversion of Inches Vacuum to Absolute

A distillation column is operating at 27.5 in. Hg vacuum, referenced to a 30-in. barometer. This is the pressure at the inlet to the ejector. Due to pressure drop through a vapor condenser and trays of a distillation column, the column's bottom pressure is 23 in. vacuum. Determine the absolute pressures at the entrance to the ejector and at the bottom of the column.

##### Solution

Ejector inlet, absolute pressure =  $30 - 27.5 = 2.5$  in. Hg  
Column's bottom, absolute pressure =  $30 - 23 = 7$  in. Hg

## 8.8 PRESSURE TERMINOLOGY

For design purposes it is necessary to use absolute pressures. In plant operation, pressures are often used as "vacuum." It is important to eliminate confusion before making a proper performance analysis (see Tables 8-4 and 8-5).

If pressure is expressed as inches of mercury vacuum, the reading of the local barometer (or a reference barometer) is necessary to establish the absolute suction pressure, or pressure in the vacuum system.

#### Absolute measurement relations (reference to mercury)

One inch mercury	= 25.4 millimeters mercury = 0.491 pound per square inch abs
One inch	= 25,400 microns (micrometers)
One millimeter	= 0.03,937 inch
One millimeter	= 1000 microns
One micron	= 0.001 millimeter
One micron	= 0.00,003,937 inch
One kilogram per square centimeter	= 28.96 inches

## 8.9 PRESSURE DROP AT LOW ABSOLUTE PRESSURES

Refer to Chapter 4.

## 8.10 PERFORMANCE FACTORS

### STEAM PRESSURE

The motive steam design pressure must be selected as the lowest expected pressure at the ejector steam nozzle. The unit will not operate stably on steam pressures below the design pressure [8].

Recommended steam design pressure = minimum expected line pressure at ejector nozzle minus 10 psi.

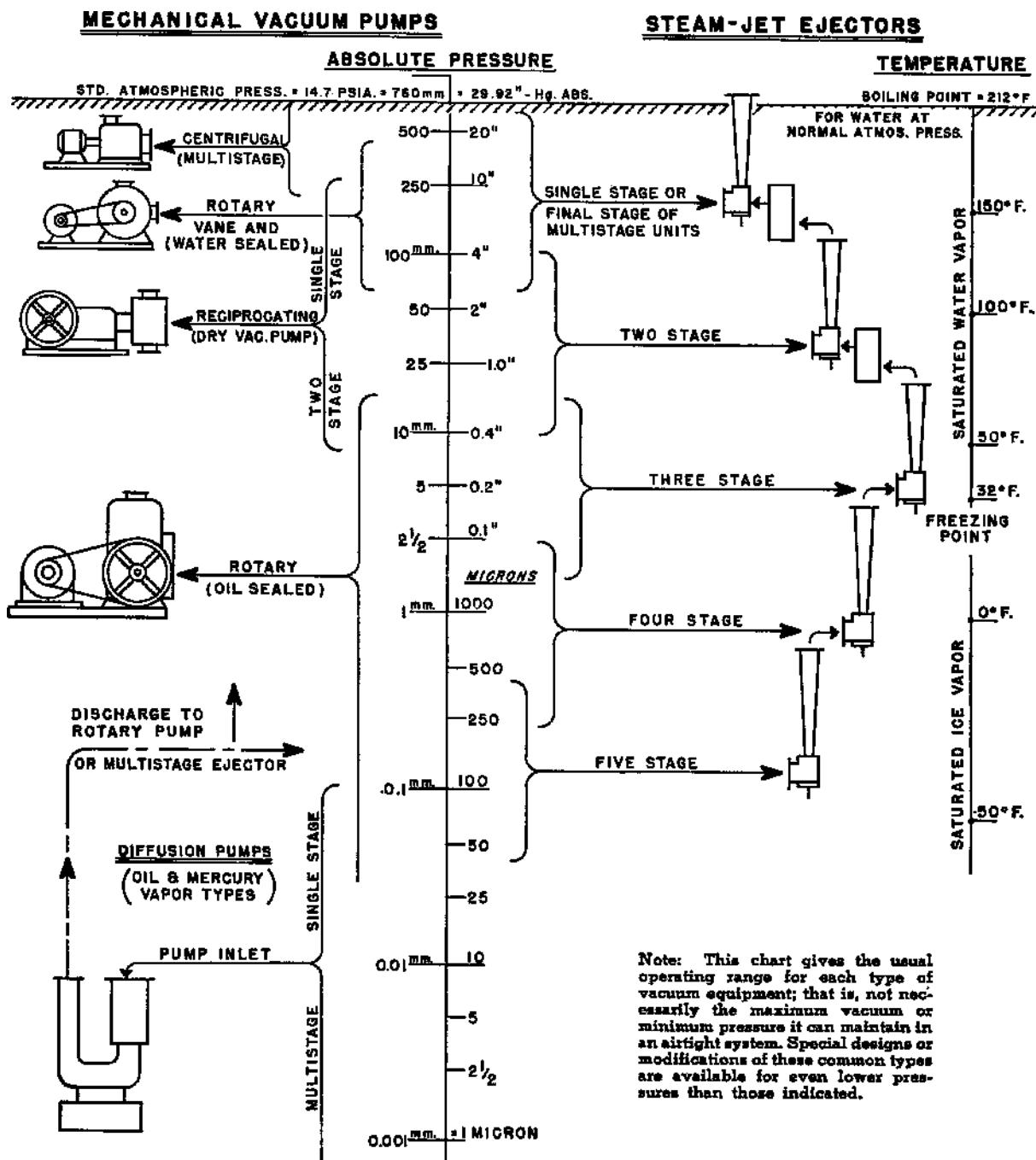


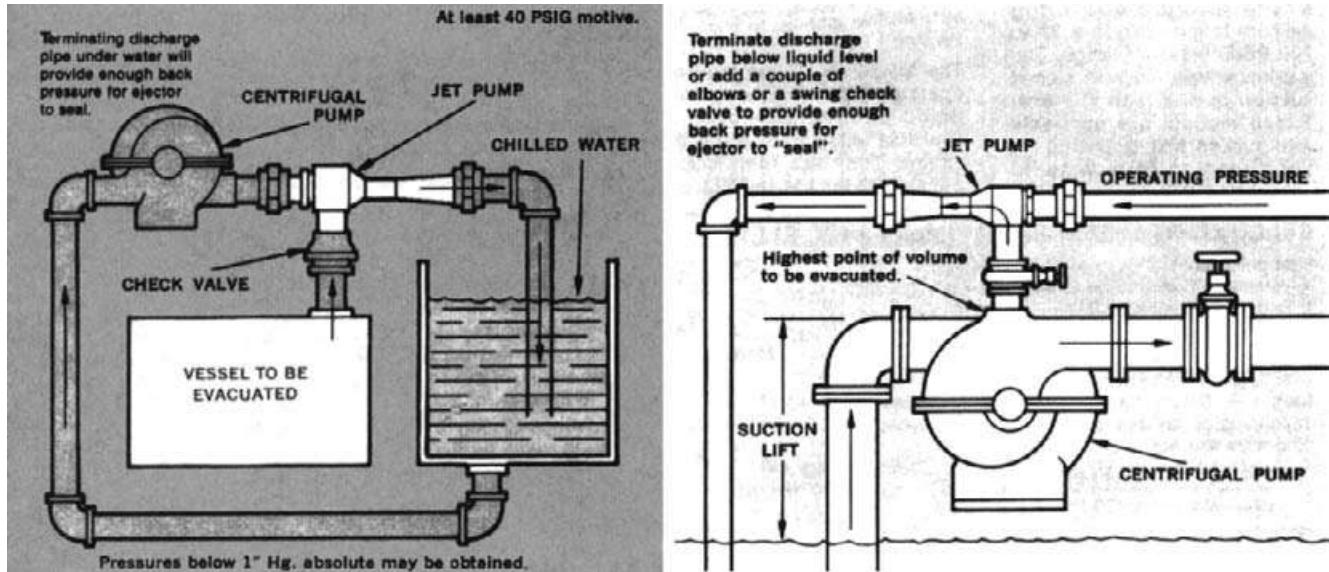
Figure 8-9a Where vacuum equipment applies. (By permission from Ingersoll Rand Co.)

This design basis allows for stable operation under minor pressure fluctuations.

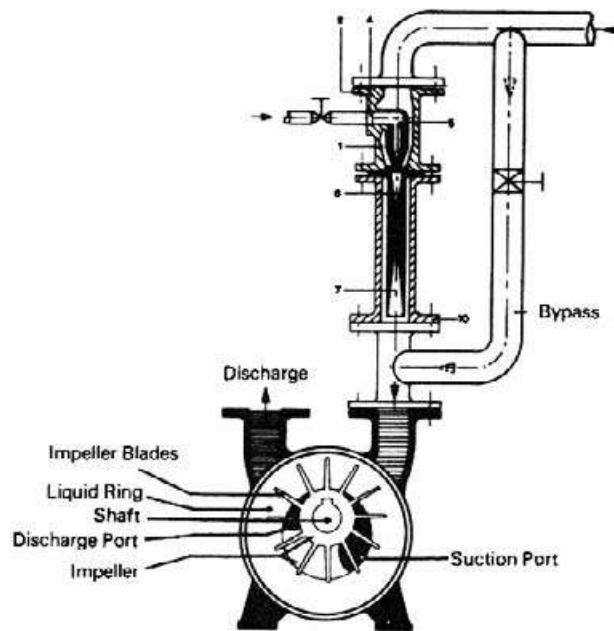
An increase in steam pressure over design will not increase vapor handling capacity for the usual "fixed capacity" ejector. The increased pressure usually decreases capacity due to the extra steam in the diffuser. The best ejector steam economy is attained when the steam nozzle and diffuser are proportioned for a specified performance [14]. This is the reason it is difficult to keep so-called standard ejectors in stock and expects to have the equivalent of a custom-designed unit. The "throttling type" ejector has a family

of performance curves depending upon the motive steam pressure. This type has a lower compression ratio across the ejector than the fixed-type. The fixed-type unit is of the most concern in this presentation.

For a given ejector, an increase in steam pressure over the design value will increase the steam flow through the nozzle in direct proportion to the increase in absolute steam pressure [8]. The higher the actual design pressure of an ejector, the lower the steam consumption. This is more pronounced in one- and two-stage ejectors. When this pressure is above about 350 psig, the



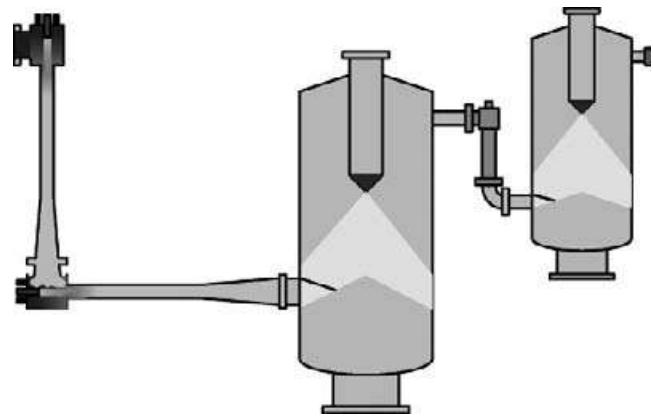
**Figure 8-9b** Typical jet (gas) system application. (By permission from Penberthy, Inc.)



**Figure 8-9c** Improving the maximum vacuum obtainable by staging an ejector ahead of the suction of a liquid ring pump any other device that can handle water into the unit, unless dry air is used. (By permission from Graham Manufacturing Co., Inc.)

decrease in steam requirements will be negligible. As the absolute suction pressure decreases, the advantages of high pressure steam become less. In very small units the physical size of the steam nozzle may place a lower ceiling on steam pressures. Figure 8-12 illustrates the effect of excess steam pressures on ejector capacity for single- and two-stage units.

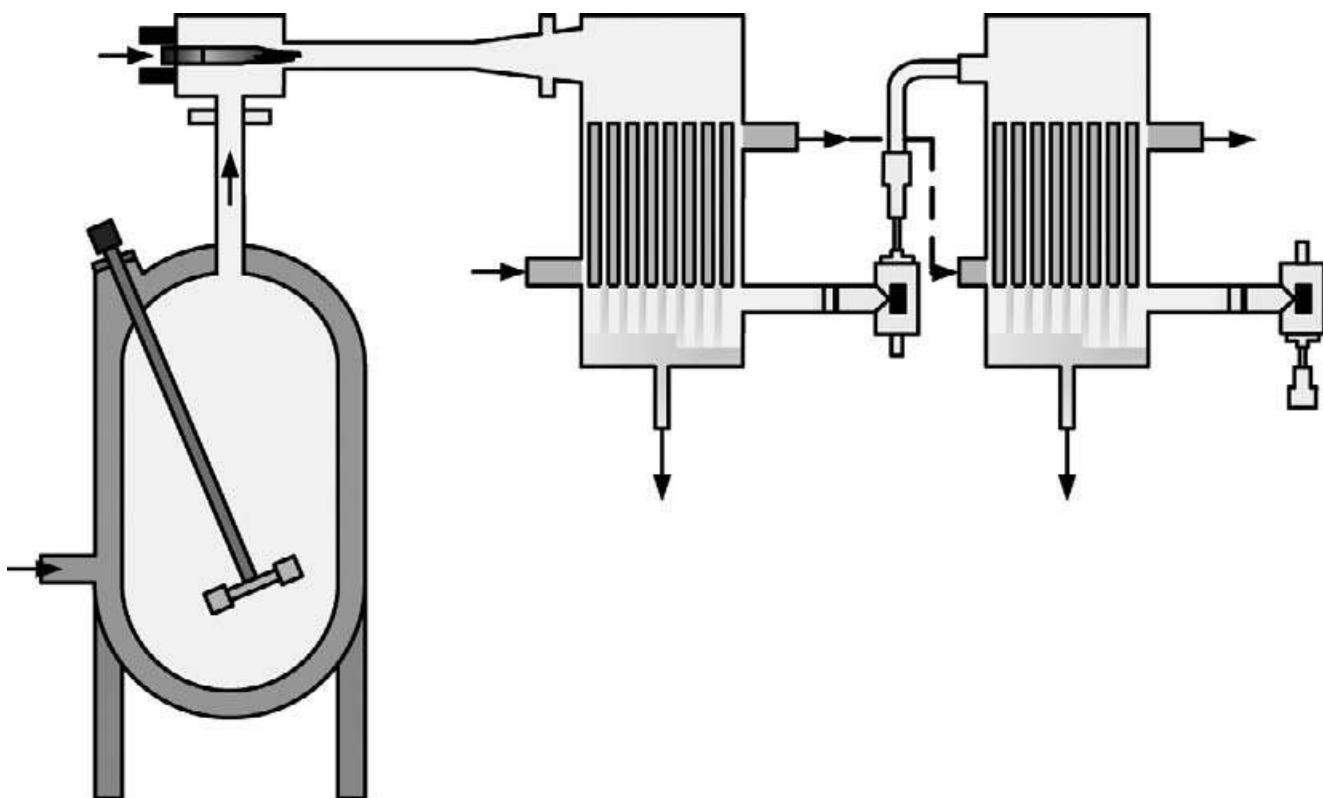
For ejectors discharging to the atmosphere, steam pressures below 60 psig at the ejector are generally uneconomical [8]. If the discharge pressure is lower as in multistage units, the steam pressure at the inlet can be lower. Single-stage ejectors designed for



**Figure 8-9d** Schematic of a three-stage system with direct contact or barometric condensers. (Source: FoxValve.)

pressures below 200 mmHg abs cannot operate efficiently on steam pressures below 25 psig [11]. The first or second stage of a multi-stage system can be designed (although perhaps not economically) to use steam pressures below 1 psig.

To ensure stable operations the steam pressure must be above a minimum value. This minimum is called the motive steam pickup pressure [11] when the pressure is being increased from the unstable region. Figure 8-13 indicates both this point and the second lower break pressure which is reached as the pressure is lowered from a stable region. As the pressure is reduced along line 5-3-1, the operation is stable until point 1 is reached. At this point the ejector capacity falls off rapidly along line 1-2. As the steam pressure is increased, stable operation is not resumed until point 4 is reached and the capacity rises along line 4-3. With further increases, it rises along 3-5 which is the stable region. Operation in the region 3-1 is unstable and the least drop in pressure can cause the system to lose vacuum. The relative location of points 3 and 1 can be controlled to some extent by ejector design; and the points may not even exist for ejectors with low ratios of compression.



**Figure 8-9e** Schematic of an agitated vessel being cooled by a three-stage ejector system. (Source: Fox Valve.)



**Figure 8-9f** Views of a single-stage ejector. (Source: Fox Valve.)

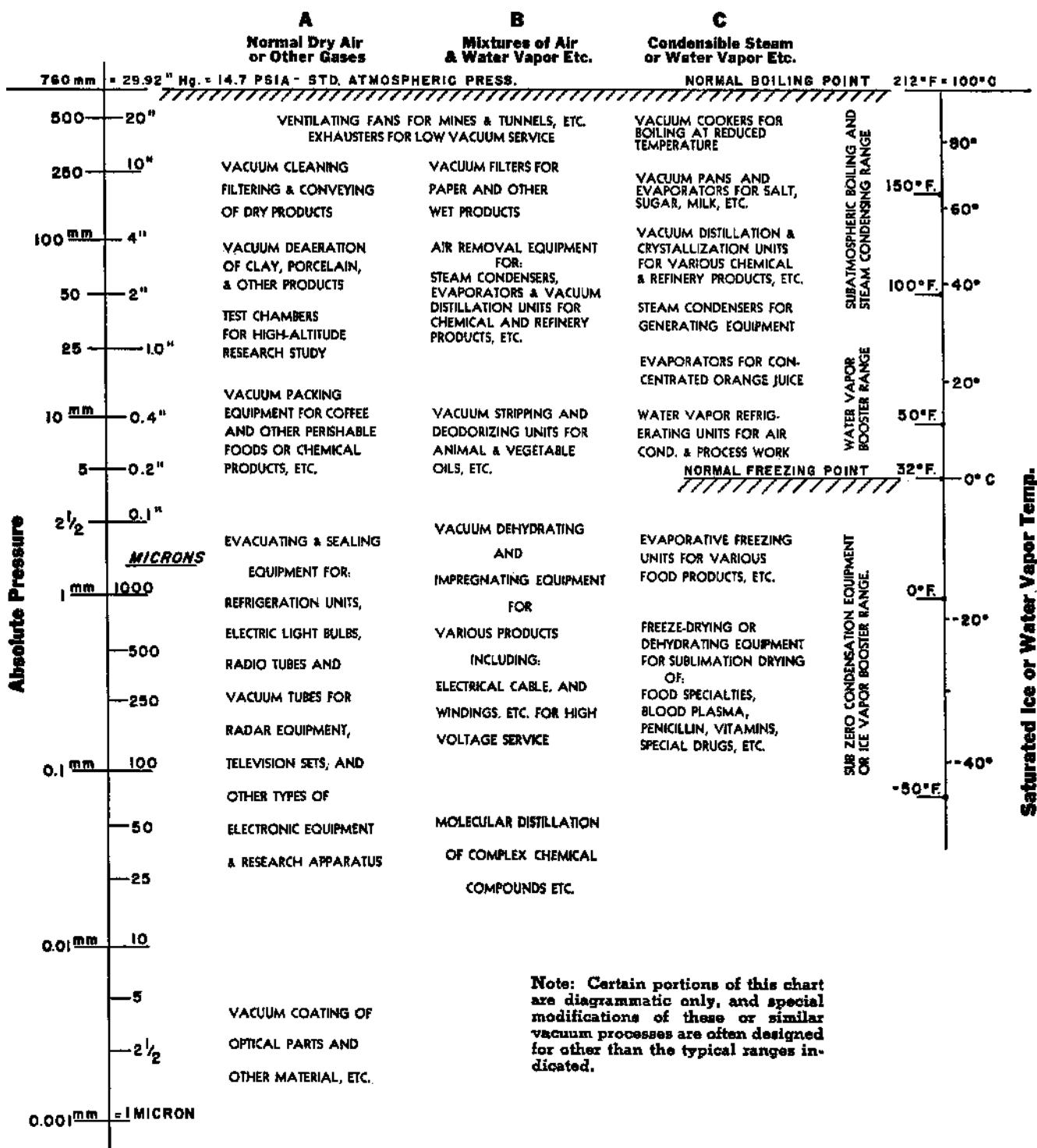
Figure 8-14 indicates the change in region of stable performance as reflected in changes in the backpressure on the ejector and the variation in steam pressure. This system backpressure might represent a variation in barometric pressure for a unit discharging to the atmosphere, or the variation in a feedwater (or other) heater operating pressure if the ejector discharges into closed system or condenser. Figure 8-14 numerically represents the latter situation, although the principle is the same.

The three motive steam pressure curves, 100–90–80%, are obtained from the ejector manufacturer as is the performance curve of suction pressure versus percent of ejector design capacity. This latter curve for an actual installation would show actual absolute suction pressures versus pounds per hour or cubic feet per minute of air or percent design capacity.

The backpressure is represented by the straight lines labeled minimum, normal, and maximum. Only one capacity curve is shown since the increase in capacity resulting from the lower steam pressure is negligible [15].

Curves 1, 2, and 3 represent the maximum safe discharge pressure, as the system will operate along the capacity curve as long as the system discharge pressure from the ejector is less than the maximum value of the curve, all for a given suction pressure [15]. The slopes of the curves are a function of the type of ejector, its physical design, and relative pressure conditions. Whenever the discharge backpressure exceeds the maximum safe discharge pressure as represented by one of the curves, the ejector operates in the “break” unstable region.

In Figure 8-14, the 100% pressure curve does not cross any of the system backpressure lines (minimum, normal, or maximum) and the ejector would be expected to operate stably over its entire range, down to shutoff. Following the 90% steam pressure curve, the ejector is stable at 100% design suction pressure and 100%

**OPERATING RANGES OF VACUUM PROCESSES**

Most of the principal industrial processes that necessitate the use of subatmospheric pressures are listed here, together with the typical pressure range for each one. The processes involve the evacuation or removal of various

combinations of air, other gases, and water vapor, as shown under the three headings A, B, and C. The charts on this page and the preceding one were compiled by J. F. Plummer, Jr., of Ingersoll-Rand Company, Phillipsburg, N. J.

Figure 8-10 Operating range of vacuum processes. (By permission from Ingersoll-Rand Co.)

**TABLE 8-3 General Pressure Ranges for Ejectors**

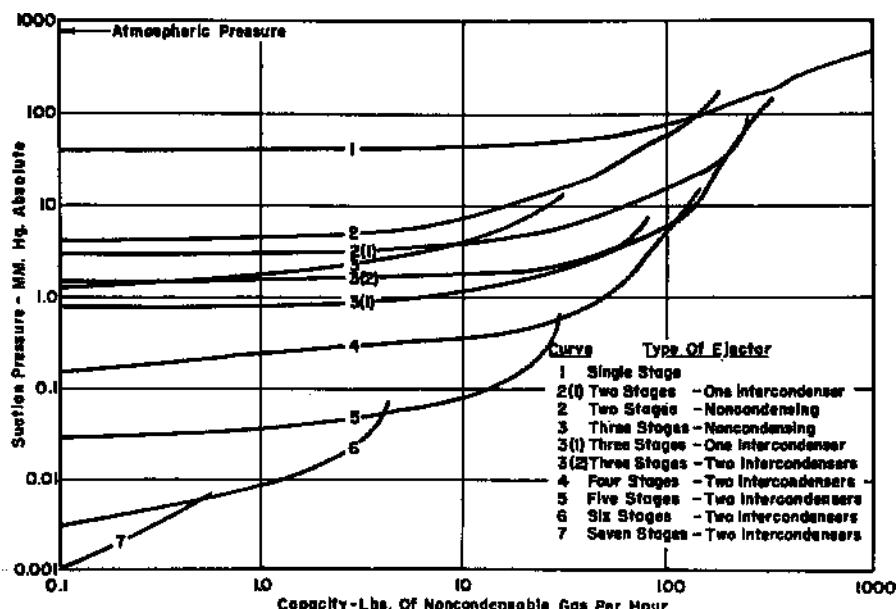
No. of Stages	Minimum Practical Absolute Pressure (mmHg)*	Range Operating Suction Pressure (mmHg) <sup>†,‡</sup>	Closed Test Pressure (mmHg) <sup>‡</sup>
1	50	75, (3") and up	37-50
2	5	10-100	5
3	2	1-25	1
4	0.2	0.25-3	0.05-0.1
5	0.03	0.03-0.3	0.005-0.01
6	0.003		
7	0.001-0.0005 <sup>§</sup>		

\* Linck, C.G., Selecting Ejectors for High Vacuum, *Chem. Eng.* Jan 13, p. 145 (1958)  
Ref. [10].

† Worthington Corp. Bul. W-205-E21 (1955), Ref. [12]

‡ The Jet-Vac Corp., Bulletin, Ref. [13]

§ Berkeley, F.D., Ejectors Have a Wide Range of Users, Pet. Ref. 37, No. 12, p. 95 (1958), Ref. [11]



**Figure 8-11a** Comparison guide for steam ejector performance. As absolute pressure is reduced, the number of stages increases for a given capacity. The same steam consumption is used for each design. (By permission from Berkley, F.D. [11].)

design capacity at the maximum back pressure. It is unstable below design load unless the heater pressure is reduced. Note that its break occurs at 20 psia and 100% design suction pressure. If the discharge-pressure is reduced to 19 psia, the unit will be stable to shutoff (zero capacity). The 80% steam pressure will allow stable operation from shutoff up through the full capacity range as long as the backpressure does not exceed 18 psia. This type of analysis is necessary to properly evaluate ejector performance with varying system conditions.

A unit is said to have 50% overload capacity when it blanks off (zero load) at a stable absolute pressure and has an operating curve which stably handles 1.5 times the design conditions of flow.

#### EFFECT OF WET STEAM

Wet steam erodes the ejector nozzle and interferes with performance by clogging the nozzle with water droplets [8]. The effect

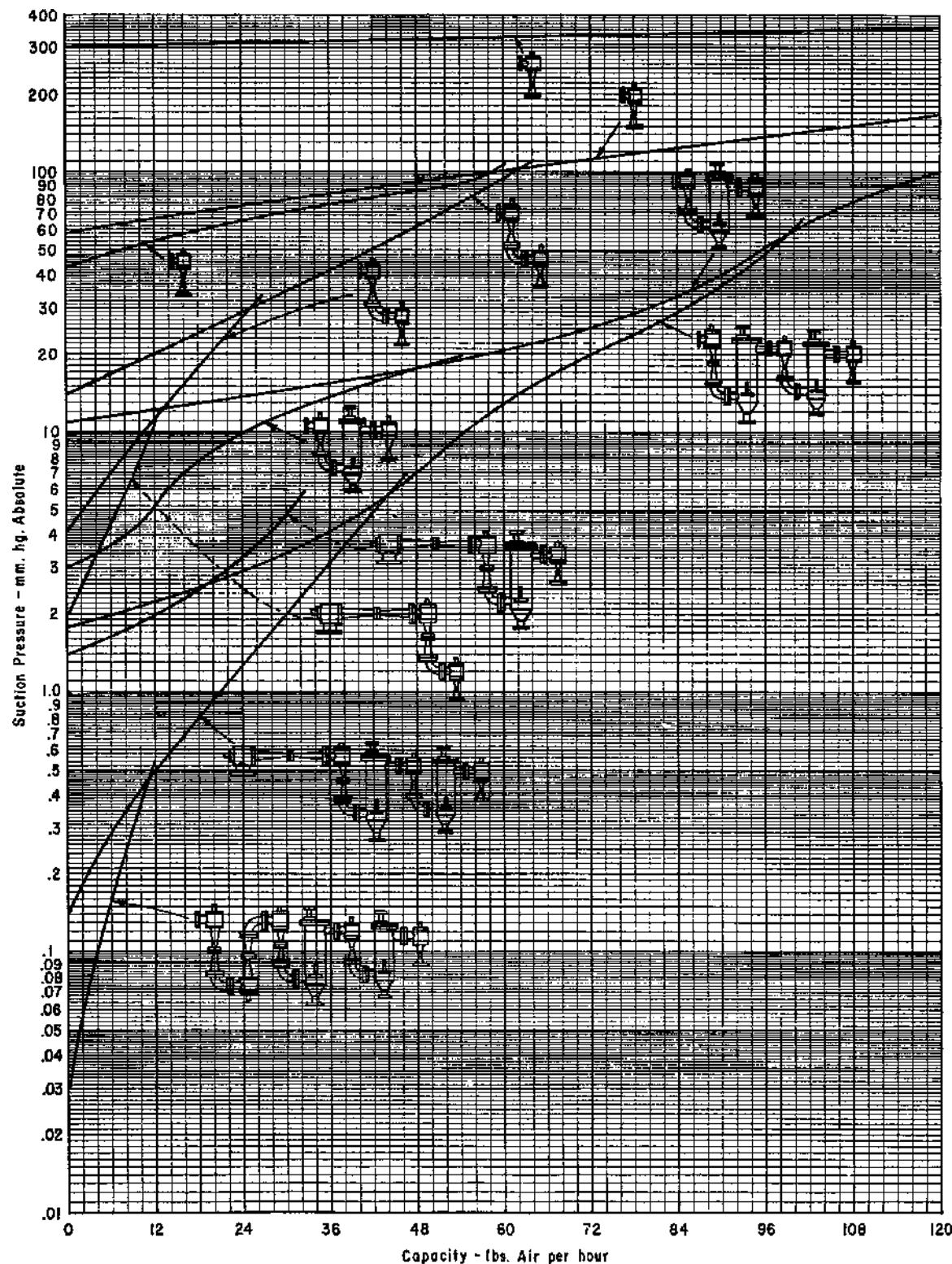
on performance is significant and is usually reflected in fluctuating vacuum.

#### EFFECT OF SUPERHEATED STEAM

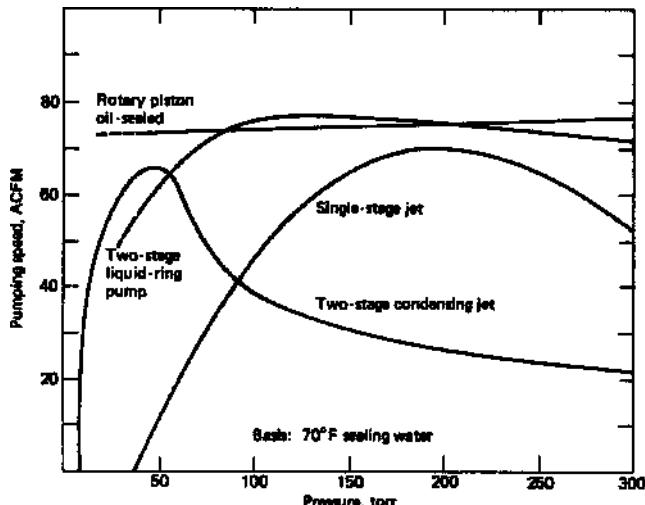
A few degrees of superheat are recommended (5–15°F), but if superheated steam is to be used, its effect must be considered in the ejector design. A high degree of superheat is of no advantage because the increase in available energy is offset by the decrease in steam density [8].

#### SUCTION PRESSURE

The suction pressure of an ejector is expressed in absolute units. If it is given as inches of vacuum it must be converted to absolute units by using the local or reference barometer. The suction pressure follows the ejector capacity curve, varying with the non-condensable and vapor load to the unit.



**Figure 8-11b** A typical relative comparison of various designs of steam ejectors based on same steam consumption (100 psig steam pressure and 85° F water). Curves represent the capacity of ejectors designed for maximum air-handling capacity at any one particular suction pressure. (By permission from Graham Manufacturing Co.)



**Figure 8-11c** Typical performance curves for steam jet ejectors, liquid ring pumps, and rotary piston oil-sealed pumps. (By permission from Ryan, J.L. and Roper, D.L. [1].)

#### DISCHARGE PRESSURE

As indicated, performance of an ejector is a function of backpressure. Most manufacturers design atmospheric discharge ejectors for a pressure of 0.5–1.0 psig in order to insure proper performance. The pressure drop through any discharge piping and aftercooler

**TABLE 8-4 Low Absolute Pressure Equivalents (References to Mercury)**

Microns	Millimeters	Inches	Inches Vac. Referred to 30" Barometer
10	0.01	0.000394	29.999606
100	0.10	0.003937	29.996063
200	0.20	0.007874	29.992126
300	0.30	0.011811	29.988189
400	0.40	0.015748	29.984252
500	0.50	0.019685	29.980315
600	0.60	0.023622	29.976378
700	0.70	0.027559	29.972441
800	0.80	0.031496	29.968504
900	0.90	0.035433	29.934567
1000	1.00	0.039370	29.960630

must be taken into consideration. Discharge piping should not have pockets for condensation collection.

Figure 8-15 indicates the effect of increasing the single-stage ejector backpressure for various suction pressures. Figure 8-16 illustrates the effect of increasing the motive steam pressure to overcome backpressure effects. When this pressure cannot be increased, the nozzle may be redesigned to operate at the higher backpressure.

#### CAPACITY

The capacity of an ejector is expressed as pounds per hour total of non-condensables plus condensables to the inlet flange of the unit. For multistage units, the total capacity must be separated into

**TABLE 8-5 Absolute Pressure Conversion Table (Millimeters to Inches Mercury)**

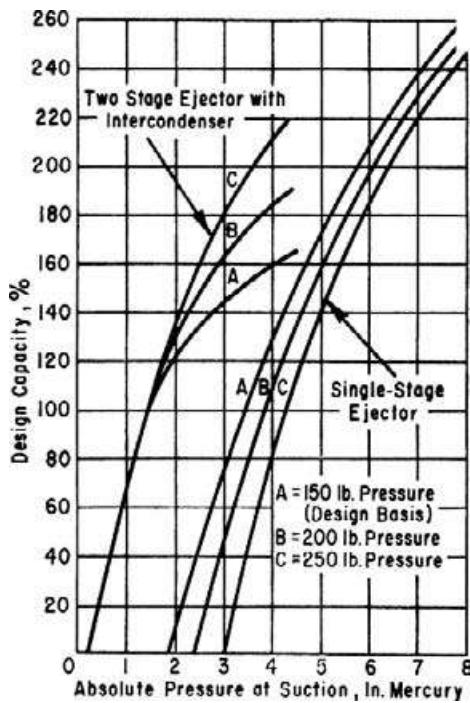
Millimeters	Inches	Millimeters	Inches	Millimeters	Inches
1	0.0394	26	1.0236	170	6.6929
2	0.0787	27	1.0630	180	7.0866
3	0.1181	28	1.1024	190	7.4803
4	0.1575	29	1.1417	200	7.8740
5	0.1969	30	1.1811	210	8.2677
6	0.2362	35	1.3780	220	8.6614
7	0.2756	40	1.5748	230	9.0551
8	0.3150	45	1.7717	240	9.4488
9	0.3543	50	1.9685	250	9.8425
10	0.3937	55	2.1653	260	10.236
11	0.4331	60	2.3622	270	10.630
12	0.4724	65	2.5590	280	11.024
13	0.5118	70	2.7559	290	11.417
14	0.5512	75	2.9528	300	11.811
15	0.5906	80	3.1496	325	12.795
16	0.6299	85	3.3465	350	13.780
17	0.6693	90	3.5433	375	14.764
18	0.7087	95	3.7402	400	15.748
19	0.7480	100	3.9370	450	17.717
20	0.7874	110	4.3307	500	19.685
21	0.8268	120	4.7244	550	21.653
22	0.8661	130	5.1181	600	23.622
23	0.9055	140	5.5118	650	25.590
24	0.9449	150	5.9055	700	27.559
25	0.9843	160	6.2992	750	29.528

Courtesy of C.H. Wheeler Mfg. Co., Philadelphia, PA

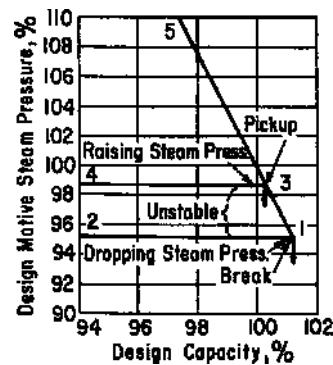
Note: To change the above values to pressure in psia, multiply by the following factors:

Multiply millimeters of mercury by 0.01934

Multiply inches of mercury by 0.4912



**Figure 8-12** Effects of excess steam pressure on ejector capacity. (By permission from C.H. Wheeler Mfg. Co.)



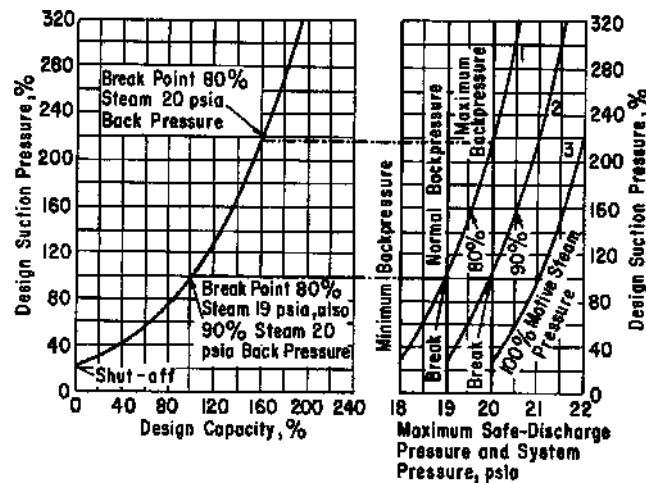
**Figure 8-13** Effect of steam pressure on capacity for constant system suction and back pressure. (By permission from Freneau, P. [15].)

pounds per hour of condensables and non-condensables. The final stages are only required to handle the non-condensable portion of the load plus the saturation moisture leaving the inter-condensers.

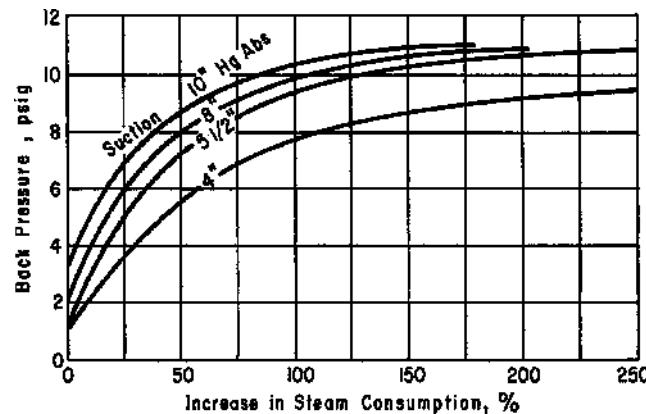
The non-condensables leaving a surface condenser are saturated with water vapor at the temperature corresponding to the pressure. For a process condenser the vapor corresponds to the process fluid.

#### 1. Surface Condenser

To provide for sufficient total capacity, the temperature at the air outlet of a well-designed surface condenser is generally assumed to be about  $7.5^{\circ}\text{F}$  below the temperature of saturated steam at the absolute pressure in the condenser [16].



**Figure 8-14** Effect of steam back-pressure variations with steam pressure. (By permission from Freneau, P. [15].)



**Figure 8-15** Effect of back pressure on single-stage exhaustors. (By permission from Ketterma, Schutte & Koerting Div. and Ketterer, S.G. and Blatchley, C.G. [14].)

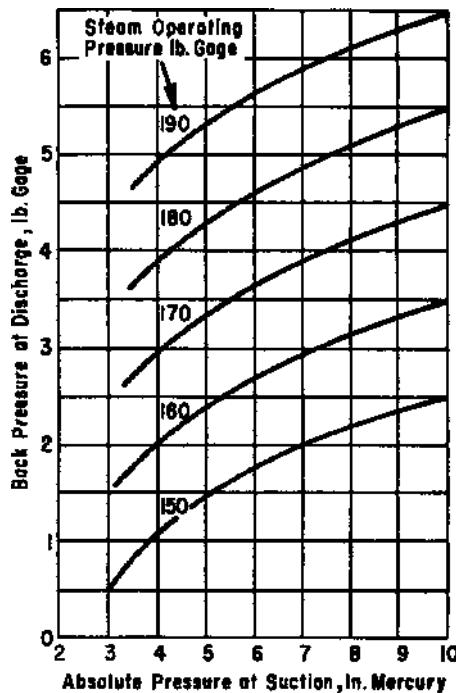
#### 2. Barometric or Low Level Jet Condenser

In this case, the temperature at the air outlet of this condenser type is generally assumed to be  $5^{\circ}\text{F}$  above the inlet temperature of the cooling water. In addition to the normally expected air leakage, an allowance must be made for air liberated from the injection water [17].

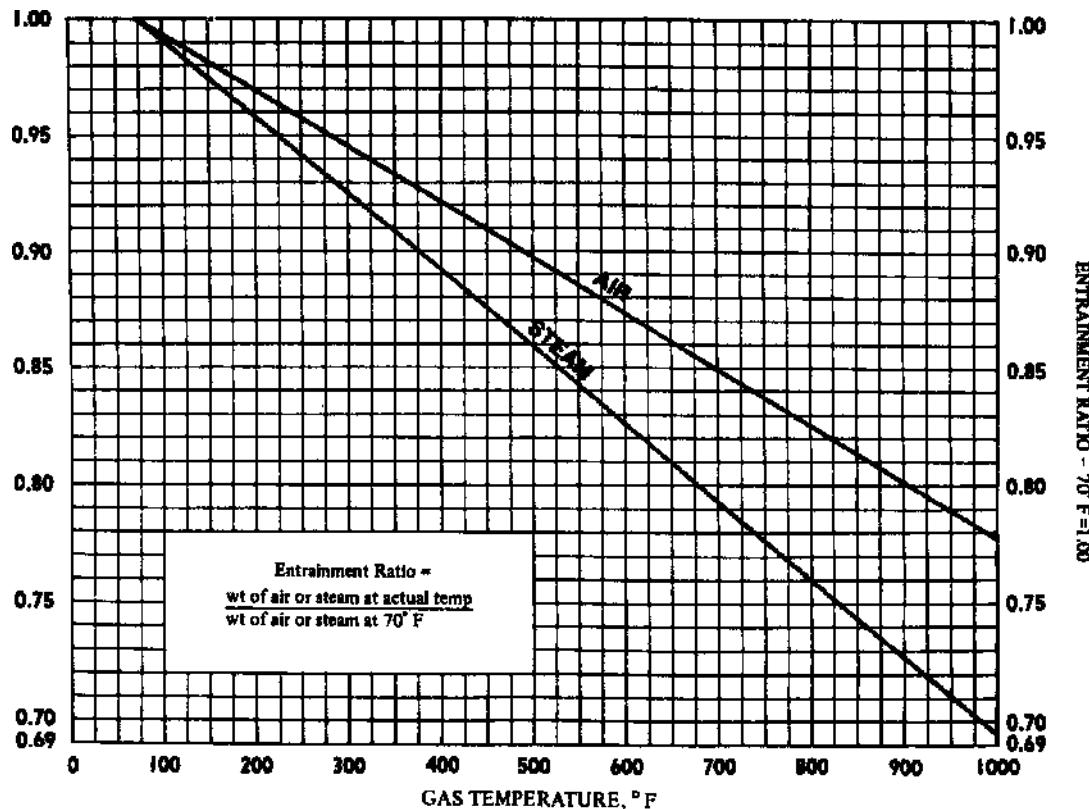
### 8.11 TYPES OF LOADS

#### AIR PLUS WATER VAPOR MIXTURES

The Heat Exchanger Institute [7] references all tests and calculations for jet performance to  $70^{\circ}\text{F}$  air equivalent. Figures 8-17 and 8-18 are used to handle the evaluation on an equivalent air basis. If actual performance curves are available for the temperature and vapor mixture in question, conversion to an equivalent basis is not necessary except for test purposes.



**Figure 8-16** Effect of high back-pressure on ejector operation. (By permission from C.H. Wheeler Mfg. Co.)



**Figure 8-17** Temperature entrainment ratio curve. (Reprinted by permission from Standards for Steam Jet Vacuum Systems 4th ed., Heat Exchange Institute, Inc., 1988.)

#### EXAMPLE 8-2

#### 70° F Air Equivalent for Air-Water Vapor Mixture

What is the 70° F air equivalent for 500 lb/h of a mixture containing 150 lb/h of air and 350 lb/h of water vapor if it is at 350° F?

#### Solution

From Figure 8-17, the entrainment ratio for air is 0.93 lb air at 70° F/lb air at 350° F.

$$\text{For air: } 70^\circ \text{ F air equivalent} = 150/0.93 = 161.3 \text{ lb/h}$$

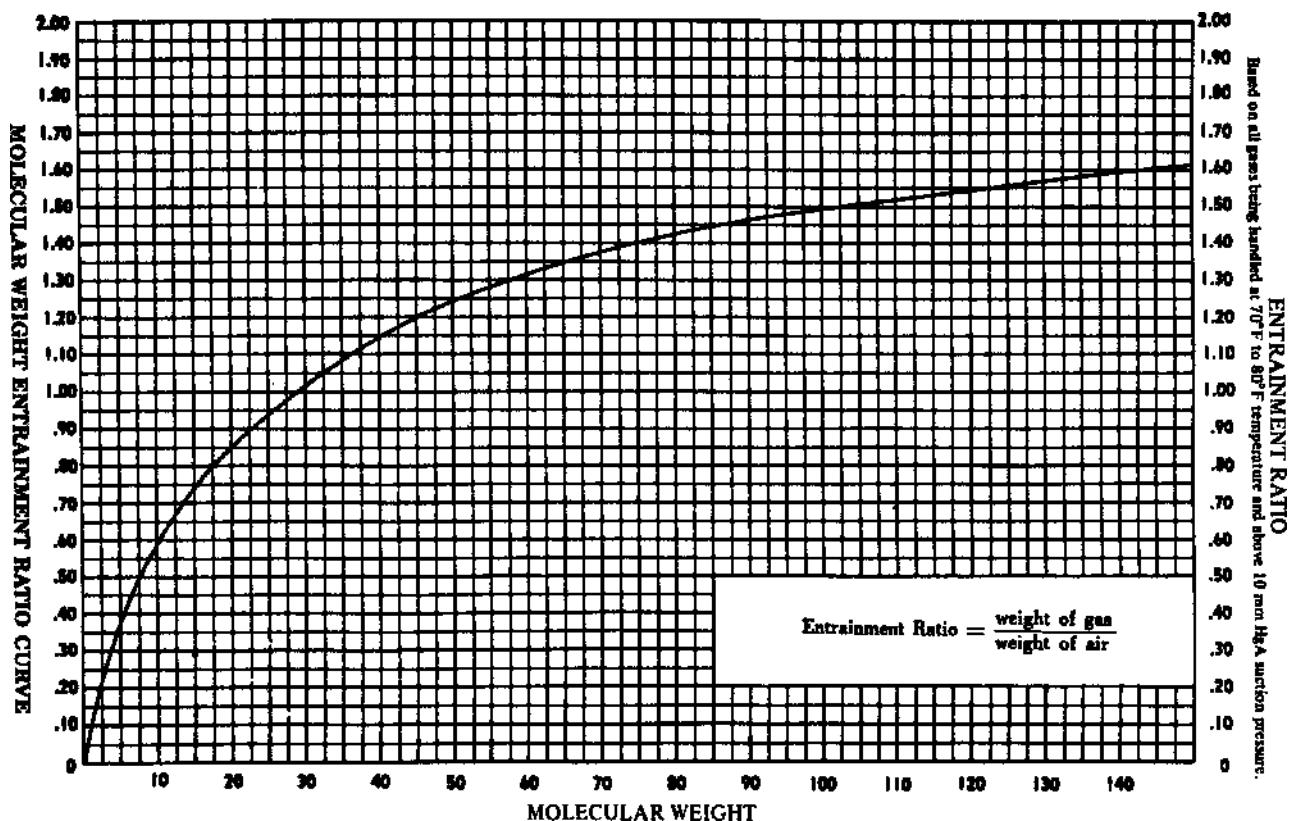
$$\text{For steam: From Figure 8-17, entrainment ratio} = 0.908 \text{ lb steam at } 70^\circ \text{ F/lb steam at } 350^\circ \text{ F}$$

$$70^\circ \text{ F steam equivalent} = 350/0.908 = 385 \text{ lb/h}$$

From Figure 8-18, molecular weight = 18, ratio = 0.81

$$\text{The } 70^\circ \text{ F air equivalent of the steam equivalent} = 385/0.81 = 476 \text{ lb/h}$$

$$\text{Mixture: The mixture } 70^\circ \text{ F air equivalent} = 161.3 + 476 = 637.3 \text{ lb/h}$$



**Figure 8-18** Molecular weight entrainment ratio curve. (Reprinted by permission from Standards for Steam Jet Ejectors, 3rd ed., Heat Exchange Institute, 1956.)

### EXAMPLE 8-3

#### Actual Air Capacity for Air-Water Vapor Mixture

Some manufacturers furnish 70°F air equivalent curves to allow the purchaser to convert performance to actual plant conditions. It is almost impossible to operate on one design point. What is the expected actual capacity at 500°F of a mixture rated at 70°F?

$$\begin{array}{rcl} \text{Air} & = 325 \text{ lb/h} \\ \text{Water vapor} & = 300 \text{ lb/h} \end{array}$$

$$\text{Total} \quad \underline{\hspace{2cm}} \quad 525 \text{ lb/h}$$

For air: From Figure 8-17, entrainment ratio = 0.897

$$\text{actual air capacity}$$

$$= 225(0.897) = 202 \text{ lb/h}$$

For water vapor: From Figure 8-17, entrainment ratio = 0.859

$$\text{actual water vapor capacity}$$

$$= 300(0.859) = 258 \text{ lb/h}$$

Total mixture actual capacity

$$= 202 + 258 = 460 \text{ lb/h}$$

*Note:* If the data for this example had been given as air equivalent, then the water vapor portion would have been corrected for molecular weight, using Figure 8-18.

If there is only one of the components in the system, its corrections are made as illustrated in the preceding examples.

#### STEAM AND AIR MIXTURE TEMPERATURE

For a mixture of steam and air handled by an ejector, the temperature of the mixture in the ejector mixing chamber is calculated by [7]:

$$t_m = \frac{W_s C_{ps} t_s + W_a C_{pa} t_a}{W_s C_{ps} + W_a C_{pa}} \quad (8-1)$$

$t_m$  = temperature of mixture at ejector suction, °F

$W_s$  = steam flow rate, lb/h

$C_{ps}$  = specific heat of steam at constant pressure corresponding to downstream absolute pressure (0.45 approx.)

$t_s$  = temperature of steam on downstream side of nozzle, °F

$W_a$  = Air flowrate, lb/h

$C_{pa}$  = specific heat of air at constant pressure (0.24 approx.)

$t_a$  = ambient air temperature, °F.

Reference [7] provides a complete procedure for testing ejector units in vacuum service, and the charts and calculation procedures for the tests.

Another design approach for calculating saturated gas loads for vacuum systems is given in Reference [18].

**EXAMPLE 8-4****Steam-Air Mixture Temperature Ejector**

Steam used to draw air out of a vessel is:

240 psig (255 psia) at 440°F total temperature  
which is 440 – 402 saturated temperature = 38 superheat

Temperature of air from vessel: 75°F

Steam flow: 475 lb/h

Air flow: 175 lb/h

Ejector suction pressure: 1.5 in. Hg abs

then, Enthalpy of steam = 1226.5 Btu/lb (from superheat vapor tables at 255 psia)

Corresponding steam temperature at 1.5 in. Hg abs and enthalpy of 1226.5 = 366°F (interpolation on superheated steam tables at 1.5 in. Hg abs)

$$t_m = \frac{(475)(0.45)(366) + (175)(0.24)(0.75)}{(475)(0.45) + (175)(0.24)} = 306^{\circ}\text{F}$$

**TOTAL WEIGHT OF A SATURATED MIXTURE OF TWO VAPORS: ONE BEING CONDENSABLE**

Often when the non-condensable quantity is known or estimated, it is important to state whether these gases are in the presence of water or other process liquid. In this case, the amount of condensable vapor above the liquid must be considered as it also will enter the ejector suction.

$$W_v = \frac{W_n M_v P_v}{M_n P_n} \quad (8-2)$$

where

n refers to the non-condensable component

v refers to the condensable vapor.

**NON-CONDENSABLES PLUS PROCESS VAPOR MIXTURE**

Many process systems fall in this group. They are handled in a similar manner to the other systems, correcting for temperature and molecular weight.

1. Calculate the average molecular weight of the mixture.
2. The air equivalent is determined from Figure 8-18 using the average molecular weight.

Air Equivalent

$$= \frac{\text{lb/h of mixture}}{\text{Ratio (from Fig 8-18) (uncorrected for temperature)}}$$

3. The 70°F air equivalent correcting for temperature is found as previously described, using the air curve of Figure 8-17.

The air bleed is used to maintain a constant condition. However, a control valve may be used instead. Control or hand valves in the lower pressure vapor lines to an ejector are not recommended, as they must be paid for in system pressure drop and ejector utility requirements.

**NON-CONDENSABLES PLUS WATER VAPOR MIXTURE**

This is also a frequent process situation. To determine the 70°F air equivalent, the non-condensables are determined as in Example 8-5 and the water vapor as in Example 8-2. The total for the mixture is the sum of these two values.

The total pressure of system at suction to ejector is

$$P = P_n + P'_{v1} + P'_{v2} + \dots \quad (8-3)$$

**AIR-WATER VAPOR MIXTURE PERCENT CURVES**

For saturated air-water vapor systems, Figures 8-20a-d are useful in solving for the pounds of water vapor per pound of air (Dalton's Law, Equation 8-2).

**EXAMPLE 8-5****Actual Capacity for Process Vapor Plus Non-Condensables**

A distillation system is to operate with a horizontal overhead condenser (Figure 8-19) and pressures are as marked. The estimated air leakage into the system is 7 lb/h. The molecular weight of the product vapor going out of the condenser into the ejector (at 80°F) is 53. The vapor pressure of the condensing vapors is 3 mmHg abs at 80°F.

*Solution*

Partial pressure of air = 5 – 3 = 2 mmHg (see Figure 8-19)

Vapor required to saturate at 80°F and 5 mm abs total pressure is given by Eq. (8-2) as

$$W_v = \frac{W_n M_v P_v}{M_n P_n} = \frac{7(53)(3)}{(29)(2)} = 19.2 \text{ lb/h}$$

Average molecular weight of mixture

$$\begin{array}{rcl} \text{Air} & 7 \text{ lb/h} & = 0.241 \text{ mol/h} \\ \text{Process vapor} & 19.2 & = 0.362 \end{array}$$

$$\text{Total vapor} \quad \overline{26.2 \text{ lb/h}} = \overline{0.603 \text{ mol/h}}$$

Mixture avg. mol. wt = 26.2/0.603 = 43.4

Molecular weight correction (from Figure 8-18) = 1.18

$$\text{Air equivalent (at } 80^{\circ}\text{F}) = \frac{26.2}{1.18} = 22.2 \text{ lb/h}$$

Temperature correction (Figure 8-17, using air curve) = 0.999

70°F air equivalent for mixture = 22.2/0.999 = 22.2 lb/h

This is the value to compare with a standard manufacturer's test or performance curve at 70°F.

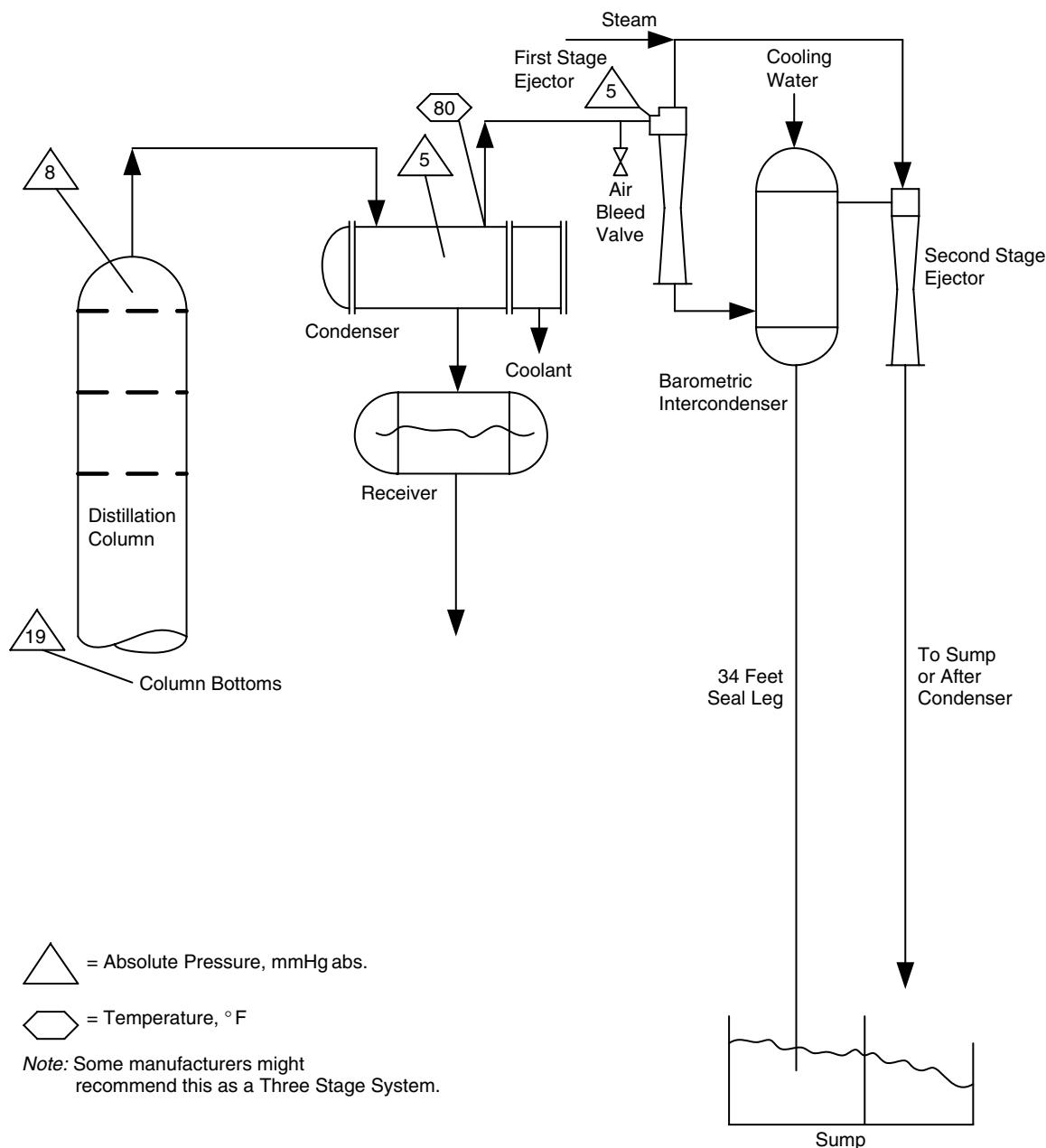


Figure 8-19 Vacuum system for distillation.

**EXAMPLE 8-6****Use of Water Vapor-Air Mixture Curves**

A system handles 50 lb/h of air that is saturated with water vapor at 3 in. Hg abs and 95° F. Find the total amount of water vapor.

*Solution*

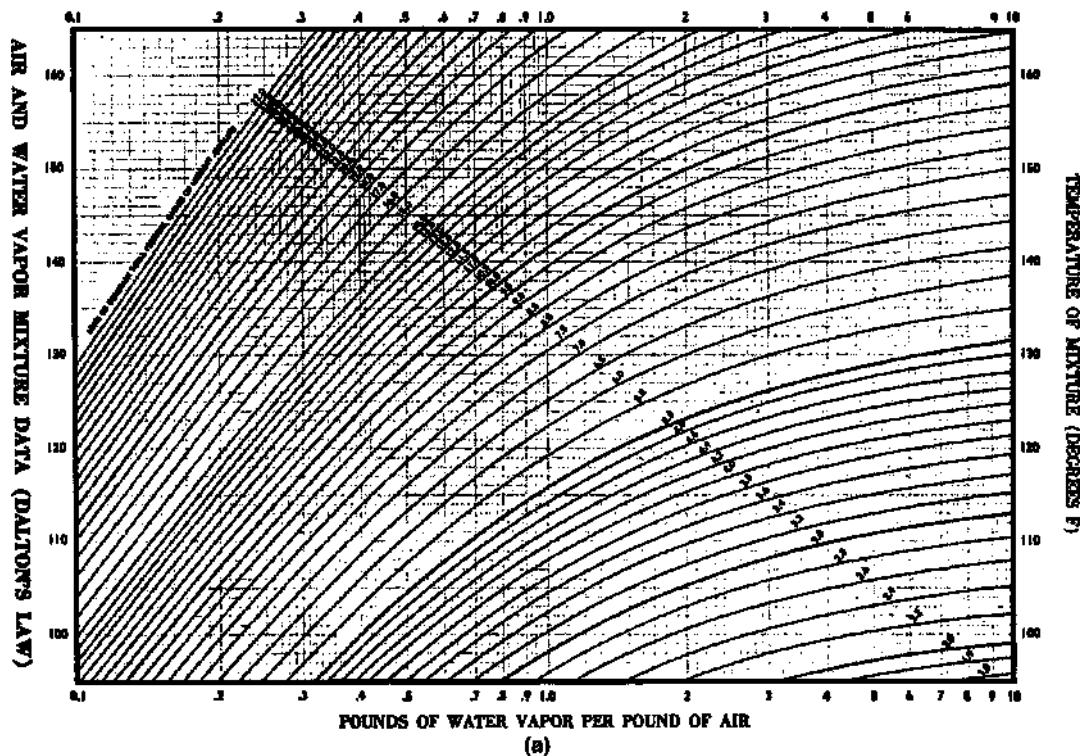
For 3 in. Hg abs and 95° F saturation, the fraction of water vapor from Figure 8-20 is 0.77 lbs water/lb air.

$$\text{Total water vapor} = (50)(0.77) = 38.5 \text{ lb/h}$$

$$\text{Total mixture} = 50 + 38.5 = 88.5 \text{ lb/h}$$

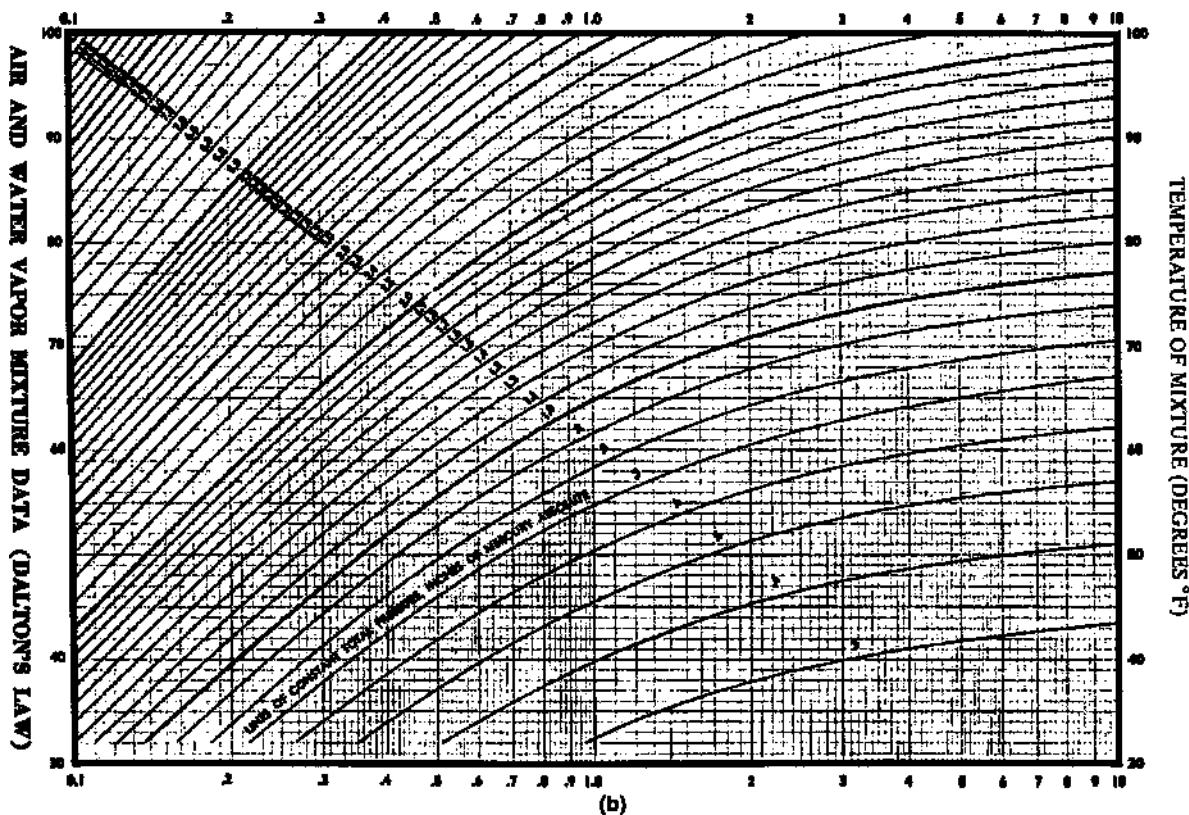
Weight of Air and Water Vapor Mixture

$$W_m = W_a + \frac{0.62(W_a)(P_v)}{P_a} \quad (8-4)$$



(a)

**Figure 8-20** Air and water vapor mixture data (Dalton's Law), saturated. (Reprinted by permission from *Standards for Steam Jet Ejectors*, 3rd ed., Heat Exchange Institute, 1956 [7].)



(b)

**Figure 8-20** (Continued)

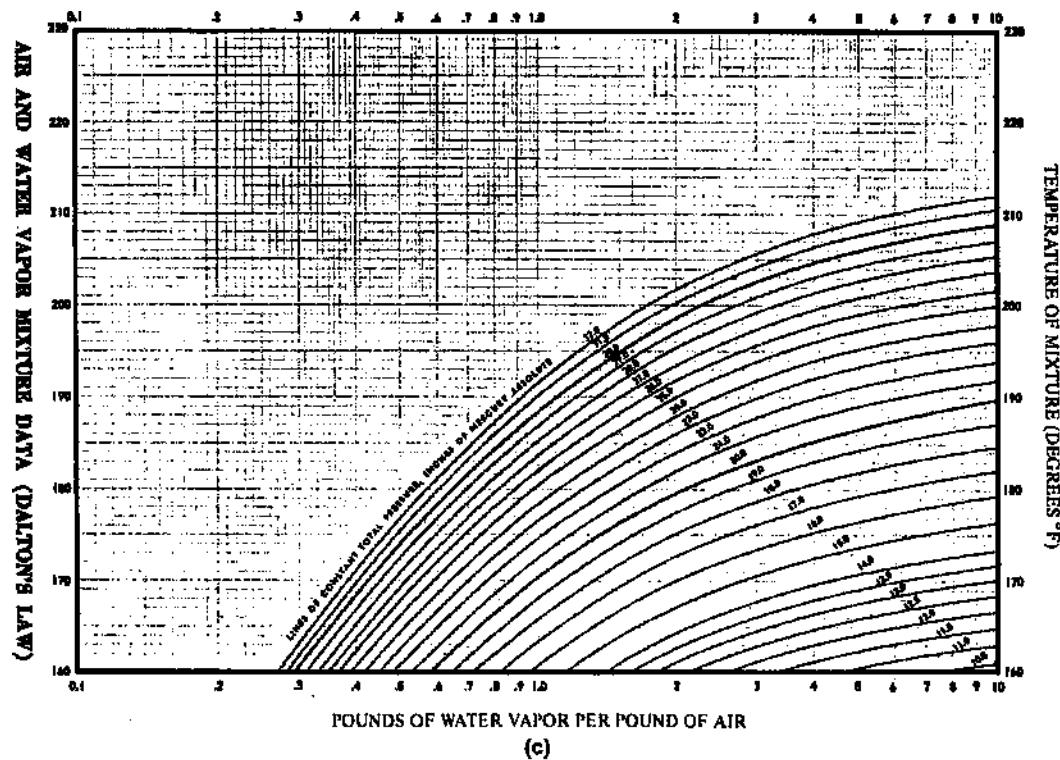


Figure 8-20 (Continued)

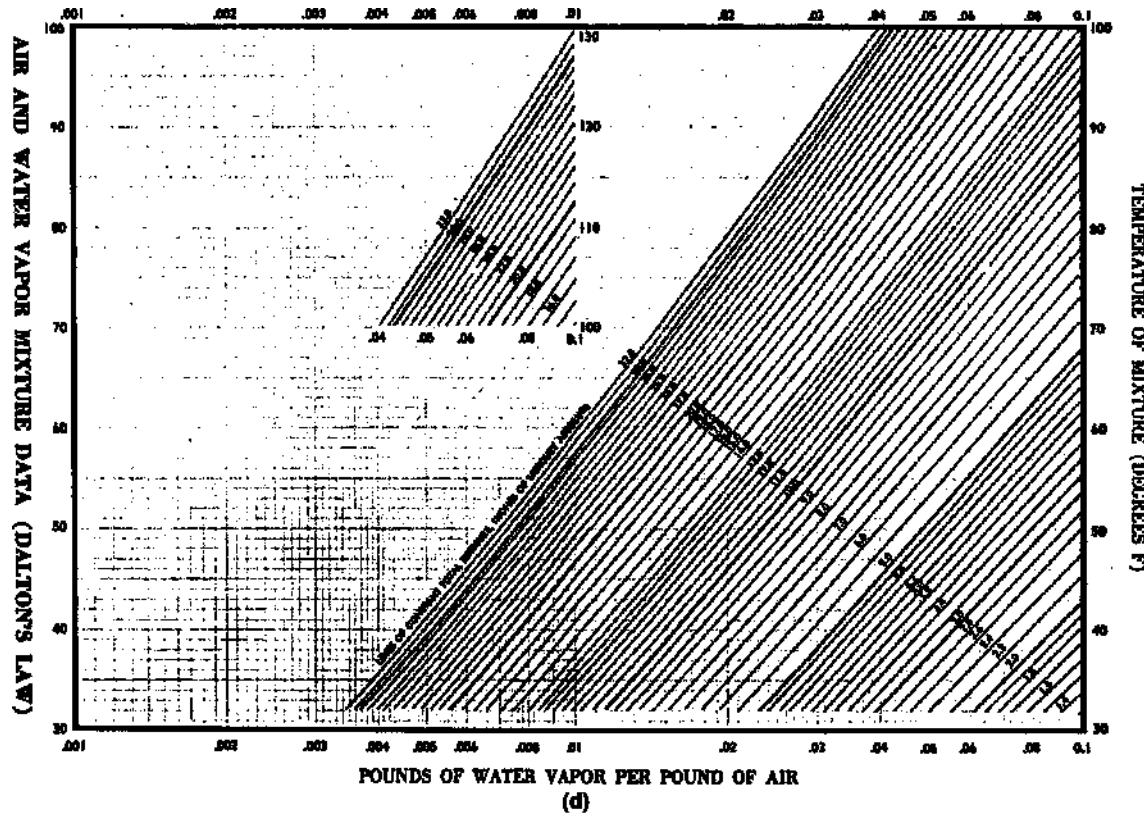


Figure 8-20 (Continued)

### EXAMPLE 8-7 Total Weight of Mixture

Calculate the total weight of mixture to be handled when evacuating 25 lb/h of air from 28.5 in. vacuum with a mixture temperature of 80° F. Barometer = 30 in. Hg.

$$\text{Absolute pressure} = 30.0 - 28.5 = 1.5 \text{ in. Hg abs}$$

$$P_v = 1.034 \text{ in. Hg abs (water vapor at } 80^\circ \text{ F)}$$

$$P_a = 1.50 - 1.034 = 0.466 \text{ in. Hg abs (air)}$$

$$W_m = 25 + 0.62(25)(1.034)/0.466 = 59.4 \text{ lb mixture/h}$$

### TOTAL VOLUME OF A MIXTURE

The total fixed volume of a mixture of gases and vapors at a given condition is the same as the volume of any one component (gas laws), and its pressure is composed of the sum of the individual partial pressures of each component.

### EXAMPLE 8-8 Saturated Water Vapor-Air Mixture

An air-water vapor mixture is saturated with water vapor at 80° F and 2 in. Hg abs total pressure. The air in the mixture is 60 lb/h. Determine the volume at these conditions.

Vapor pressure water at 80° F = 1.034 in. Hg abs

Partial pressure air = 2.0 - 1.034 = 0.966 in. Hg abs

Weight of mixture:

$$W_m = 60 + 0.62(60)(1.034)/0.966 = 99.8 \text{ lb/h}$$

Weight of water vapor = 39.8 lb/h (from above)

Volume of air under its condition in the mixture:

Volume of air = volume of mixture

$$= V = \frac{WRT}{70.73 P_a}, \text{ use in. Hg} \quad (8-5)$$

$$\text{Gas constant, } R = 1544/\text{MW} = 1544/29 = 53.3 \quad (8-6)$$

$$V = \frac{60(53.3)(460+80)}{70.73(0.966)} = 25,300 \text{ ft}^3/\text{h}$$

$$= 421 \text{ ft}^3/\text{min at } 80^\circ \text{ F}$$

From steam tables, specific volume of water vapor at 80° F and 1.034 in. Hg abs = 633.8 ft<sup>3</sup>/lb

$$\text{Volume} = (39.8)(633.8) = 25,225 \text{ ft}^3/\text{h} = 420.4 \text{ ft}^3/\text{min}$$

As an alternate method, total moles could be calculated and converted to volume at 80° F and 2 mmHg abs.

### AIR INLEAKAGE INTO SYSTEM

Few vacuum systems are completely airtight although some may have extremely low leakage rates. For the ideal system the only load for the ejector is the non-condensables of the process

(absorbed gases, air, etc.) plus the saturated vapor pressure equivalent of the process fluid. Practice has proven that allowance must be made for air leakage, considering the air and non-condensables. For "base" ejector capacity determine inert gases only by:

$$\begin{aligned} & (\text{lb/h air} + \text{non-condensables} + \text{process released air} \\ & + \text{process released non-condensables})/\text{h} \\ & \cong \text{Air inleakage, lb/h} \end{aligned} \quad (8-7)$$

For the design of a new system, it is recommended that the results of the summation above be multiplied by 2 or 3 to establish the jet system inert (non-condensables) capacity, and add to this the non-condensed process vapors that are released into the jet suction system.

Air leakage occurs at piping connections (flanges, screwed fittings, valves), stuffing boxes, mechanical equipment seals, and so on. Whenever possible, a system should be tested to determine the air leakage [16, 16], but for new designs and those situations where tests cannot be made, the recommended values of the Heat Exchange Institute are given in Table 8-6 for ejectors serving surface condensers, and are minimum safe values. A very tight system will show better performance.

Figure 8-21 gives maximum air leakage values for commercially tight process systems which do not include any agitator equipment. For design purposes the ejector is usually purchased to operate on a load at about twice these values. For systems with agitators and ordinary shaft seals, the system leakage should be increased by 5 pounds of air per hour [16] per agitator. If special seals are used this value may be reduced to 1 or 2. The more rotating shafts which must be sealed to the outside atmosphere, the more likely will be the possibilities of increased leakage.

An alternate design for air inleakage used by some manufacturers and process engineers is shown in Figure 8-21 plus the summation obtained by examining the process system using the factors of Table 8-7. This method is considered to be conservative; however, as in general the incremental cost may be very small between a unit barely large enough and one which has ample capacity to take surges in air leakage.

Since the determination of air inleakage involves considerable knowledge of vacuum systems and judgment, no empirical method can be expected to yield exact and correct values. Most manufacturers use one of the methods presented here, together with a factor to account for the basic type of plant, maintenance practices, operational techniques of the production personnel, and other related items. Thus, for a tight and efficient plant, the leakage values of Figure 8-21 may sometimes be reduced to 0.75 of the values read, while for a sloppy, loose-run plant, the values might be multiplied by 2 or 3, or the alternate method using Table 8-7 may be checked, or even multiplied by 2 or 3.

Another recommendation based upon years of experience is reasonably conservative [12]:

Suction pressure (in. Hg abs)	Allowance for air leakage (lb/h)
8-15	30-40
5-8	25-30
3-5	20-25
1-3	10-20

For systems with moving sealed parts, make extra allowance and consult the seal manufacturer.

Air leakage into systems operating at or below 0.53 atm, or 15 in. Hg abs, is constant and approximately independent of the

**TABLE 8-6 Vacuum Pump Capacities from Steam Surface Condensers**

Maximum Steam Condensed (lb/h)	Dry Air at 70° F			
	Serving Turbines		Serving Engines	
	SCFM	lb/h	SCFM	lb/h
Up to 25,000	3.0	13.5	6.0	27.0
25,001–50,000	4.0	18.0	8.0	36.0
50,001–100,000	5.0	22.5	10.0	45.0
100,001–250,000	7.5	33.7	15.0	67.4
250,001–500,000	10.0	45.0		
500,001 and up	12.5	56.2		

Rapid Evacuator Capacities, dry Air, cfm at 70° F, 15 in. Hg abs	
Up to 75,000	150
75,000–250,000	300
250,001–600,000	600
600,001 and up	900

By permission from Standards of Heat Exchange Institute, *Steam Surface Condensers*, 3rd ed., Ref. [16].

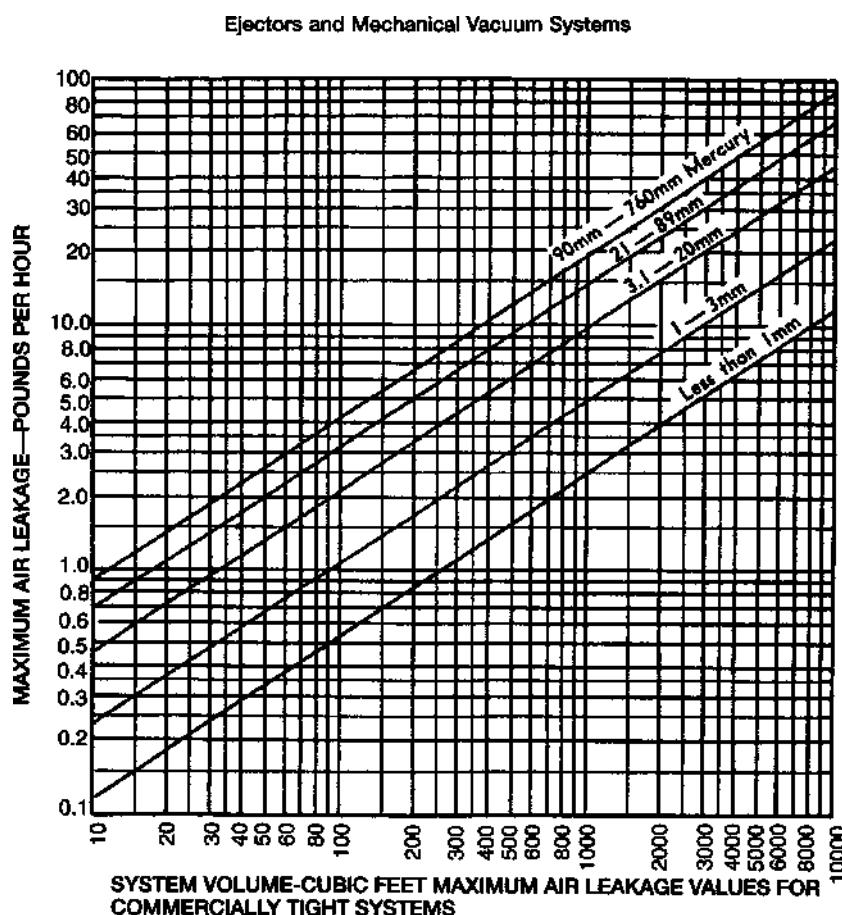


Figure 8-21 Maximum air leakage values for commercially tight systems. (Reprinted by permission from [7].)

**TABLE 8-7 Estimated Air Leakage into Equipment Vacuum System**

Type Fitting	Estimated Average Air Leakage (Lb/h)
Screwed connections in sizes up to 2 in.	0.1
Screwed connection in sizes above 2 in.	0.2
Flanged connections in sizes up to 6 in.	0.5
Flanged connections in sizes 6–24 in. including manholes	0.8
Flanged connections in sizes from 24 in. to 6 ft	1.1
Flanged connections in sizes above 6 ft	2.0
Packed valves up to 1/2" stem diameter	0.5
Packed valves above 1/2" stem diameter	1.0
Lubricated plug valves	0.1
Petcocks	0.2
Sight glasses	1.0
Gage glasses including gage cocks	2.0
Liquid-sealed stuffing box to shaft of agitators, pumps and so on, per inch shaft diameter	0.3
Ordinary stuffing box, per inch of diameter	1.5
Safety valves and vacuum breakers, per inch of nominal size	1.0

From D.H. Jackson, "Selection and Use of Ejectors", *Chem. Eng. Prog.*, Vol. 44, 1948, p. 347.

process itself. From atmospheric down to 15 in. Hg, the air leakage increases as the pressure decreases.

#### DISSOLVED GASES RELEASED FROM WATER

When ejectors pull non-condensables and other vapors from a direct contact water condenser (barometric, low level jet, deaerator), there is also a release of dissolved gases, usually air, from water. This air must be added to the other known load of the ejector. Figure 8-22 presents the data of the Heat Exchange Institute [17] for the amount of air that can be expected to be released when cooling water is sprayed or otherwise injected into open-type barometric or similar equipment.

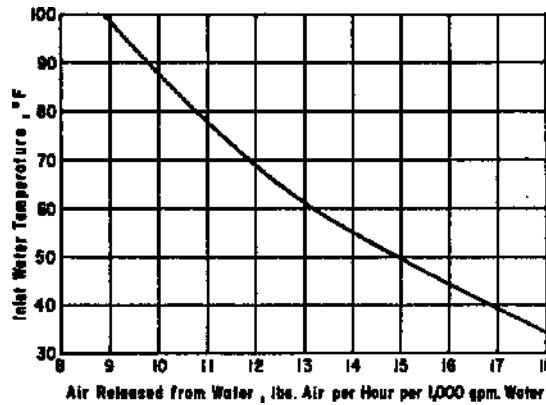
#### EXAMPLE 8-9 Ejector Load For Steam Surface Condenser

A surface condenser condensing the steam from a process turbine drive operates at 1.0 in. Hg abs. The condensing load is 85,000 lb/h steam. What is the capacity of the ejector?

#### *Solution*

The temperature of the condensing steam at 1.0 in. Hg is 79° F (from steam-tables).

The saturation pressure corresponds to a temperature of 79 – 7.5° F = 71.5° F, based on condenser-ejector design practice. The pressure from steam tables = 0.78 in. Hg abs. The water vapor to saturate the air going to the ejector is:



**Figure 8-22** Dissolved air released from water on direct contact in vacuum systems. (Reprinted by permission from *Standards for Direct Contact Barometric and Low Level Condensers*, 4th ed., Heat Exchange Institute, 1957.)

Alternate, for additional air inleakage, calculations are presented by Reference [19] based on industrial studies and experience. The objective is to be able to develop reliable specifications for "rough vacuum" equipment. The procedure is [3]:

1. Estimate air leakage into the system based on possible weld cracks, metal weld, or metal porosity:

$$1 \leq P < 10 \text{ torr}; W'_a = 0.026P^{0.34}V^{0.60} \quad (8-9)$$

$$10 \leq P < 100 \text{ torr}; W'_a = 0.032P^{0.26}V^{0.60} \quad (8-10)$$

$$100 \leq P < 760 \text{ torr}; W'_a = 0.106V^{0.60} \quad (8-11)$$

where

$P$  = system operating pressure, torr

$V$  = system volume, ft<sup>3</sup>

$W'_a$  = air inleakage resulting from metal porosity and cracks along weld lines, lb/h

2. Estimate acceptable air leakages resulting from leakage around static and rotary seals, valves, access ports, and other items of mechanical nature required for process operation from the

$$w'_v = 0.62 \frac{P_v}{P_a} = (0.62)(0.78)/(1 - 0.78) \quad (8-8)$$

$$= 2.20 \text{ lb water vapor/lb air}$$

From Table 8-6 the recommended dry air SCFM (standard cubic feet per minute) = 5.0 (22.5 lb/hr)

The equivalent

$$\text{lb/h} = (5.0)(60)(0.075 \text{ lb/ft}^3) = 22.5$$

$$\text{Total water vapor} = (2.20)(225) = 49.5 \text{ lb/h}$$

$$\text{Total vapor mixture to ejector} = 49.5 + 22.5 = 72 \text{ lb/h}$$

For ejector design a value of 1.25 times this value is recommended.

**TABLE 8-8 Specific Air Inleakage Rates for Rough Vacuum (For Use with Equations)**

Component	$\theta = \text{specific leak rate}^* (\text{lb/h/in.})$
Static seals	
O-ring construction	0.002
Conventional gasket seals	0.005
Thermally cycled static seals	
$t \times 200^\circ\text{F}$	0.005
$200 \times t < 400^\circ\text{F}$	0.018
$t \times 400^\circ\text{F}$	0.032
Motion (rotary) seals	
O-ring construction	0.10
Mechanical seals	0.10
Conventional packing	0.25
Threaded connections	0.015
Access ports	0.020
Viewing windows	0.015
Valves used to isolate system	
Ball	0.02
Gate	0.04
Globe	0.02
Plug-cock	0.01
Valves used to throttle control	
Gas into vacuum system	0.25

By permission from Ryans and Croll [3].

\* Assumes sonic (or critical) flow across the component.

following equations and the specific leak rates  $\theta$  indicated in Table 8-8, which is based on  $w_a \leq 10 \text{ lb/h}$ .

$$1 \leq P < 10 \text{ torr}; w_a = \pi D \theta P^{0.34} \quad (8-12)$$

$$10 \leq P < 100 \text{ torr}; w_a = 1.2 \pi D \theta P^{0.26} \quad (8-13)$$

$$100 \leq P < 760 \text{ torr}; w_a = 3.98 \pi D \theta \quad (8-14)$$

where

$D$  = sealed diameter, in. (estimates of nominal diameter acceptable)

$w_a$  = acceptable air leakage rate assigned to a system component, lb/h

$\theta$  = specific air leakage rate, lb/h/in.

$P$  = system operating pressure, torr

$W_{Ta} = W_T$  = total calculated air inleakage, lb/h

3. Calculate the total acceptable air leakage rate,  $W_T$ , lb/h, by adding  $\sum W'_a$  to the sum of the leak rates assigned to the individual system components,  $w_a$

$$W_T = \sum W'_a + \sum w_a \text{ lb/h} \quad (8-15)$$

To determine the capacity of the vacuum pump, the values of  $W_T$  above should not be used. It is necessary to apply over-design or safety factors to ensure reliability [19] because pump capacity decreases with time and wear and air inleakage surges can occur due to a wide variety of leak-developing situations that result in more air or surges of air inleakage. The over-design factor should be applied only to the pump inlet throughput specification. The recommended [3] over-design factor should be 1.5 to 2.0 times the air inleakage rate [3], and should also be applied to saturated vapors entering the vacuum equipment at the suction conditions required. Do not put a safety factor on the suction condition of temperature or pressure, provided the worst expected conditions of operation are specified. This requires a close examination of the process flowsheet range

of operation. A safety factor of 2.0 is recommended for multistage steam jets with compression ratio above 6:1; while a 1.5 factor is adequate for most mechanical pumps and single-stage jets with a compression ratio of under 4:1 [3]. The above procedure can be simplified for preliminary inleakage calculations [3]:

$$\text{Multiply } W'_a \times 2$$

For equipment with rotary seals, allow additional 5 lb/h for each conventional seal (packing type) and 2 lb/h for each mechanical seal and O-ring.

To account for air inleak for vessels containing a liquid level (portion of vessel submerged), the following applies.

If there is a large pinhole leak a few inches below the liquid surface, it will behave like a leak above the liquid. However, a small pinhole leak in the same location may have zero inleakage due to capillary effects. The problem becomes complicated as the depth becomes large, and flow transition occurs from critical (sonic) to subsonic flow through any particular leak. These factors can be calculated by conventional fluid flow methods if the size of the leak is estimated, which then is a real problem, or by using the data of Table 8-8 and applying fluid flow head losses.

The hydraulic effects for the submerged portions of the vessel or system can be ignored when the total static head on the submerged portion is not greater than 0.53 times atmospheric pressure plus the hydraulic head [3], that is  $P_s$ :

$$P_s = (p/760) + (h_L \rho_L/34) \quad (8-16)$$

where

$P_s$  = static pressure, atm

$p$  = atmospheric pressure, mmHg

$h_L$  = liquid height, ft, below liquid surface

$\rho_L$  = specific gravity of liquid, relative to water = 1.0

#### ACCEPTABLE AIR INLEAKAGE RATES [1]

In order to estimate an acceptable air inleakage rate for sizing a vacuum pump for use in the medium to high vacuum system, consider:

100 microns (0.1 torr) to 0.1 torr Range

Estimate  $W'_a$ :

$$0.1 \leq P < 1 \text{ torr}; W'_a = 0.026 P^{0.64} V^{0.6} \quad (8-17)$$

Estimate air inleakage for individual system-specific leak rates,  $\theta$ , from Table 8-8 [3] and from ( $w \leq 5 \text{ lb/h}$ )

$$0.1 \leq P < 1 \text{ torr}; w_a = \pi D \theta P^{0.64} \quad (8-18)$$

Note that estimating the maximum acceptable differs from the design equations for  $w_a$  and  $W'_a$

Calculate total acceptable air inleakage rate,  $W_T$  using Eq. (8-15)

$$W_T = \sum W'_a + \sum w_a$$

A simplified alternate to the previously cited procedures is suggested by Gomez [5] for calculating air inleakage, but it is not presented in detail here.

## TOTAL CAPACITY AT EJECTOR SUCTION

The total capacity is the sum of all the expected condensables and non-condensable flow quantities (in lb/h) which will enter the suction inlet of the ejector. It consists of the following:

1. Air leakage from surrounding atmosphere.
2. Non-condensable gases released from gases originally injected into the process for purge, products of reaction, and so on.
3. Non-condensable gases, usually air, released from direct contact water injection.
4. Condensable vapors saturating the non-condensables.

Reasonable factors of safety should be applied to the various loads in order to insure adequate capacity. Excess ejector capacity can be handled by pressure control and some adjustment in steam flow and pressure, but insufficient capacity may require ejector replacement. Factors of 2.0–3.0 are not uncommon, depending upon the particular type of system and knowledge of similar system operations.

## CAPACITIES OF EJECTOR IN MULTISTAGE SYSTEM

When the ejector system consists of one or more ejectors and intercondensers in series, the volume as pounds per hour of mixture to each succeeding stage must be evaluated at conditions existing at its suction. Thus, the second-stage unit after a first-stage barometric intercondenser handles all of the non-condensables of the system plus the released air from the water injected into the intercondenser, plus any condensable vapors not condensed in the condenser at its temperature and pressure. Normally the condensable material will be removed at this point. If the intercondenser is a surface unit, there will not be any air released to the system from the cooling water.

## BOOSTER EJECTOR

Booster ejectors are designed to handle large volumes of condensable vapors at vacuums higher than that obtainable with standard condensers using cooling water at the maximum available temperature. They are usually used with a barometric (or surface) condenser

and standard two-stage ejectors. The booster picks up vapors from the process system at high vacuum (low absolute pressure, around 0.5 in. Hg abs) and discharges them together with its own motivating steam to a lower vacuum condition (compresses the mixture) where the condensable vapors can be removed at the temperature of the condenser water. The non-condensable vapors leave the condenser, passing to the two-stage ejector system. This overall system allows a constant vacuum to be maintained in the process, unaffected by the temperature of the cooling water. Booster ejectors are used with barometric and surface steam jet refrigeration systems, degassing of liquids, high vacuum distillation, evaporation, vacuum cooling and vacuum drying, or other systems where large volumes of condensable materials are to be removed at high vacuum. Figure 8-23 illustrates one application.

## EVACUATION EJECTOR

An evacuation booster or "hogging" ejector is sometimes used to remove air from a system on start-ups. Its capacity is set to bring the system pressure down to near operating conditions before the continuous operating ejector system takes over. Figure 8-23 illustrates the installation of such a unit.

When an extra jet for this purpose is not desirable, the secondary jet of a multiple system is often sized to have sufficient air removal capacity to pump down the system in a reasonable time.

## 8.12 LOAD VARIATION

Figure 8-24 illustrates three different multistage ejector designs, A, B, and C, which indicate that design A is quite sensitive to changes in load above the design point. Design B or C is less sensitive. The curve extended toward point D indicates the capacity of the primary or first stage when the entire vapor is condensed in the intercondenser; or if handling air or an air-vapor mixture, the performance when the secondary jets have sufficient capacity to take all the non-condensables.

The curve labeled A indicates performance at overload when the air-handling capacity of the secondary stage is limited.

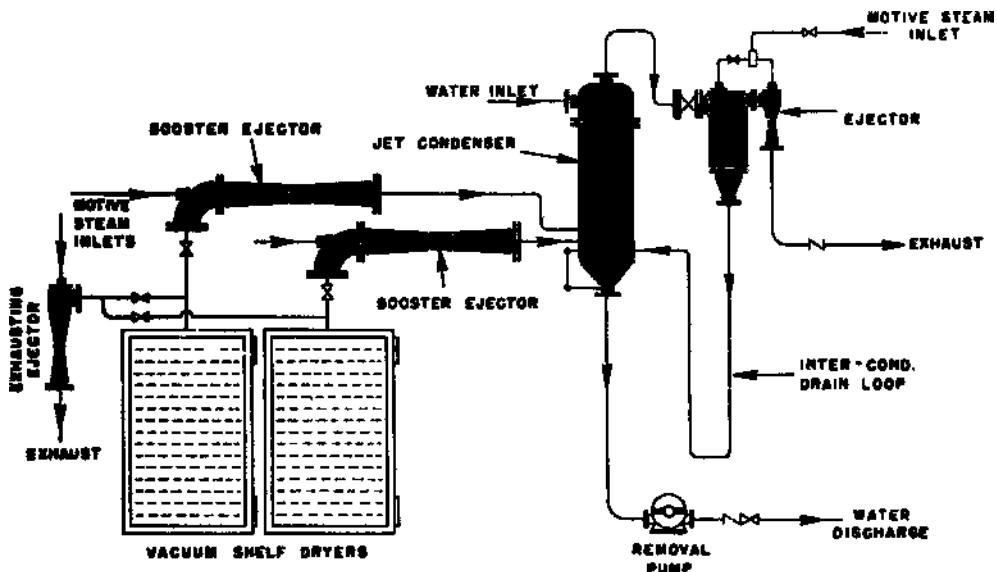
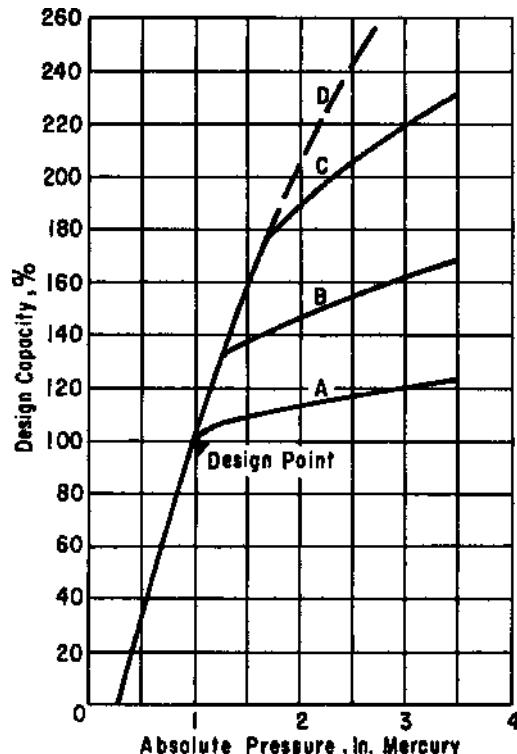


Figure 8-23 Drying-special three-stage assembly serving two high vacuum dryers. (By permission from C.H. Wheeler Mfg. Co.)



**Figure 8-24** Multistage ejector designs. (By permission from C.H. Wheeler Mfg Co.)

This condition arises as a result of design for steam economy. If the capacity of the secondary jets is larger, the performance along curve B or C can be expected. When the secondary jet capacity is limited as curves A, B, and C indicate, a capacity increase brings a rise in suction pressure when the load increase is mainly air or non-condensables. The increase in pressure is less when the load increase is due to condensables. This emphasizes the importance in sizing the secondary jets for ample non-condensable capacity, and the importance of specifying the range and variety of expected conditions which may confront the system.

Once a system has been evacuated to normal operating conditions, it is possible for the capacity to fall to almost zero when the only requirement is air inleakage or small quantities of dissolved gases. Under these conditions, it is important to specify an ejector system capable of stable operation down to zero load or "shut-off" capacity. The curve of Figure 8-24 represents such a system.

### 8.13 STEAM AND WATER REQUIREMENTS

Figure 8-25 presents estimated steam requirements for several ejector systems. Exact requirements can be obtained only from the manufacturers, and these will be based on a specific performance.

Figures 8-26a and b give typical good estimating selection curves for single-stage ejectors. Table 8-9 gives evacuation factors.

Size selection: Locate size at intersection of ejector suction pressure and capacity on Figure 8-26a.

Steam consumption: Read values on curves or interpolate.

$$W_s = W_{s90}F \quad (8-19)$$

Evacuation:

$$W'_m = EV/t \quad (8-20)$$

**TABLE 8-9 Evacuation Factors for Single-Stage Jet**

Final Suction Pressure (in. Hg Abs)	Evacuation Factor ( <i>E</i> )
10	1.9
8	1.0
6	1.5
5	1.0
4	1.3

By permission from Worthington Corp., Bulletin W-205-S1A.

### EXAMPLE 8-10

#### Size Selection – Utilities and Evacuation Time for Single-Stage Ejector

Total mixture to be handled = 60 lb/h

Suction pressure: 4 in. Hg abs

Steam pressure: 125 psig

Size selection: 2-in. length (Figure 8-26a)

Steam consumption: 440 lb/h at 90 psig

at 125 psig,  $F = 0.88$  (Figure 8-26b)

$$W_s = 440(0.88) = 397 \text{ lb/h}$$

Evacuation: System volume = 300 ft<sup>3</sup>

$$E = 1.3 \text{ (Table 8-9)}$$

$$V = 300$$

$$W'_m = 60$$

$$60 = 1.3(300/t)$$

$$t = 6.5 \text{ min to evacuate the volume with the 2-in. length ejector}$$

Rating of the two-stage non-condensing ejectors is handled in the same manner as for a single-stage ejector, using Figures 8-27a and b, and Table 8-10.

Figures 8-28a–e give representative estimating data for two-stage ejectors with barometric intercondenser.

Size selection: locate size at intersection of ejector suction pressure and capacity on Figure 8-28a.

Steam consumption: Use Figure 8-28b; for *K* use Figure 8-28c

$$W_s = W'_m W_m K F \quad (8-21)$$

Water consumption:

$$\text{GPM (approximate)} = 0.06 W_s$$

$$\text{Minimum GPM} = 10$$

### 8.14 EJECTOR SYSTEM SPECIFICATIONS

Figure 8-29 is helpful in summarizing specifications to the ejector manufacturer for rating of ejectors and intercondensers to serve a specific process application. Most conditions require a detailed evaluation with the manufacturer's test curves, as very few complicated systems requiring high vacuum can be picked from stock items. The stock items often fit single-stage vacuum requirements for process and such standardized situations as pump priming.

Figure 8-29 is also adaptable to air and water ejector applications.

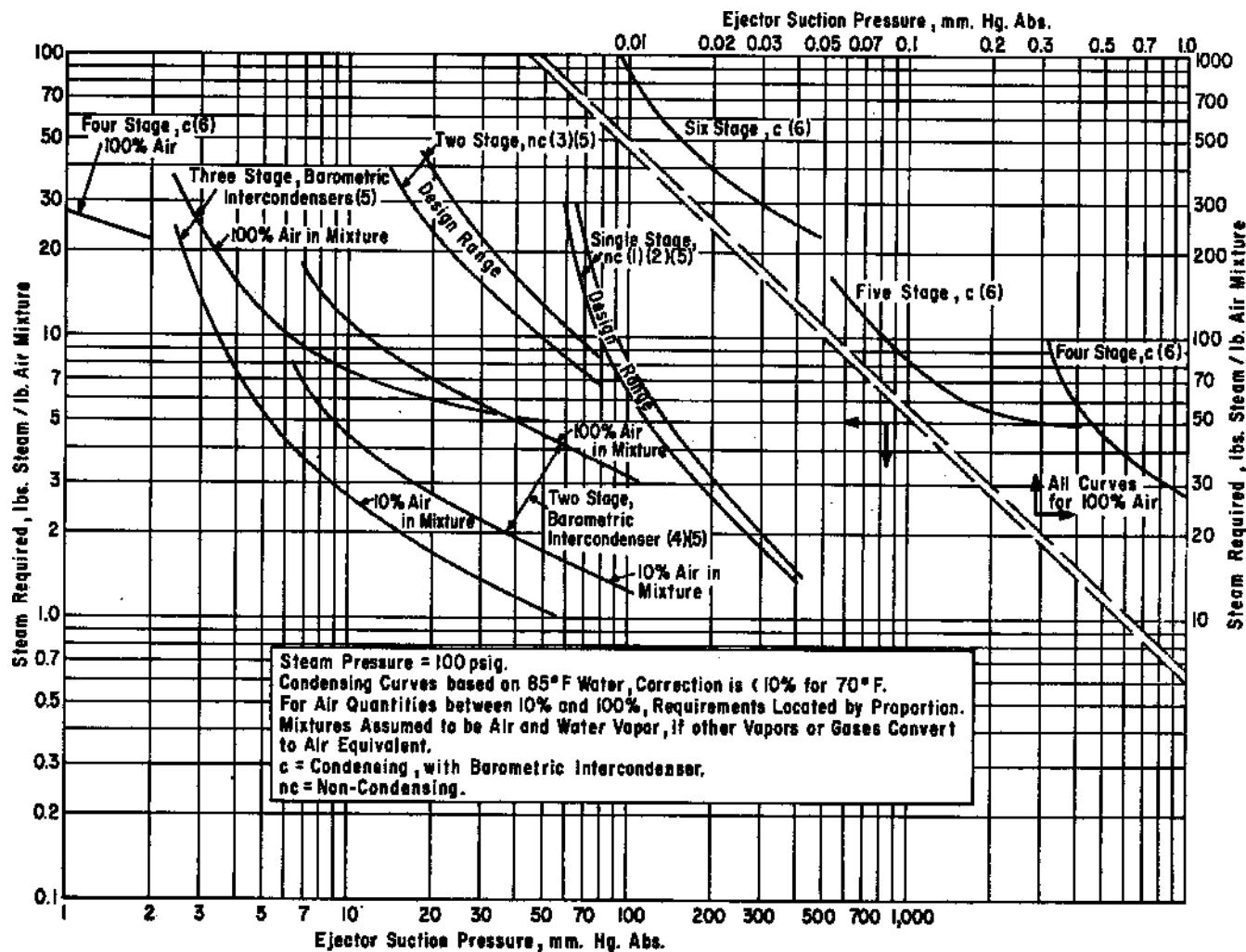


Figure 8-25 Estimating steam requirements for ejectors.

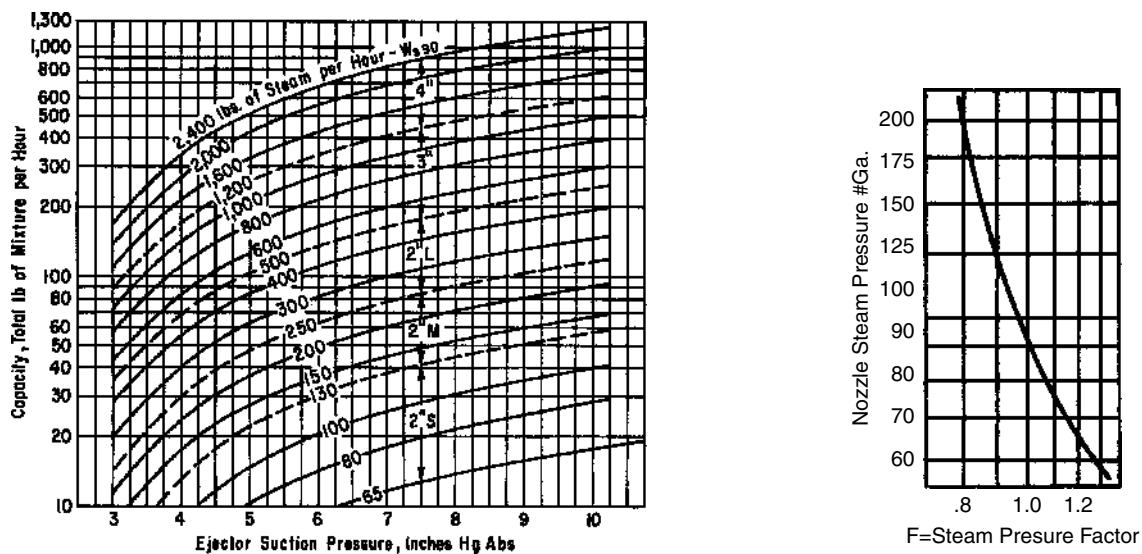
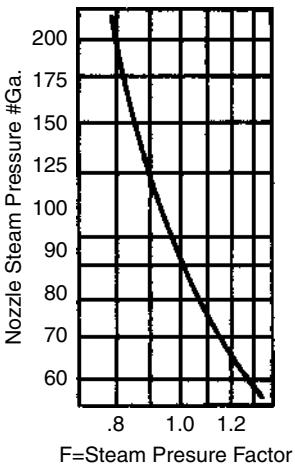
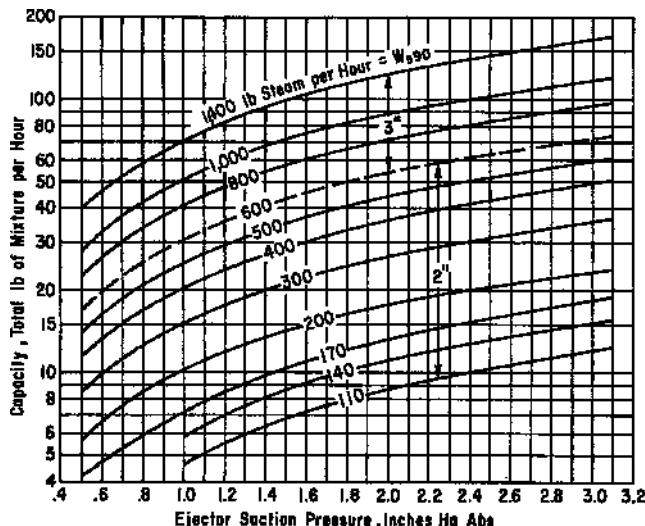


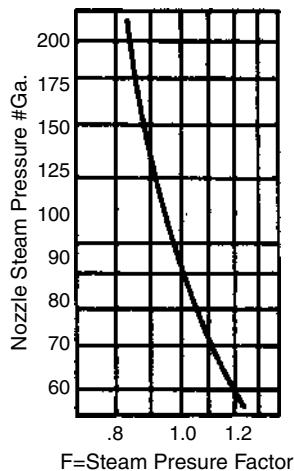
Figure 8-26a Size ejector for 20 psig steam consumption, single stages 3–10 in. Hg abs. (By permission from Worthington Corp.)

Figure 8-26b Steam pressure factor for Figure 8-26a. (By permission from Worthington Corp.)





**Figure 8-27a** Size ejector for 90 psig steam consumption, two-stage, non-condensing 0.6–3.0 in. Hg abs. (By permission from Worthington Corp.)



**Figure 8-27b** Steam pressure factor for Figure 8-27a. (By permission from Worthington Corp.)

In all cases, it is important to describe the system, its requirements, control, and method of operation in the specifications. The manufacturer needs complete data concerning the motive steam (air or water) and the condensable and non-condensable vapors.

Utility unit costs as well as any preference for maximum operating economy or minimum first cost should be stated if the manufacturer is to make a selection to best-fit the plant system and economics.

Tables 8-11–8-14 provide useful reference data.

### 8.15 EJECTOR SELECTION PROCEDURE

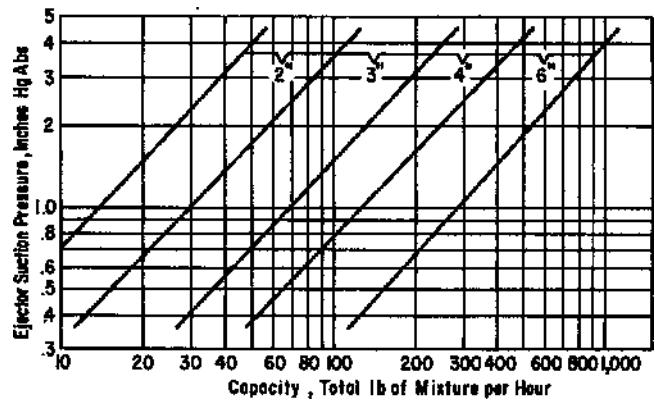
As a guide, the following is a suggested procedure for rating and selecting an ejector system for vacuum operation.

1. Determine vacuum required at the critical process point in system.
2. Calculate the pressure drop from this point to the process location of the suction flange of the first-stage ejector.

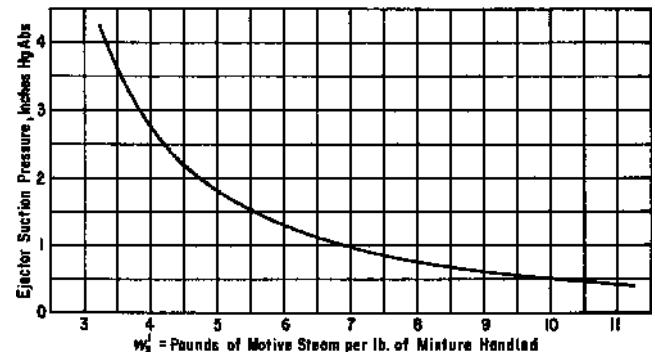
**TABLE 8-10 Evaluation Factors for Two-Stage Non-Condensing Ejectors**

Final Suction Pressure (in. Hg abs)	Evacuation Factor ( <i>E</i> )
0.5	0.48
1.0	0.67
1.5	0.81
2.0	0.92
2.5	1.00
3.0	1.10

By permission from Worthington Corp., Bulletin W-205-S1A.



**Figure 8-28a** Ejector size, single-stage – typical. (By permission from Worthington Corp.)



**Figure 8-28b** Steam consumption factor. (By permission from Worthington Corp.)

3. At ejector suction conditions, determine:
  - a. Pounds/hour of condensable vapor
  - b. Pounds/hour of non-condensable gases
    - i. dissolved
    - ii. injected or carried in process
    - iii. formed by reaction
    - iv. air inleakage
4. Prepare specification sheet, Figure 8-29, and forward to manufacturers for recommendation.
5. For checking and estimating purposes, follow the guides as to the number of stages, utility requirements, and so on presented herein.

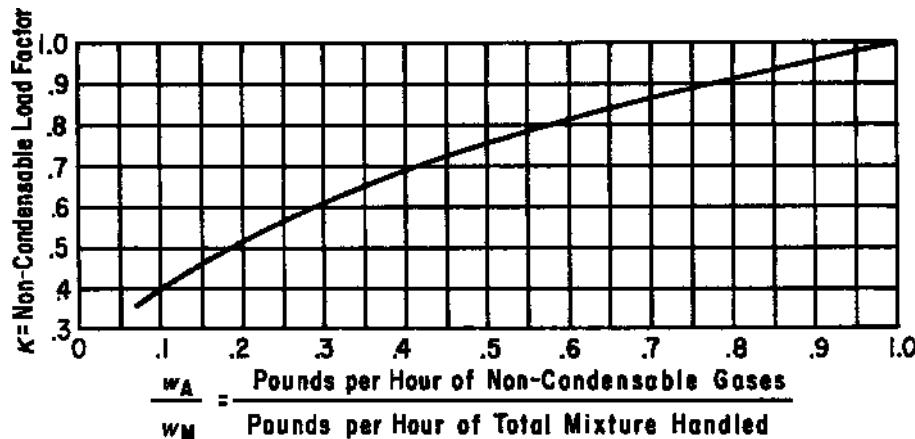


Figure 8-28c Non-condensable load factor. (By permission from Worthington Corp.)

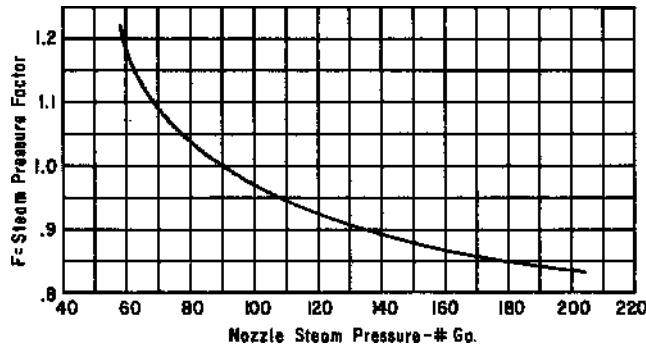


Figure 8-28d Steam pressure factor. (By permission from Worthington Corp.)

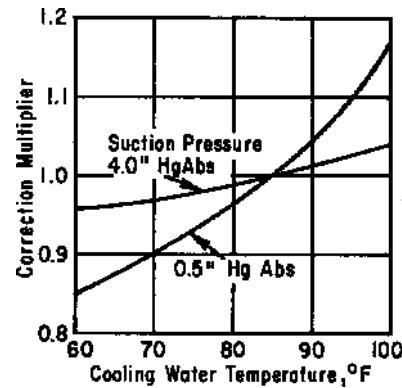


Figure 8-28e Steam requirement correction for two-stage unit with barometric intercooling. (By permission from Fondrk, V.V., *Petroleum Refiner*, Vol. 37, No. 12, 1958 [20].)

#### EXAMPLE 8-11

#### Size Selection and Utilities for Two-Stage Ejector with Barometric Intercondenser

Total mixture to be handled = 40 lb/h

Pounds of air in mixture = 14 lb/h

Suction pressure = 1.5 in. Hg abs

Steam pressure at ejector nozzle = 150 psig

Size selection, using Figure 8-28a: 2 in. (inlet and outlet connections)

Steam consumption:

$$W'_s = 5.55 \text{ (Figure 8-28b)}$$

$$W_m = 40$$

$$K = 0.67 \text{ (Figure 8-28c) at } W_a/W_m = 15/40 = 0.375$$

$$F = 0.88 \text{ (Figure 8-28d)}$$

$$W_s = 5.55(40)(0.67)(0.88)$$

$$= 131 \text{ lb steam/h}$$

Water consumption:

$$\text{GPM} = 0.06(131) = 7.86$$

Use 10 gpm minimum

#### BAROMETRIC CONDENSERS

Barometric condensers are direct contact coolers and condensers. They may be counter flow or parallel flow. Good contact direct cooling is an efficient inexpensive design, being considerably cheaper and more efficient than indirect surface or tubular coolers.

#### TEMPERATURE APPROACH

When serving vacuum equipment, the temperatures are usually set as follows when the non-condensables do not exceed 1% of the total water vapor being condensed. See Figures 8-20a-d.

Terminal Difference, steam temperature corresponding to vacuum-less outlet water temperature = 5°F.

Job. No.	SPEC. DWG. NO.				
B/M No.	A-	Page	of	Pages	
	Condenser:	<input type="checkbox"/>	Barometric	<input type="checkbox"/>	Surface
<b>STEAM EJECTOR SPECIFICATIONS</b>					
<b>PERFORMANCE</b>					
Make	Type				
Service	Condenser:	<input type="checkbox"/>	Barometric	<input type="checkbox"/>	Surface
No. of Stages	No. of Ejectors per Stage				
Suct. Press.	mmHg abs.	Suct. Temp.	°F	Max. Dsich. Press.	mmHg abs.
Steam: Min. Press.	psia	Temp.	°F	Quality	%
Water: Source		Max. Press.	psia	Max. Temp.	°F
Vol. of Evacuated System		Cu. Ft.			
Expected Air Leakage		lb/h			
Max. Evacuating Time		Min.			
Ejector Load	lb/h	Mol. Wt.	Cp, btu/lb-°F	Latent Ht., btu/lb	
Condensables					
Non-Condensables					
<b>DESIGN</b>					
	1st Stage	2nd Stage	3rd Stage	4th Stage	5th Stage
Propelling Steam, lb/h					
Steam: Inlet Size					
Press. Class & Facing					
Water, GPM					
Water $\Delta T$ , °F					
Water: Inlet Size					
Press. Class & Facing					
Water: Exit Size					
Press. Class & Facing					
Suct. Chamber Press. mmHg abs.					
Suct. Chamber Temp. °F					
Condensers: Pre-Inter-After					
Barometric: No. Contact Stages					
Surface: Outside Tube Area Sq. Ft.					
<b>MATERIALS OF CONSTRUCTION</b>					
Ejector: Steam Chest		Steam Nozzles			
Diffuser: Inlet	Discharge	Suct. Chamber			
1st Stage Suct. Chamber Inlet (Size × Pr. Cl.00 × Facing)	x	x			
Barometric Condenser: Shell	Baffles	Nozzles			
Surface Condenser: Shell	Head				
Tubes (O.D. × BWG × L)	x	x	Material		
Tube Sheet	Baffles	Shut Off Valves			
Steam Strainer					
<b>REMARKS</b>					
Tail Pipes Furnished by					
Interconnecting Piping by					
By	Chk'd.	App.	Rev.	Rev.	Rev.
Date					
P.O. To:					

PURCHASE ORDER NUMBER

**Figure 8-29** Steam ejector specifications.

Exit Air or Non-Condensables, temperature to be 5° F higher than inlet water temperature to barometric.

$$\text{GPM cooling water required} = W_s L / (\Delta t_w 500) \quad (8-22)$$

where  $W_s$  is the pounds of steam to condense and  $L$  is the latent heat of vaporization, usually taken as 1000 Btu/lb for process applications and 950 Btu/lb for turbine exhaust steam [21, 16].

### 8.16 WATER JET EJECTORS

Ejectors using water as the motive fluid are designed for reasonable non-condensable loads together with large condensable flows. Water pressures as low as 10–20 psig are usable, while pressures of 40 psig and higher will maintain a vacuum of 1–4 in. of Hg absolute in a single-stage unit [11]. Combinations of water and steam ejectors are used to efficiently handle a wide variety of vacuum situations. The water ejector serves to condense the steam from the steam ejector.

**TABLE 8-11 Air Density Table**

Temperature (° F)	Density (lb/ft <sup>3</sup> )	Lb/h/CFM at 30" Hg abs
30	0.08105	4.86
40	0.07943	4.76
50	0.07785	4.66
60	0.07635	4.58
70	0.07493	4.50
80	0.07355	4.42
90	0.07225	4.34
100	0.07095	4.25
110	0.06966	4.18
120	0.06845	4.10
130	0.06730	4.04
140	0.06617	3.97
150	0.06510	3.91

**TABLE 8-12 Gas Constant**

Gas or Vapor	Formula	Molecular Weight	Gas Constant (R)
Hydrogen	H <sub>2</sub>	2	722
Carbon Monoxide	CO	28	55.1
Oxygen	O <sub>2</sub>	32	48.3
Methane	CH <sub>4</sub>	16	96.5
Ethylene	C <sub>2</sub> H <sub>4</sub>	28	55.1
Nitrogen	N <sub>2</sub>	28	55.1
Ammonia	NH <sub>3</sub>	17	90.8
Carbon Dioxide	CO <sub>2</sub>	44	35.1
Steam (Water vapor)	H <sub>2</sub> O	18	85.8
Sulfur Dioxide	SO <sub>2</sub>	64	24.1
Air	-	29	53.3

Note: PV = WRT      R = 1544/Molecular weight  
For Use with Units of: ft<sup>3</sup>, lb/ft<sup>2</sup>abs, °R, lb

**TABLE 8-13 Temperature–Pressure–Volume of Saturated Water Vapor over Ice**

Temperature (° F)	Absolute Pressures			Specific Volume (ft <sup>3</sup> /lb)
	in. Hg	mmHg	Microns	
32	0.1803	4.580	4580	3, 306
30	0.1645	4.178	4178	3, 609
25	0.1303	3.310	3310	4, 508
20	0.1028	2.611	2611	5, 658
15	0.0806	2.047	2047	7, 140
10	0.0629	1.598	1598	9, 050
5	0.0489	1.242	1242	11, 530
0	0.0377	0.958	958	14, 770
-5	0.0289	0.734	734	19, 040
-10	0.0220	0.559	559	24, 670
-15	0.0167	0.424	424	32, 100
-20	0.0126	0.320	320	42, 200
-25	0.0094	0.239	239	55, 800
-30	0.0071	0.180	180	74, 100
-35	0.0051	0.130	130	99, 300
-40	0.0038	0.097	97	133, 900

Values obtained from Keenan and Keyes, "Thermodynamic Properties of Steam", John Wiley & Sons, 1936, by permission.

**EXAMPLE 8-12****Temperatures at Barometric Condenser on Ejector System**

A barometric condenser is to condense 8500 lb/h of steam at 3.5 in. Hg abs using 87°F water. The non-condensables are 43 lb/h. Note that the non-condensables are less than 1% of the steam.

Steam temperature (steam tables) at 3.5 in. Hg abs	120.6°F
Temperature difference	5
Outlet water temperature from barometric	115.6°F
Inlet water temperature	85
Water temperature rise	30.6°F

$$\text{Air temperature leaving barometric} = 85 + 5 = 90^\circ\text{F}$$

$$\text{GPM cooling water required} = \frac{8500(1000)}{(30.6)(500)} = 556$$

Water ejectors and water jet eductors are also used for mixing liquids, lifting liquids, and pumping and mixing suspended solids and slurries. Sizes range from 1/2 to 24 in. The ejectors are usually used in pumping air or gases while the eductors are used in pumping liquids.

**8.17 STEAM JET THERMOCOMPRESSORS**

Steam jet thermocompressors or steam boosters are used to boost or raise the pressure of low pressure steam to a pressure intermediate between this and the pressure of the motive high pressure steam. These are useful and economical when the steam balance allows the use of the necessary pressure levels. The reuse of exhaust steam from turbines is frequently encountered. The principle of operation is the same as for other ejectors. The position of the nozzle with respect to the diffuser is critical, and care must be taken to properly position all gaskets, and so on. The thermal efficiency is high as the only heat loss is due to radiation [22].

**8.18 EJECTOR CONTROL**

Ejectors do not respond to wide fluctuations in operating variables. Therefore, control of these systems must necessarily be through narrow ranges as contrasted to the usual control of most other equipment.

For the single-stage ejector, the motive steam flow cannot be decreased below critical flow in the diffuser [23] (Figure 8-30). Units are usually designed for stable operation at zero suction flow with the motive fluid maintaining the required volume and energy to produce the necessary diffuser velocity. This is called "shut-off" operation. A decrease in motive pressure below the stability point will cause a discontinuity in operation and an increase in suction pressure. If the motive fluid rate increases, the suction pressure increases or capacity decreases at a given pressure.

Figure 8-31 illustrates control schemes for the single-stage unit, which allows greater stability in performance. As the load changes for a fixed suction pressure, the process fluid is replaced by an artificial load (usually air; Figure 8-31, item 1) to maintain constant ejector operation. An artificial pressure drop can be imposed by valve (2), although this is not a preferred scheme. When the addition of air (1) overloads the aftercondenser, the discharge mixture can be recycled to control the pressure [20].

**TABLE 8-14 Pressure-Temperature-Volume of Saturated Steam**

Absolute Pressure (in. Hg)	Temperature (° F)	Volume (ft³/lb)	Absolute Pressure (in. Hg)	Temperature (° F)	Volume (ft³/lb)	Absolute Pressure (in. Hg)	Temperature (° F)	Volume (ft³/lb)
0.1803	32	3306	3.0	115.06	231.6	20.0	192.37	39.07
0.20	34.57	2996.0	3.1	116.22	224.5	21.0	194.68	37.32
0.25	40.23	2423.7	3.2	117.35	217.9	22.0	196.90	35.73
0.30	44.96	2039.4	3.3	118.44	211.8	23.0	199.03	34.28
0.35	49.06	1761.0	3.4	119.51	205.9	24.0	201.09	32.94
0.40	52.64	1552.8	3.5	120.56	200.3	25.0	203.08	31.70
0.45	55.89	1387.7	3.6	121.57	195.1	26.0	205.00	30.56
0.50	58.80	1256.4	3.7	122.57	190.1	27.0	206.87	29.50
0.60	63.96	1057.1	3.8	123.53	185.5	28.0	208.67	28.52
0.70	68.41	913.8	3.9	124.49	181.0	29.0	210.43	27.60
0.80	72.32	805.7	4.0	125.43	176.7	29.922	212	26.80
0.90	75.84	720.8	4.5	129.78	158.2	30	212.13	26.74
						(lb/in. <sup>2</sup> abs)		
1.00	79.03	652.3	5.0	133.76	143.25	14.696	212	26.80
1.10	81.96	596.0	5.5	137.41	131.00	15	213.03	26.29
1.20	84.64	549.5	6.0	140.78	120.72	20	227.96	20.089
1.30	87.17	509.1	6.5	143.92	112.00	30	250.33	13.746
1.40	89.51	474.9	7.0	146.86	104.46	40	267.25	10.498
1.50	91.72	444.9	7.5	149.63	97.92	50	281.01	8.515
1.60	93.81	418.5	8.0	152.24	92.16	60	292.71	7.175
1.70	95.78	395.3	8.5	154.72	87.08	70	302.92	6.206
1.80	97.65	374.7	9.0	157.09	82.52	80	312.03	5.472
1.90	99.43	356.2	9.5	159.48	78.48	90	320.27	4.896
2.00	101.14	339.2	10.0	161.49	74.76	100	327.81	4.432
2.10	102.77	324.0	11.0	165.54	68.38	125	344.33	3.587
2.20	104.33	310.3	12.0	169.28	63.03	150	358.42	3.015
2.30	105.85	297.4	13.0	172.78	58.47	175	370.75	2.602
2.40	107.30	285.8	14.0	176.05	54.55	200	381.79	2.288
2.50	108.71	274.9	15.0	179.14	51.14	225	391.79	2.0422
2.60	110.06	265.0	16.0	182.05	48.14	250	400.95	1.8438
2.70	111.37	255.7	17.0	184.82	45.48	275	409.43	1.6804
2.80	112.63	247.2	18.0	187.45	43.11	300	417.33	1.5433
2.90	113.86	239.1	19.0	189.96	40.99	350	431.72	1.3260
						400	444.59	1.1613

Values obtained directly or by interpolation from Keenan and Keyes, "Thermodynamic Properties of Steam", John Wiley & Sons, 1936, by permission from and courtesy of C.H. Wheeler Co., Philadelphia, PA.

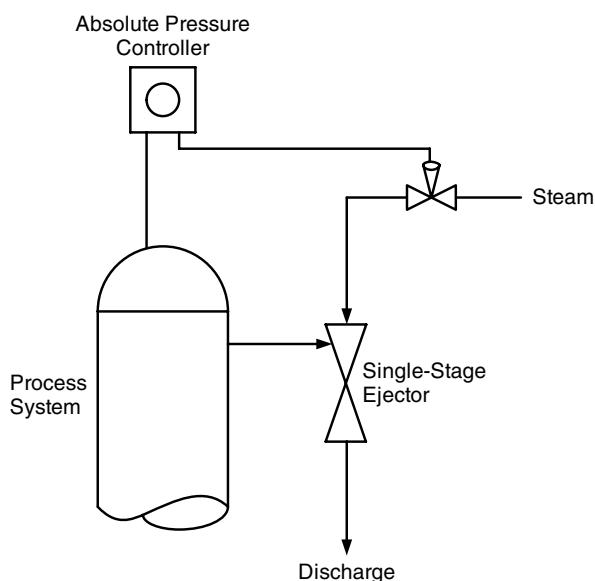


Figure 8-30 Single-stage ejector control.

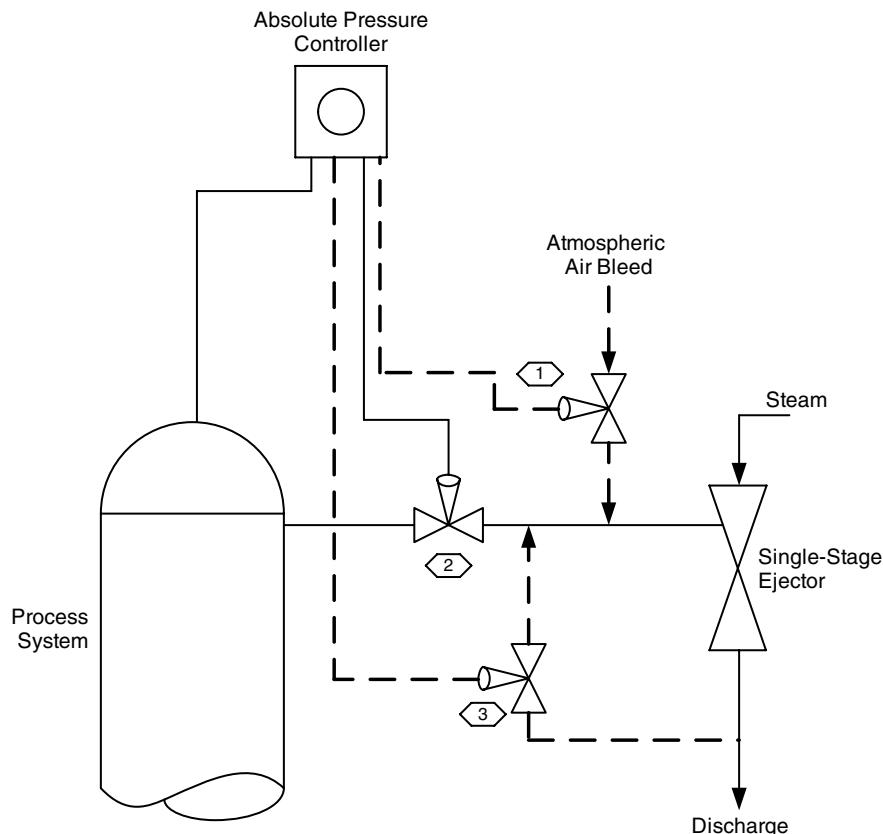
Figure 8-32 illustrates ejector or systems with large condensable loads which can be at least partially handled in the pre-condenser. Controls are used to maintain constant suction pressure at varying loads (air bleed), or to reduce the required cooling water at low process loads or low water temperatures [23]. The cooler water must not be throttled below the minimum (usually 30–50% of maximum) for proper contact in the condenser. It may be controlled by tail water temperature, or by the absolute pressure.

The controls for larger systems involve about the same principles unless special performance is under consideration.

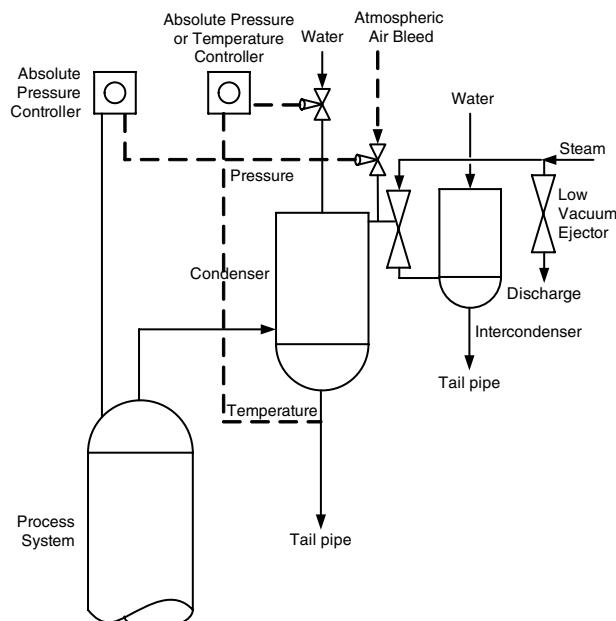
### 8.19 TIME REQUIRED FOR SYSTEM EVACUATION

It is difficult to determine the time required to evacuate any particular vessel or process system including piping down to a particular pressure level below atmospheric. When using a constant displacement vacuum pump, this is estimated by O'Neil [24]:

$$P_d = \frac{V}{t} \log_e \frac{P_2 - P_c}{P_1 - P_c} \quad (8-23)$$



**Figure 8-31** Single-stage ejector control with varying load.



**Figure 8-32** System handling large quantities of condensable vapors.

where

$P_d$  = piston displacement,  $\text{ft}^3/\text{min}$   
 $V$  = system volume,  $\text{ft}^3$

$t$  = evacuation time, min

$P_2$  = absolute discharge pressure of pump, psia

$P_c$  = absolute intake pressure of pump with closed intake

$P_1$  = absolute intake pressure of pump.

The relation above is theoretical, and does not take into account any inleakage while pumping. It is recommended that a liberal multiplier of perhaps 2 or 3 be used to estimate closer-to-actual time requirements.

An alternate relation for calculating evacuation time is from the Heat Exchange Institute [7]:

$$W_{Ta} = \frac{0.15v}{t} (P_2 - P_1) \text{ lb/h} \quad (8-24)$$

## 8.20 ALTERNATE PUMPDOWN TO A VACUUM USING A MECHANICAL PUMP

For large process system of vessels, piping, and other equipment, the downtime required to evacuate the system before it is at the pressure (vacuum) level and then to maintain its desired vacuum condition can become an important consideration during start-up, repair, and restart operations.

Reference [19] suggests an improved calculation:

$$t_s = \frac{\ln(P''_n/P''_o)}{R_{ps} \ln(V'/[V' + V_o])} \quad (8-25)$$

where

- $t_s$  = pumpdown time, s
- $P''_n$  = final pressure in vessel or system, torr
- $P''_o$  = starting pressure in vessel or system, torr
- $R_{ps}$  = pump speed, rotations (or strokes)/s
- $V'$  = volume of vessel or system, l
- $V'_o$  = volume of pump chamber, l
- $S$  = pump speed, l/s
- $S_o$  = pump speed at  $P''_o$ , l/s
- $S_n$  = pump speed at  $P''_n$ , l/s.

### EXAMPLE 8-13 Determine Pump Downtime for a System

Calculate the pump downtime for a system of vessels and piping with a volume of 500 L. The final pressure is to be 0.01 torr, starting at atmospheric. From the speed pressure curve of a manufacturer's pump at 0.01 torr, speed is 2.0 l/s. At atmospheric pressure,  $S_o = 2.75$  l/s with  $P''_o = 760$  torr. From the manufacturer's data,  $R_{ps} = 15$  and  $V'_o = 0.5$  l.

Solving Eq. (8-25):

$$t = \frac{\ln\left(\frac{0.01}{760}\right)}{(15) \ln\left(\frac{500}{500 + 0.5}\right)}$$

$$t = \frac{\ln(0.0000131579)}{(15) \ln(0.999000999)} = \frac{-11.23841862}{15(-0.0009995003)}$$

$$t = 749.6 \text{ s} = 12.49 \text{ min}$$

## 8.21 EVALUATION WITH STEAM JETS

### ROUGH ESTIMATE OF SYSTEM PUMPDOWN USING STEAM JETS [1]

The remarks presented earlier regarding the use of steam jets for pumping down a system apply. The method of power [25] presented by Reference [1] is:

$$t = [2.3 - 0.003(P'_s)] V/w_j \quad (8-26)$$

where

- $t$  = time required to evacuate a system from atmospheric pressure to the steady state operating pressure, min
- $P'_s$  = design suction pressure of ejector, torr
- $V$  = free volume of the process system, ft<sup>3</sup>
- $w_j$  = ejector capacity, 70° F dry air basis, lb/h

This assumes dry air with no condensables and negligible pressure drop through the system to the ejector. Also, the jet air handling capacity is assumed approximately twice the design capacity, and air inleakage during evacuation is negligible.

When considering time to evacuate a system using a steam jet, first recognize that securing reasonable accuracy is even more difficult than for a positive displacement pump. The efficiency of the ejector varies over its operating range; therefore, as the differential pressure across the unit varies, so will the volume handled. Consequently, evacuation time is difficult to establish except in

broad ranges. The above relations can be adopted to establish the order of magnitude only.

A recommended evacuation calculation is given in Reference [26]. This is specific to Penberthy equipment, but is considered somewhat typical of other manufacturers.

- Establish suction load of air to be evacuated in cubic feet volume of vessel/system.
- Establish the required time to evacuate, in min.
- Determine operating steam pressure, psig.
- Determine/establish required final suction pressure in vessel/system, in. Hg abs.
- Establish discharge pressure required (usually to atmosphere), psig.
- Note that the performance is specific to the ejector used.

### EXAMPLE 8-14

#### Evacuation of Vessel Using Steam Jet for Pumping Gases

The performance and procedure use the data of Penberthy for this illustration (by permission).

*Evacuation-Selection Procedure* (refer to U Evacuation Time chart)

**Step 1** Determine evacuation time in minutes per hundred cubic feet.

**Step 2** Go to the left-hand column in Table 8-15, final Suction Pressure (hs). Read across to find evacuation time equal to or less than that determined in Step 1. Read to the top of table and note unit number (see Table 8-15).

**Step 3** Read the steam consumption of unit selected off Capacity Factor Chart (see Table 8-16).

#### Evacuation – Example

To evacuate 3000 ft<sup>3</sup> vessel full of air at atmospheric pressure:

Operating Steam Pressure, psig (hm)	100
Final Suction Pressure, in. Hg abs (hs)	5
Time to evacuate, h	2.5
Discharge Pressure (hd)	Atmosphere

**Step 1** Determine evacuation time in minutes per hundred cubic feet.

$$\frac{2.5 \text{ h} \times 60}{30(\text{hundred}) \text{ ft}^3} = 5 \text{ min}/100 \text{ ft}^3$$

**Step 2** Go to the final pressure on left of Evacuation Time chart (5 in. Hg hs). Read across and find evacuation time equal to or less than 5 min. (see Table 8-15). The U-2 will evacuate the tank in 5.33 min per hundred cubic feet and the U-3 will complete the evacuation in 3.42 min per hundred cubic feet.

**Step 3** Read the steam consumption of selected unit off Capacity Factor Chart. See Table 8-16. The unit to select would be the U-3 in this case and its steam consumption is 195 lb/h.

There are often useful operations performed by jet equipment, such as pumping air or gases, exhausting systems, heating liquids, mixing of liquids, priming (removal of air) for centrifugal pumps, and many others (see Figures 8-9b and 8-10).

**TABLE 8-15 Example Using Penberthy Model U Ejector for Evaluation Time**

Suction Press in. Hg abs (lbs)	U Model Evaluation Time (in mi/100 ft <sup>3</sup> at 100 psig Operating Steam Pressure)															
	Model Number															
U-1H	U-2H	U-3H	U-4H	U-5H	U-6H	U-7H	U-8H	U-9H	U-10H	U-11H	U-12H	U-13H	U-14H	U-15H	U-16H	
12"	4.68	3.08	1.98	1.37	1.01	0.769	0.610	0.494	0.409	0.343	0.293	0.253	0.206	0.171	0.145	0.123
11"	5.06	3.32	2.14	1.48	1.09	0.830	0.657	0.532	0.441	0.370	0.316	0.273	0.222	0.185	0.156	0.133
10"	5.44	3.57	2.30	1.59	1.17	0.894	0.707	0.572	0.474	0.398	0.340	0.293	0.239	0.198	0.168	0.143
9"	5.85	3.84	2.46	1.71	1.26	0.960	0.760	0.615	0.510	0.427	0.365	0.315	0.257	0.213	0.180	0.154
8"	6.29	4.14	2.66	1.84	1.35	1.04	0.818	0.662	0.549	0.460	0.393	0.339	0.276	0.230	0.194	0.165
7"	6.76	4.45	2.86	1.98	1.46	1.12	0.880	0.771	0.590	0.495	0.423	0.365	0.297	0.247	0.209	0.178
6"	7.35	4.84	3.10	2.15	1.58	1.21	0.955	0.774	0.640	0.537	0.460	0.396	0.323	0.268	0.227	0.193
5"	8.10	5.33	3.42	2.37	1.74	1.33	1.06	0.853	0.706	0.592	0.507	0.437	0.356	0.295	0.250	0.213
4"	9.32	6.13	3.94	2.73	2.01	1.54	1.22	0.981	0.813	0.683	0.584	0.504	0.410	0.340	0.288	0.245
3"	11.6	7.60	4.87	3.38	2.48	1.90	1.50	1.22	1.01	0.845	0.721	0.623	0.507	0.422	0.356	0.304

By permission from Penberthy Inc.

**TABLE 8-16 Example Ejector Capacity Factor and Steam Consumption**

U and L Capacity Factor and Steam Consumption																
Model Number Capacity	L-1H U-1H	L-2H U-2H	L-3H U-3H	L-4H U-4H	L-5H U-5H	L-6H U-6H	L-7H U-7H	L-8H U-8H	L-9H U-9H	L-10H U-10H	L-11H U-11H	L-12H U-12H	L-13H U-13H	L-14H U-14H	L-15H U-15H	L-16H U-16H
Factor operating steam	0.293	0.445	0.694	1.00	1.36	1.78	2.25	2.78	3.36	4.0	4.69	5.43	6.66	8.03	9.49	11.12
Consumption (lb/h) (Qm)	85	125	195	270	370	480	610	755	910	1090	1280	1480	1820	2190	2580	3030
(Valid at standard nozzle pressure of 80, 100, 120, 140, 160, 180, or 200 psig)																

By permission from Penberthy Inc.

## 8.22 MECHANICAL VACUUM PUMPS

The process designer or mechanical engineer in a process plant is not expected to, nor should he/she actually design a mechanical vacuum pump or steam jet, but rather he/she should be knowledgeable enough to establish the process requirements for capacity, pressure drops, and so on, and understand the operation and details of equipment available.

Mechanical vacuum pumps are eight to ten times more efficient users of energy than steam jets; although steam jets are reliable and cost less [27] (see Table 8-17).

These units are mechanical compressors but are designed to operate at low suction pressures absolute. They require special seals to prevent leakage of air or other vapors that could create suction performance problems. They also require special clearances between the housing and the pressure-producing element(s).

**TABLE 8-17 Typical Operation Range of Vacuum Generating Equipment**

Type of Equipment	Absolute Pressure Range (Discharge, then Lower or Suction)	
	mmHg abs	mmHg abs
<i>Rotary piston (liquid)</i>		
Single-stage	760	250
Two-stage	250	150
<i>Rotary lobe (two impellers)</i>		
Single-stage	760	250
Two-stage	250	75
<i>Helical rotary lobe</i>		
Single-stage	760	200
Two-stage	200	75
<i>Centrifugal</i>		
Single-stage	760	130
<i>Rotary sliding vane</i>		
Single-stage	760	75
Two-stage	75	10
<i>Reciprocating</i>		
Single-stage	760	30
Two-stage	30	10
<i>Rotary piston (oil sealed)</i>		microns, abs
Single-stage	760	60
Two-stage	60	1
<i>Rotary vane (oil sealed)</i>		
Single-stage	760	7
Two-stage	7	0.08

Compiled from various published references and manufacturer's literature.

*Note:* For ejectors, see Figures 8-9a and 8-25.

Varies with manufacturer's equipment.

Figures 8-9a and 8-10 present representative diagrams of operating ranges of vacuum pumps and ejectors.

The chapter on Compression in Volume 3 of this series presents details of several mechanical vacuum units, and this information will not be repeated here. However, more specific vacuum units and system-related data are given.

Figure 8-33 diagrams vacuum system arrangements for process systems. It is important to examine the plant economics for each system plus the performance reliability for maintaining the desired vacuum for process control.

The most used mechanical vacuum pumps or compressors are reciprocating, liquid-ring, rotary-vane, rotary blower, rotary piston, and diaphragm.

Although the thermal efficiencies of various mechanical vacuum pumps and even steam jet ejectors vary with each manufacturer's design and even size, the curves of Figure 8-34 present a reasonable relative relationship between the types of equipment. Steam jets shown are used for surface intercondensers with 70°F cooling water. For non-condensing ejectors, the efficiency would be lower.

Combinations of steam jet ejectors operating in conjunction with mechanical pumps can significantly improve the overall system efficiency, especially in the lower suction pressure torr range of 1–100 torr. They can exist beyond the range cited, but tend to fall off above 200 torr. Each system should be examined individually to determine the net result, because the specific manufacturer and the equipment size enter into the overall assessment. Some effective combinations are:

- Steam jet–liquid ring vacuum pump
- Rotary blower–liquid ring vacuum pump
- Rotary blower–rotary vane compressor
- Rotary blower–rotary piston pump

## 8.23 LIQUID RING VACUUM PUMPS/COMPRESSOR

Figure 8-35 provides a pictorial cross section of this unit type along with an actual photograph of disassembled major components.

### HOW IT WORKS: TYPICAL OF THIS CLASS OF PUMP (BY PERMISSION OF [28])

"The Nash vacuum pump or compressor has only one moving part – a balanced rotor that runs without any internal lubrication. Such simplicity is possible because all functions of mechanical pistons or vanes are performed by a rotating band of liquid compressant."

While power to keep it rotating is transmitted by the rotor, this ring of liquid tends to center itself in the cylindrical body. Rotor axis is offset from body axis. As the schematic diagram in Figure 8-34 shows, liquid compressant almost fills, then partly

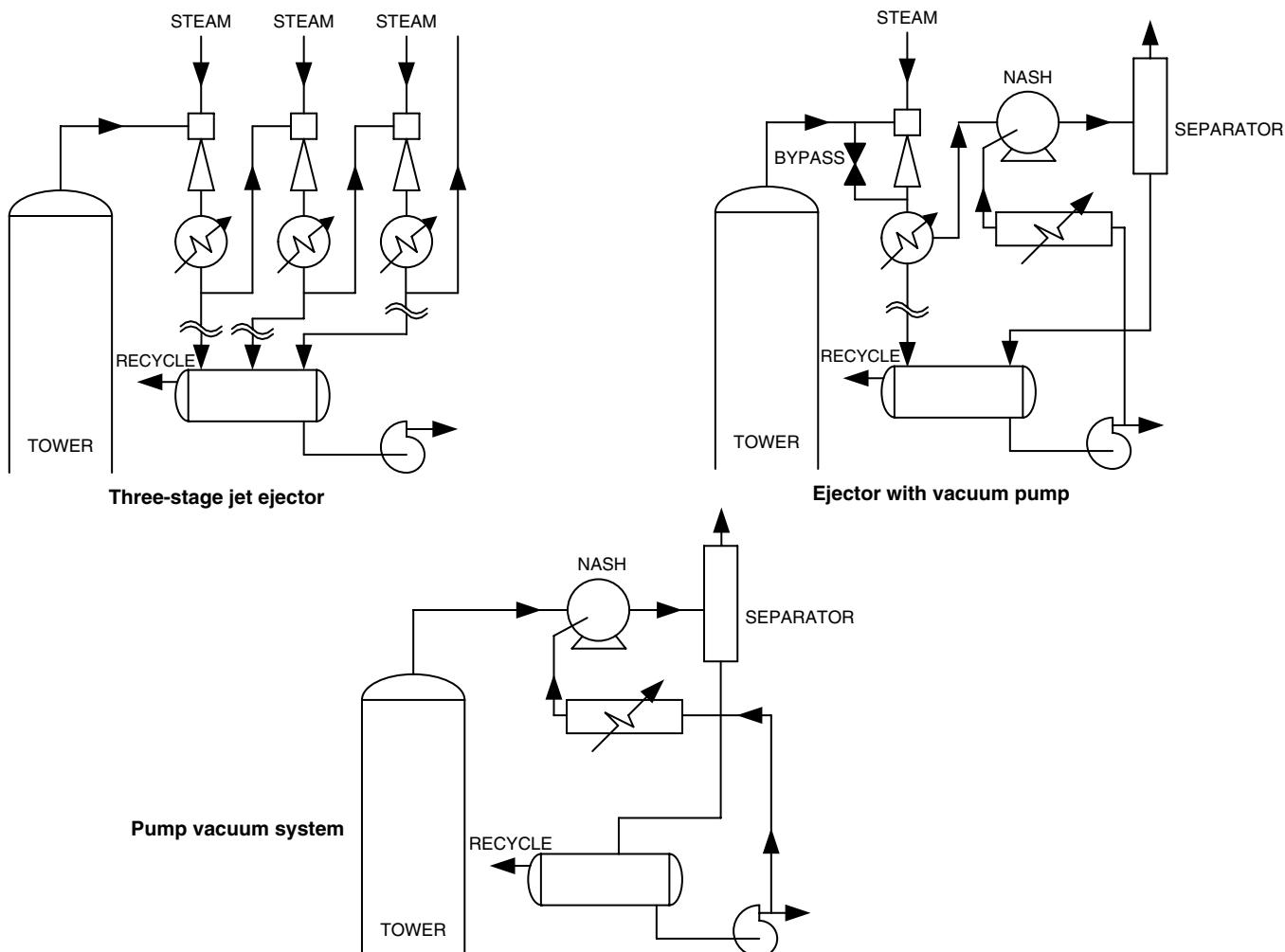


Figure 8-33 Typical vacuum systems holding vacuum on a vessel. (By permission from NASH Engineering Co.)

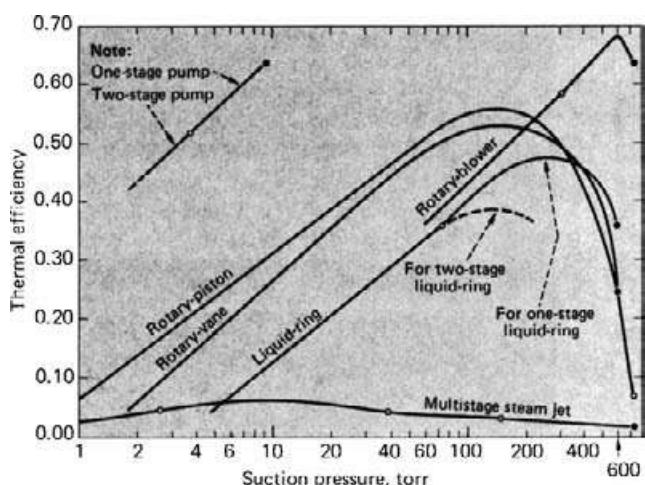


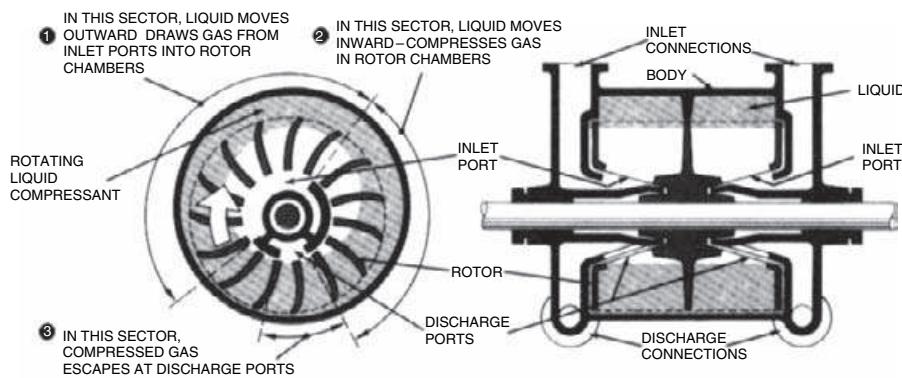
Figure 8-34 Rough estimates of thermal efficiency of various vacuum-producing systems. (By permission from Ryans, J.L. and Croll, S., *Chem. Eng.*, Vol. 88, No. 25, 1981, p. 72 [3].)

empties each rotor chamber during a single revolution which sets up the piston action. Stationary cones inside the rotor have closed sections between ported openings that separate gas inlet and discharge flows.

A portion of the liquid compressant passes out with discharged air or gas. It is usually taken out of the stream by a discharge separator furnished by the manufacturer. Make-up is regulated by orifices and manual valve adjustments. There is an optimum flow rate, but performance is not seriously affected by variations.

Moisture or even slugs of liquid entering the inlet of a liquid-ring vacuum pump will not harm it. Such liquid becomes an addition to the liquid compressant. Vapor is often condensed in a vacuum pump. The condensate is also added to the liquid compressant. In a typical closed-loop system, liquid from the separator is cooled in a heat exchanger and then re-circulated. Any excess liquid added by mist and vapor flows out through a level control valve.

The liquid ring pump/compressor is available from several manufacturers, with about the same operating principle, but differing in mechanical assembly and sealing details as well as ranges of operation. This unit type has only one moving part; that is, a balanced rotor (see Figure 8-35) that does not require



Nash vacuum pump schematic

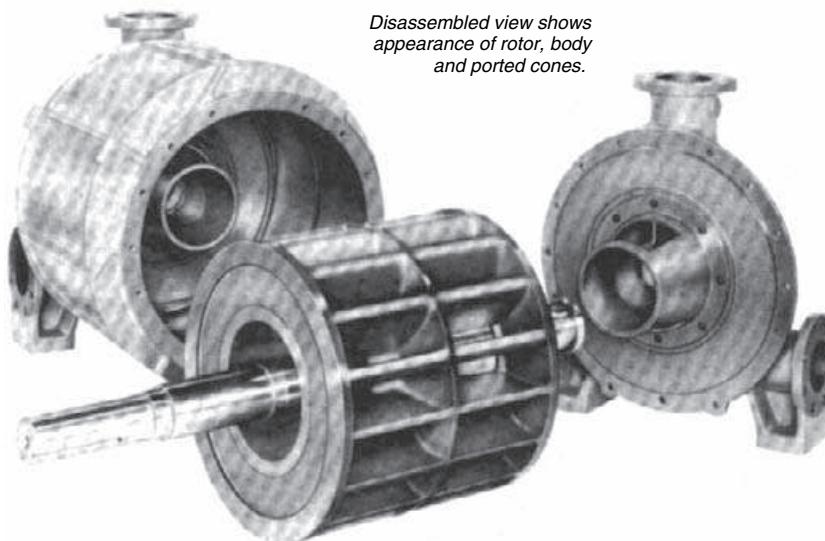


Figure 8-35 Diagram of liquid ring vacuum pump features. (By permission from Nash Engineering Co.)

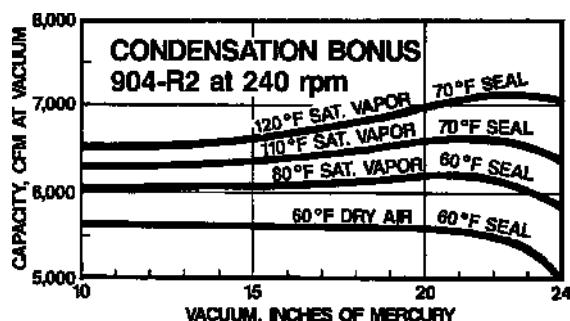


Figure 8-36 Typical capacity increases gained when a vacuum pump sealed with relatively cool water handles air saturated with water vapor. (By permission from Nash Engineering Co.)

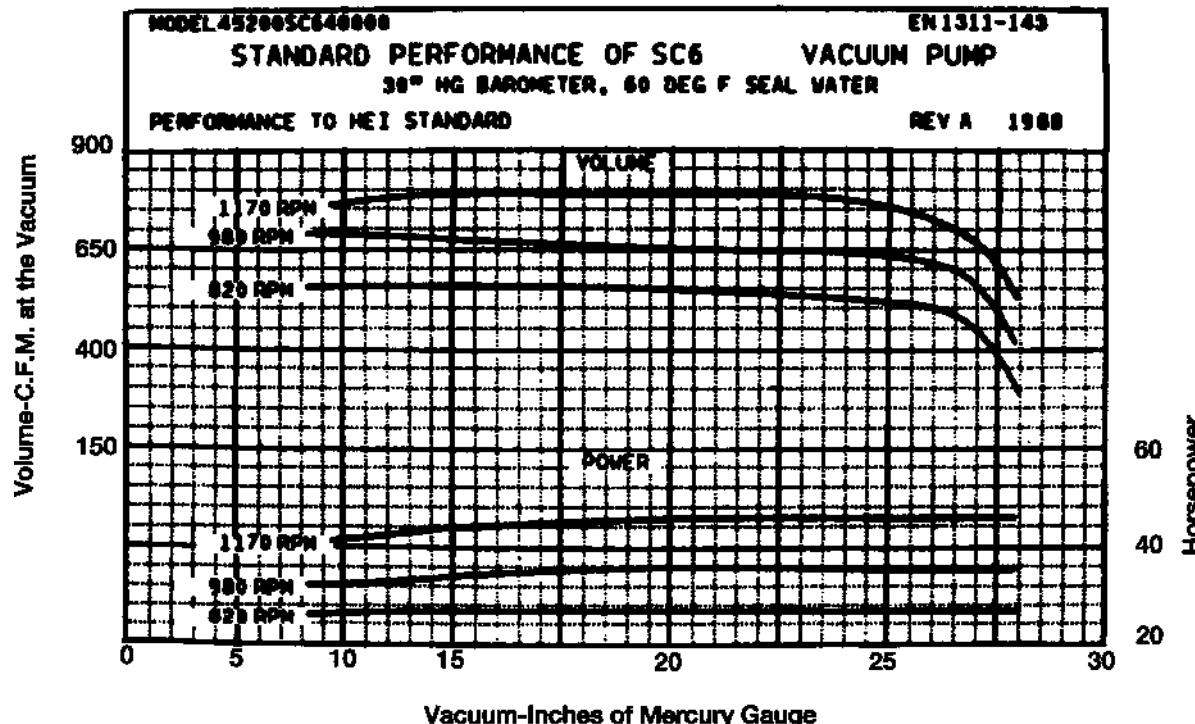
internal lubrication, because the circulating sealing liquid inside the unit provides cooling and lubrication (even when the liquid is water). There is a ring of rotating liquid in the case that is circulated by the rotor. To accomplish the pumping action, the rotor is offset from the axis of the case or body. The rotating liquid fills and empties the chamber/case of the rotor during each revolution. There are no inlet or outlet valves. The inlet and

outlet ports or openings are ported, so the rotary rotor with its stationary cones are closed between the ported openings that separate the gas inlet and discharge gas-liquid mixture to a mechanical separator.

The vapor or gas becomes separated and flows out either to the atmosphere (if air or environmentally acceptable) or a condensable vapor can be condensed inside the pump by using recirculated chilled coolant directly as the circulating liquid. The excess liquid from the condensation can be drawn off the separator.

These types of units can accept wet vapors or gases coming into the suction, as well as corrosive vapors when the proper circulating liquid compressant is selected. Some types of entrained solids can be pumped through the unit, while abrasive solids will naturally do some erosive damage. An important significance of this unit type is that the liquid used to compress the incoming vapors can be selected to be compatible with the vapors, and does not have to be water. For dry air applications, these units normally operate with 60°F seal compressing water. This keeps the water loss low. For saturated air or other vapors, the chilled recirculating liquid reduces the volume that the pump must handle and, thereby, increases pump capacity. Figure 8-36 illustrates, for a particular pump, the relative increase in expected capacity as recirculated liquid (seal) inside the pump casing vary in temperatures.

### Liquid Ring Vacuum Pump



**Figure 8-37** Typical capacity performance curve for a process liquid ring vacuum pump. Note that the vacuum is expressed here as gauge, referred to a 30 in. Hg barometer, when 60° F seal water is used. For higher temperature water, the vacuum will not be as great. (By permission from Nash Engineering Co.)

Materials of construction for this type of unit are usually modular cast iron rotors on steel shafts and cast iron casings or bodies. For special requirements in corrosive situations that cannot be remedied by changing the seal liquid, pumps can be furnished in Type 316 stainless steel or other alloys which are expensive.

Performance capacity curves are based on standard dry air with 60° F water as the liquid compressant or seal liquid. The pumps operate on a displacement or volumetric basis; therefore, the CFM capacities are about the same for any particular pump for any dry gas mixture. To calculate pounds/hours of air or gas mixture, the appropriate calculation must be made.

Figure 8-37 presents a typical performance curve for this type of vacuum pump. Note that it is specific to 60° F seal or compressant water, and that the capability to hold a vacuum with higher seal water temperature is reduced. Typically, for 95° F seal water, the vacuum may only be 26–27 in. Hg compared to the 28-in. Hg vacuum gauge shown. Also, care must be taken to recognize that the curves represent inches of mercury, vacuum gauge, not absolute. To convert, use the 30-in. Hg barometer less the vacuum reading to attain absolute vacuum, in. Hg abs. The estimated pump-down capacity performance for a typical liquid ring vacuum pump is given in Figure 8-38.

### 8.24 ROTARY VANE VACUUM PUMPS

Figure 8-39 illustrates a single-stage rotary vane vacuum pump without external cooling jacket. The sliding vanes (No. 7 in illustration) are oiled by a closed system to aid in sealing the moving vane against the casing. The rotary shaft is off-center in the casing to

provide a continuously decreasing volume from suction to discharge of the machine. Other styles of rotary vane units do not have suction or discharge valves. Note that the volume pumped is expressed as free air, which is measured at 60° F and 14.7 psia. Figure 8-40 illustrates a typical performance curve.

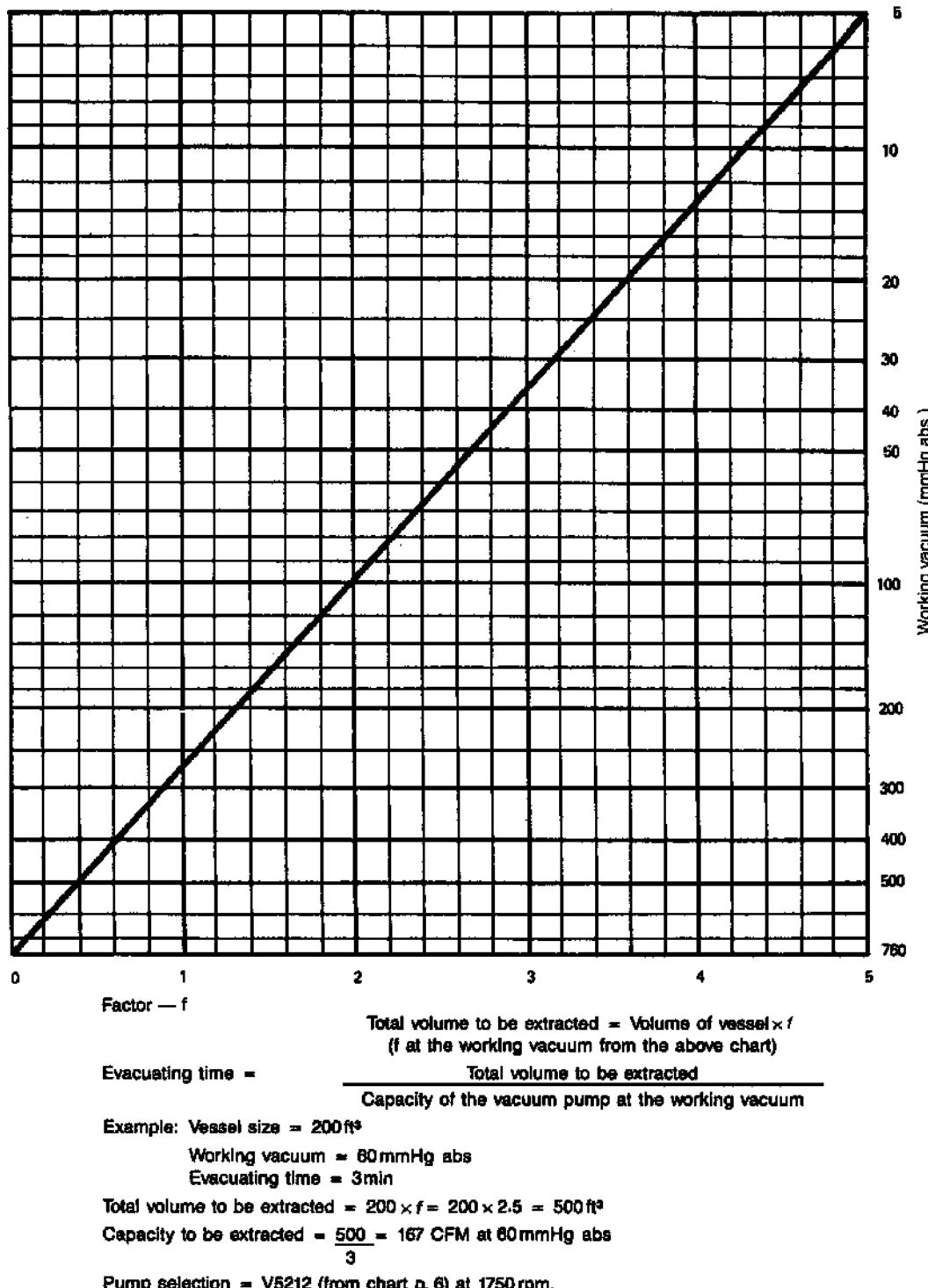
These pumps are relatively simple mechanical units compared to rotary piston pumps. The “pumping compartment vanes” are spring loaded to hold them against the off-centered position in the casing/housing. Some designs do not use springs on the vanes, but rely on centrifugal force to position the sliding vanes sealing against the casing wall. These pumps are used for medium vacuum of less than 1 torr [29]. Also see Parkinson [30].

Units without an oil pump rely on once-through oiling for vacuum sealing. The oil usage is low for most units. These units cannot handle liquid slugs or dirt particles; hence, they require gas/vapor cleaning before entering the unit. For units with cooling jackets and automatic temperature regulation of the process gas, the temperature can range as high as 302° F [29].

By adjusting the jacket temperatures, condensation or polymerization can be avoided inside the pump. The volume handled (Figure 8-40) is often expressed as “free air, CFM.” This is simply the mechanical displacement of the pumping volume of the unit for the particular driven speed of the unit.

### 8.25 ROTARY BLOWERS OR ROTARY LOBE-TYPE BLOWERS

These units are useful in both vacuum and pressure ranges and can usually develop compression ratios of 3:1 to 10:1, depending



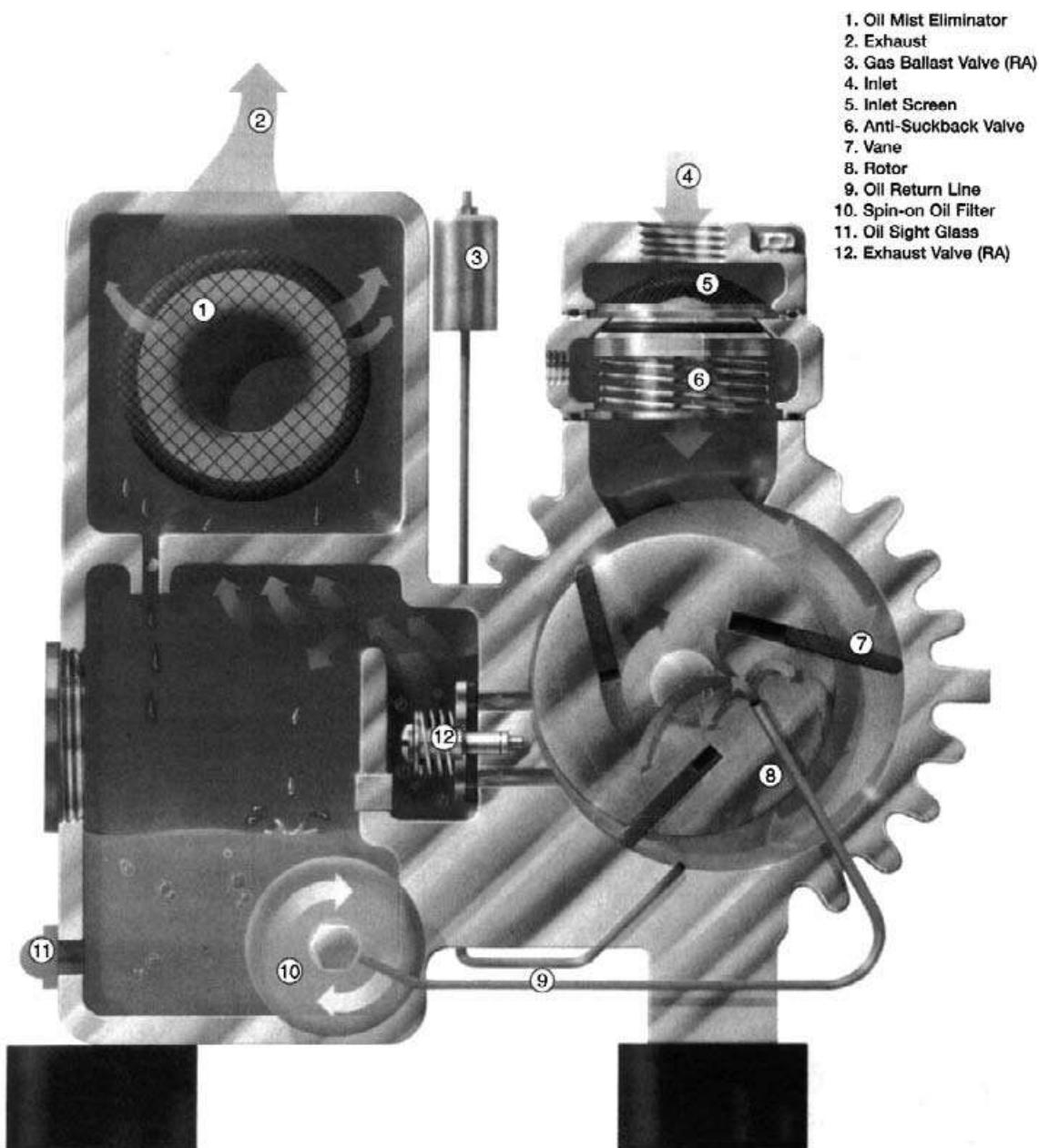
**Figure 8-38** Chart for liquid ring vacuum pump to estimate the total volume to be displaced to evacuate a closed vessel to a predetermined vacuum. (By permission from Graham Manufacturing Co., Inc.)

on the inlet absolute pressure. The units are positive displacement in performance.

The units use meshing balanced lobes (Figures 8-41–8-47) or screw-type rotors that are synchronized by timing gears and do not touch each other or the housing with very close tolerances.

The shafts of the driving shaft and the driven shaft are sealed with labyrinth-type seals for a minimum leakage. Purged labyrinth or mechanical-type seals are available.

The rotors do not require lubrication per se because they do not touch; however, some designs add a small amount of sealing



### R5 Operating Principle

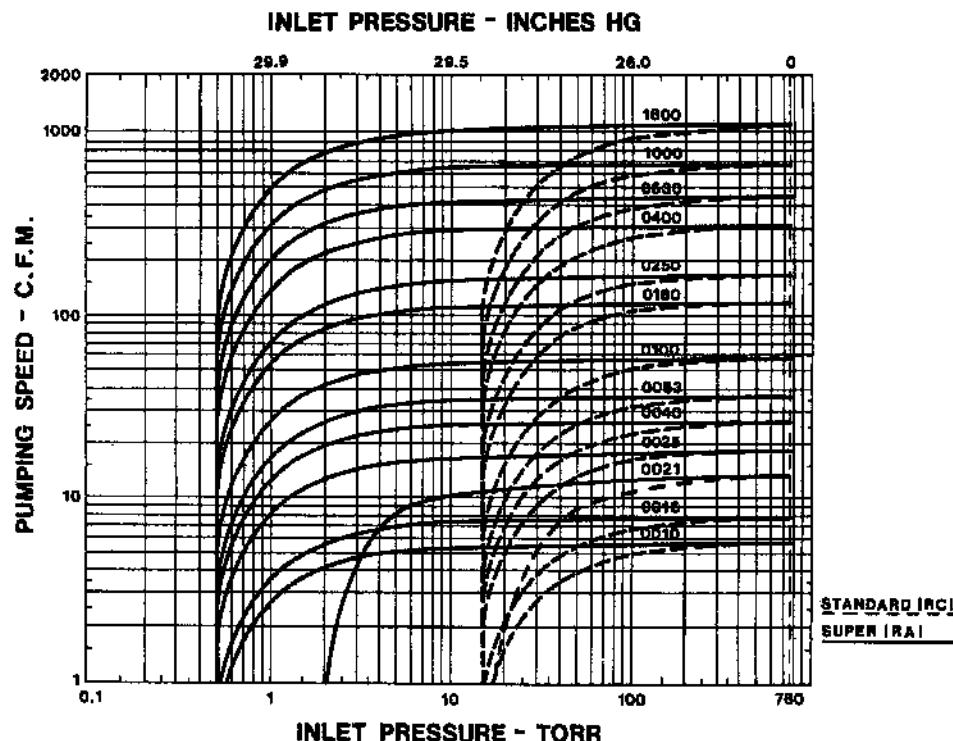
Rotation of the pump rotor, which is mounted eccentrically in the pump cylinder, traps the entering vapor between rotor vane segments. As rotation continues, vapor is compressed and then discharged into the exhaust box. Vapors then pass through four stages of internal oil and smoke eliminators to remove 99.9% of lubricating oil from the exhaust. Oil is then returned to the recirculating oil system.

The four-stage exhaust box includes the oil box separator, the demister pad, the oil mist eliminator, and the synthetic oil baffle. Additional features include an automotive type spin-on oil filter, a built-in inlet anti-suckback valve that prevents oil from being drawn into the system when the pump is stopped, and a built-in gas ballast, available on the RA version, which permits pumping with high water vapor loads.

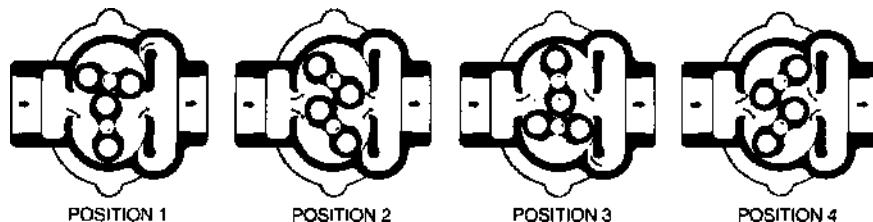
**Figure 8-39** Rotary vane-type vacuum pump without external cooling jacket. (By permission from Busch, Inc.)

liquid such as water or other compatible fluid to reduce the “slip” or backflow as the lobes rotate. Depending on the type of design, that is, lobe or screw-type cycloidal or helical rotors, the discharge pressure may have some pulsation, such as  $-60$  to  $140 \pm \%$  of gas discharge pressure or it may be essentially “smooth,” almost

pulsation free. The shafts of the rotors rotate in opposite directions by means of the drive gears. The air or process gas/vapor is drawn into the suction or inlet cavity of the intersecting and rotating “lobes” by the rotor mesh. As the rotors continue to rotate, the cavity of suction gas is sealed from the inlet by the moving and/or



**Figure 8-40** Typical performance curves for rotary vane-type vacuum pump. (By permission from Busch, Inc.)



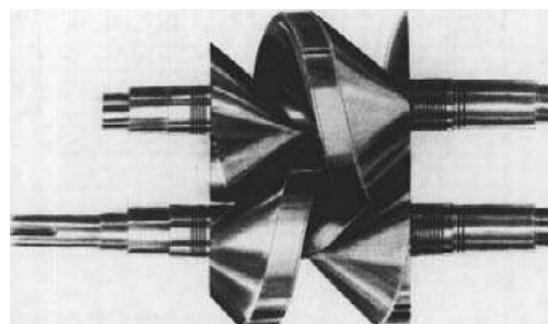
**Figure 8-41** Rotating lobe vacuum blowers showing lobe rotation as gas moves through the unit. (By permission from Roots Division, Dresser Industries, Inc.)



**Figure 8-42** Helical four-fluted gate blower rotors/two lobes (main rotor intermeshing) (see Figure 8-44). (By permission from Gardner-Denver Indus. trial Machinery-Cooper Industries.)

advancing “lobes” as they pass the fixed boundary of the inlet opening of the casing. As the rotors rotate, the gas is compressed against the discharge head plate, similar in concept to a piston compressor against a piston compressor cylinder head [31].

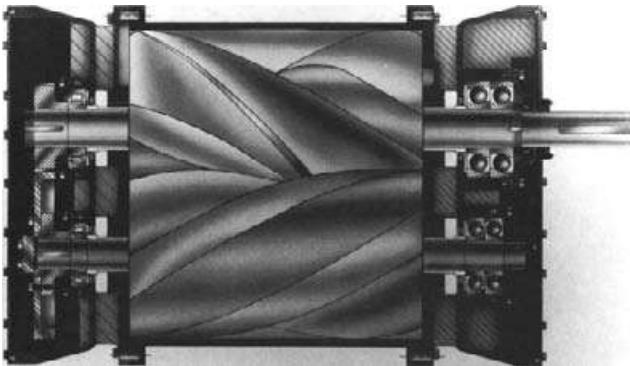
Booster vacuum pumps are used to shorten the pump down on evacuation (time) of a vacuum system before switching to the



**Figure 8-43** Screw-type rotors for rotary lobe blower. (By permission from Roots Division, Dresser Industries, Inc.)

smaller vacuum pump to maintain the system opening vacuum and to handle the air leakage to the system.

A typical performance range of capacities of rotary lobe vacuum pumps is shown in Figures 8-48a–c. Another set of curves for rotary lobe pumps, shown in Figure 8-49, provides the brake

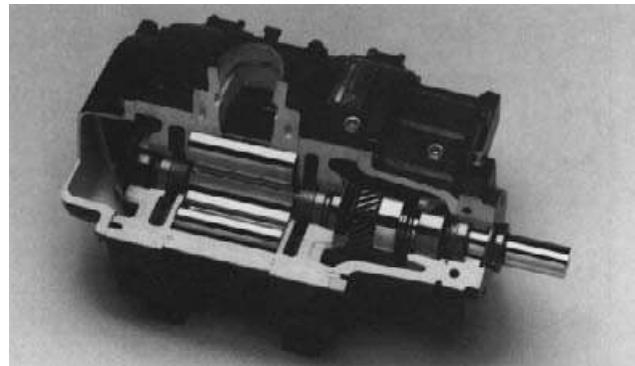


**Figure 8-44** Rotary blower with screw-type rotors. (By permission from Gardner-Denver Industrial Machinery-Cooper Industries.)

horsepower, airflow at inlet (CFM) (referred to as their standard volume at 70° F and 29.92 in. Hg abs discharge pressure – essentially atmosphere), and the temperature rise through a non-cooled (no internal or external cooling) vacuum. All data are referenced to the indicated vacuum, not absolute pressure (see the beginning of this chapter for clarification).

## 8.26 ROTARY PISTON PUMPS

These units are also positive displacement oil-sealed pumps. The pumping action is shown in Figure 8-50. The action of the rotating piston assembly draws in a set volume (depending on the pump's size/capacity) of gas, compresses it in the eccentric rotating cylinder against the inside wall of the casing, and exhausts it to the atmosphere, as do most other vacuum pumping devices. As the oil-sealed piston revolves, it opens the inlet port, draws in gas, and traps it in the fixed suction space. As the piston rotates, the gas is compressed, and the discharge valve opens to discharge the gas.

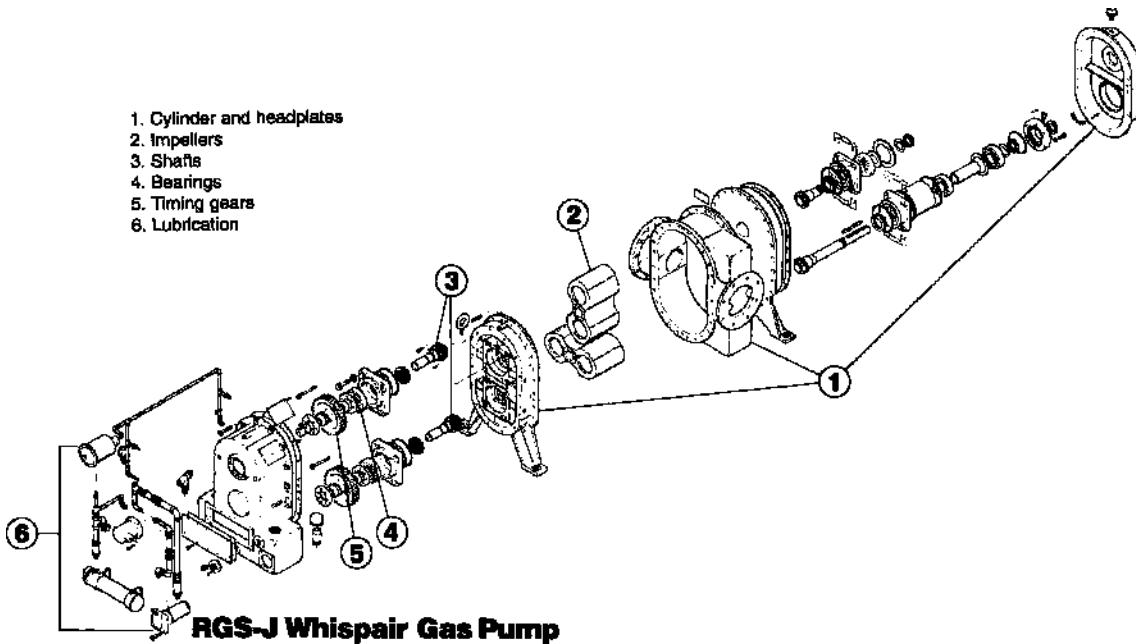


**Figure 8-45** Cut-away view of internal assembly of rotary lobe vacuum pump. (By permission from Tuthill Corp., M.D. Pneumatics Division.)

A single-stage pump can have compression ratios up to 100,000:1 when discharging to the atmosphere [3].

Mechanically, the pump operates in an oil bath, with the sealing oil lubricating the pump and seals against back flow from the exhaust to the intake/suction. These pumps cannot effectively handle condensation of vapors inside the unit, because the capacity is reduced and the condensate creates lubrication problems, which in turn leads to mechanical breakdown.

To prevent/reduce the undesirable condensation in the pump, a small hole is drilled in the pump head to admit air or other process non-condensable gas (gas ballast) into the latter portion of the compression stroke. This occurs while the vapor being compressed is sealed off from the intake port by the piston. By reducing the partial pressure of the vapor's condensables, the condensation is avoided. Obviously, this can reduce the capacity of the pump, as the leakage past the seals allows the gas ballast to dilute the intake volume of base suction gas. For most process applications, the effect of this leakage is negligible, unless the vacuum system suction is below 1 torr [3].



**Figure 8-46a** Exploded view of rotary lobe gas pump with mechanical seals and pressure lubrication for bearings. (By permission from Roots Division; Dresser Industries, Inc.)

### Detail – Mechanical seal

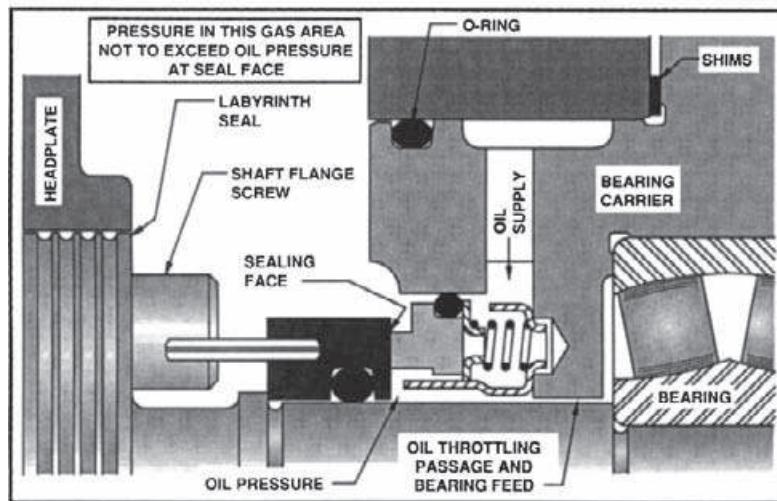
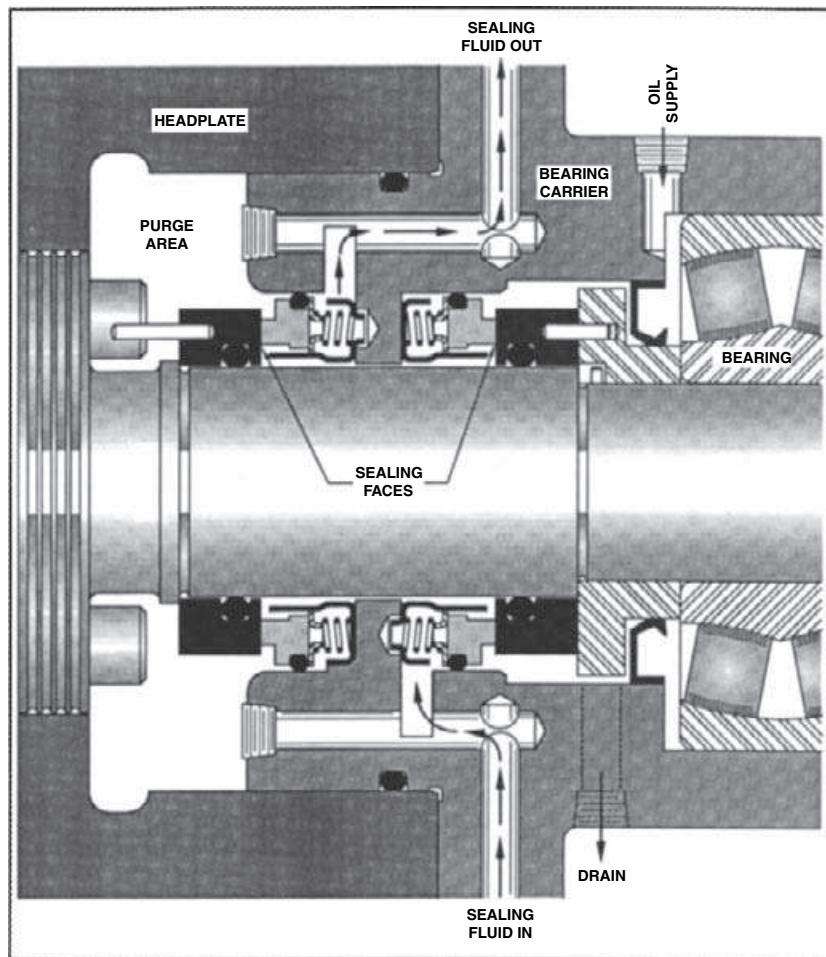
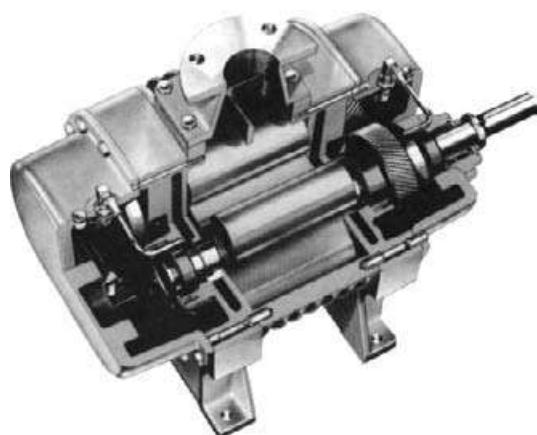


Figure 8-46b Mechanical seal to be used with Figure 8-46a. (By permission from Roots Division, Dresser Industries, Inc.)

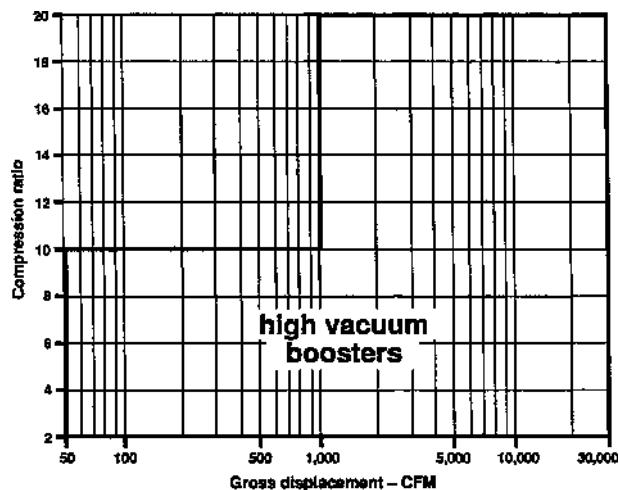


### PROCESS INDUSTRY REQUIREMENT DOUBLE MECHANICAL SEAL

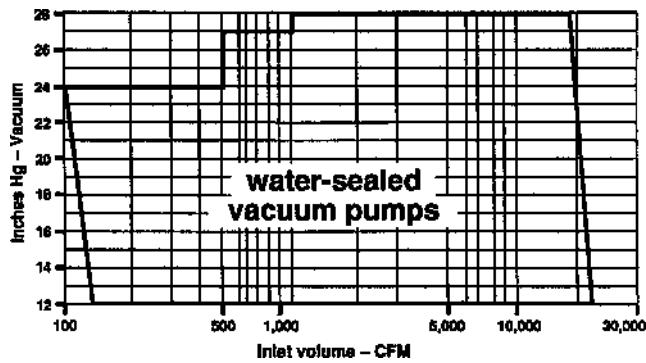
Figure 8-46c Double mechanical seal used for special gas sealing requirements in Figure 8-46a, and substitutes for the single seal of Figure 8-46. (By permission from Roots Division, Dresser Industries, Inc.)



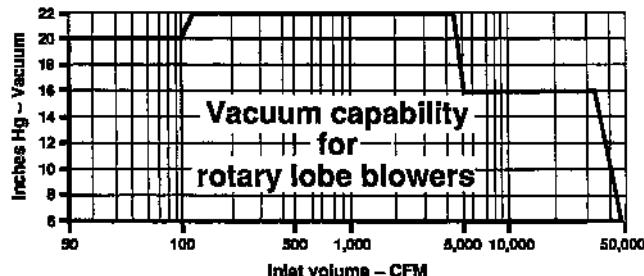
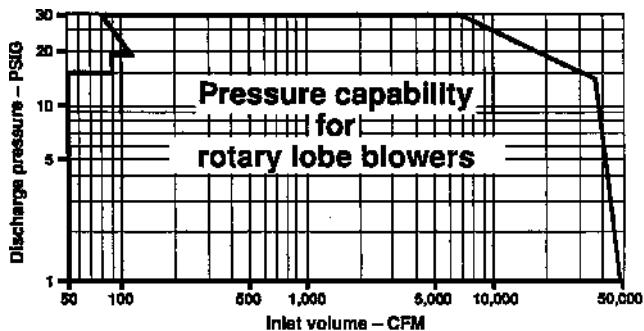
**Figure 8-47** Cut-away view of internal assembly of rotary lobe vacuum. (By permission from Tuthill Corp. M. D. Pneumatics Div., Bull. A5/888.)



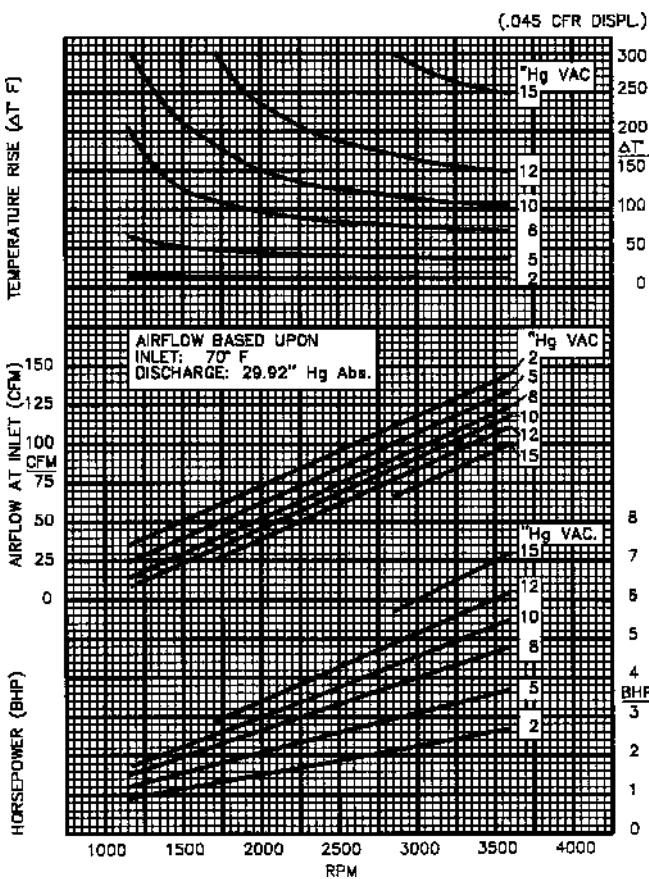
**Figure 8-48a** Typical performance of high vacuum booster lobe-type high volume draw-down for evacuating vacuum systems before use of higher vacuum (lower absolute pressure) pump. (By permission from Roots Division, Dresser Industries, Inc.)



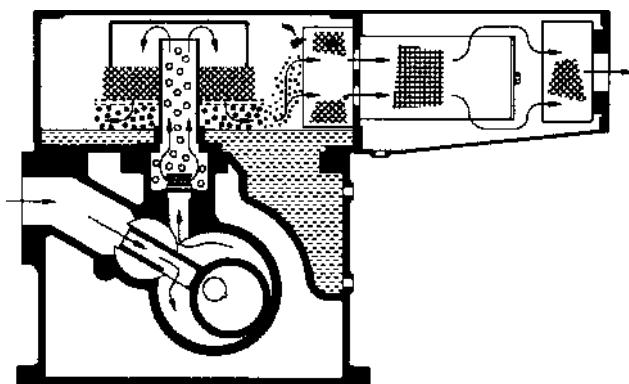
**Figure 8-48b** Performance of lobe-type vacuum pump using water spray internally to reduce slip of gases from discharge to suction. (By permission from Roots Division, Dresser Industries, Inc.)



**Figure 8-48c** Typical application rotary lobe vacuum blower performance. (By permission from Roots Division, Dresser Industries, Inc.)



**Figure 8-49** Typical performance curves for rotary lobe vacuum pump. (By permission from Tuthill Corp., M.D. Pneumatics, Division.)



**Figure 8-50** Typical rotary displacement vacuum pump, oil sealed, single-stage. (By permission, Kinney Vacuum Co.)

Huff [27] found that the reciprocating and rotary piston pumps were the most economical mechanical systems for their range of application. Obviously, the economic discussions are dependent on the vacuum expected and the local utility costs, plus the cost of maintenance.

#### NOMENCLATURE

$C_{pa}$	Specific heat of air at constant pressure (0.24 approx.)
$C_{ps}$	Specific heat of steam at constant pressure corresponding to downstream absolute pressure (0.45 approx.)
$D$	Sealed diameter, in. (estimates of nominal diameter acceptable)
$E$	Evacuation factor, at final evacuation suction pressure
$F$	Steam pressure factor
$h_L$	Liquid height, ft
$K$	Non-condensable load factor
$l$	Latent heat of vaporization of steam, Btu/lb
$M$	Average molecular weight of system vapors
$M_n$	Molecular weight of non-condensable gas
$M_v$	Molecular weight of condensable vapor
$P$	Total absolute pressure, lb/in. <sup>2</sup> absolute (or other consistent units), or system operating pressure, torr
$P_a$	Partial pressure of air in mixture, lb/in. <sup>2</sup> abs
$P_c$	Absolute intake pressure of pump
$P_d$	Piston displacement, ft <sup>3</sup> /min
$P_n$	Partial pressure of non-condensable gas, psia (or other absolute units)
$P_s$	Static pressure, atm
$P_v$	Vapor pressure of condensable vapor, psia (or other absolute units)
$P'$	Partial pressure of a vapor in a mixture, psia
$P'_s$	Design suction pressure of ejector, torr
$P''_n$	Final pressure in vessel or system, torr
$P''_o$	Starting pressure in vessel or system, torr
$P$	atmospheric pressure, mmHg
$P_1$	Intake pressure of pump, psia; or, initial pressure in system, in. Hg abs
$P_2$	Discharge pressure of pump, psia
$P_c$	Intake pressure of pump with closed intake, psia
$R$	Gas constant = 1544/mol weight
$R_{ps}$	Pump speed, revolutions (or strokes) per second
$S$	Pump speed, L/s
$S_n$	Pump speed at $P''_n$ , L/s
$S_o$	Pump speed at $P''_o$ , L/s

$T$	Temperature, °R = 460 + °F
$t$	Evacuation pump downtime, min
$t_s$	Evacuation pump downtime, s
$t_a$	Ambient air temperature, °F
$t_m$	Temperature of mixture at ejector suction, °F
$t_s$	Temperature of steam on downstream side of nozzle, °F
$\Delta t_w$	Temperature rise of water, °F
$V$	Volume of tank or system, ft <sup>3</sup>
$V'$	Volume of vessel or system, L
$V'_o$	Volume of pumping chambers, L
$W$ or $W_a$	Flow rate, lb/h
$W_n$	Weight of non-condensable gas, lb/h
$W_m$	Total pounds of mixture handled per hour
$W_s$	Steam flow rate, lb/h
$W_v$	Weight of condensable vapor, lb/h
$W_t$	Total weight of gas, lb
$W_{Ta}$ or $W_T$	Total calculated air inleakage, lb/h
$W'_a$	Air inleakage resulting from metal porosity and cracks along weld lines, lb/h
$W'_m$	Ejector capacity at final evacuation suction pressure, lb/h
$W'_s$	Pounds of motive steam per pound of mixture handled
$W'_v$	Pounds water vapor per pound air
$w$	Constant flow rate, lb/min
$w_a$	Acceptable air leakage rate assigned to a system component, lb/h
$w_j$	Ejector capacity, 70° F dry air basis, lb/h

#### GREEK SYMBOLS

$\rho$	Density, lb/ft <sup>3</sup>
$\rho_L$	Specific gravity of liquid, relative to water = 1.0
$\theta$	Specific air leakage rate, lb/h/in.
$\pi$	$\Pi = 3.1418$

#### SUBSCRIPTS

a	air
1,2, and so on	Refers to components in system, or initial and final system pressures, psia
90	Steam pressure, psia
v	Condensable vapor
n	Non-condensable gas/vapor

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## WEBSITES ON EJECTORS, VACUUM SYSTEMS, AND SCRUBBERS

1. [www.cheresources.com/ejectors.shtml](http://www.cheresources.com/ejectors.shtml)
2. [www.artisanind.com/jetvac/singlestage.htm](http://www.artisanind.com/jetvac/singlestage.htm)
3. [www.s-k.com](http://www.s-k.com)
4. [www.foxvalve.com](http://www.foxvalve.com)
5. [www.transvac.co.uk](http://www.transvac.co.uk)
6. [www.labconco.com](http://www.labconco.com)
7. [www.graham-mfg.com](http://www.graham-mfg.com)
8. [www.sourceesb.com](http://www.sourceesb.com)

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## PROCESS SAFETY AND PRESSURE-RELIEVING DEVICES

**T**he subject of process safety is so broad in scope that this chapter must be limited to the application, design, rating, and specifications for process overpressure-relieving devices for flammable vapors and dusts; process explosions, runaway chemical reactions and external fires on equipment; and the venting or flaring of emergency or excess discharge of gases to a vent flare stack. The subject of fire protection cannot be adequately covered; however, the engineer is referred to texts dealing with the subject in a thorough manner [1–6].

The possibilities for development of excess pressure exist in nearly every process plant. Due to the rapidly changing and improved data, codes, regulations, recommendations, and design methods, it is recommended that reference be made to the latest editions of the literature listed in this chapter. While attempting to be reliable in the information presented, I am not responsible or liable for interpretation or the handling of the information by experienced or inexperienced engineers. This chapter's subject matter is vital to the safety of plants' personnel and facilities.

It is important to understand how the overpressure can develop (source) and what might be the eventual results. The mere solving of a formula to obtain an orifice area is secondary to an analysis and understanding of the pressure system. Excess pressure can develop from explosion, chemical reaction, reciprocating pumps or compressors, external fire around equipment, and an endless list of related and unrelated situations. In addition to the possible injury to personnel, the loss of equipment can be serious and an economic setback.

Most states have laws specifying the requirements regarding application of pressure-relieving devices in process and steam power plants. In essentially every instance, at least part of the reference includes the A.S.M.E. Boiler and Pressure Vessel Code, Section VIII, Division 1 (Pressure Vessels) and/or Division 2 [1]; and Section VII, Recommended Rules for Care of Power Boilers [7]. In addition, the publications of the American Petroleum Institute (API) are helpful in evaluation and

design. These are API-RP-520 [8], Design and Installation of Pressure-Relieving Systems in Refineries; Part I-Design; Part II-Installation; and API-RP-521 [9], Guide for Pressure Relief and Depressurizing Systems, ANSI/ASME B31.1 Power Piping; B16.34; and NFPA 1; [10], Sections 30, 68, and 69.

The ASME code requires that all pressure vessels be protected by a pressure-relieving device that shall prevent the internal pressure from increasing more than 10% above the maximum allowable working pressures (MAWP) of the vessel to be discussed later. Except where multiple relieving devices are used, the pressure shall not increase more than 16% above the MAWP or, where additional pressure hazard is created by the vessel being exposed to external heat (not process related) or fire, supplemented pressure-relieving devices must be installed to prevent the internal pressure from rising more than 21% above the MAWP. (see [1] Sections U-125 and UG-126). The best practice in industrial design recommends that (a) all pressure vessels of any pressure be designed, fabricated, tested, and code-stamped as per the applicable ASME code [1] or API Codes and Standards [5] and (b) that pressure-relieving devices be installed for pressure relief and venting per codes [1, 5, 8 and 9]. Although not specifically recognized in the titles of the codes, the rupture disk as a relieving device is, nevertheless, included in the requirements as an acceptable device.

Usual practice is to use the term "safety valve" or "relief valve" to indicate a relieving valve for system overpressure and this will be generally followed here. When specific types of valves are significant, they will be emphasized.

This chapter reviews runaway chemical reactions in process equipment in greater details, and further presents design methodologies for sizing vents involving two-phase flow and compressible flow for discharge piping. It introduces hazard and operability (HAZOP) studies in process plant design, and inherently safer plant design in reactor system.

### 9.1 TYPES OF POSITIVE PRESSURE-RELIEVING DEVICES (SEE MANUFACTURERS' CATALOGS FOR DESIGN DETAILS)

**Relief Valve:** A relief valve is an automatic spring-loaded pressure-relieving device actuated by the static pressure upstream of the valve, and which opens further with increase in pressure over the opening pressure. It is used primarily for liquid service [1, 8] (Figures 9-1a and b). The rated capacity is usually attained at 25% overpressure.

**Safety Valve:** This is an automatic pressure-relieving device actuated by the static pressure upstream of the valve and

characterized by rapid full opening or "pop" action upon opening [1, 8], but does not reseat. It is used for steam or air service (Figure 9-2). The rated capacity is reached at 3, 10, or 20% overpressure, depending upon applicable code.

**Safety relief Valve:** This is an automatic pressure-relieving device actuated by the static pressure upstream of the valve and characterized by an adjustment to allow reclosure, either a "pop" or a "non-pop" action, and a nozzle type entrance; and it reseats as pressure drops. It is used on steam, gas, vapor, and liquid (with adjustments), and is probably the most general type of valve in petrochemical and chemical plants (Figures 9-3a, b, and 9-4). The

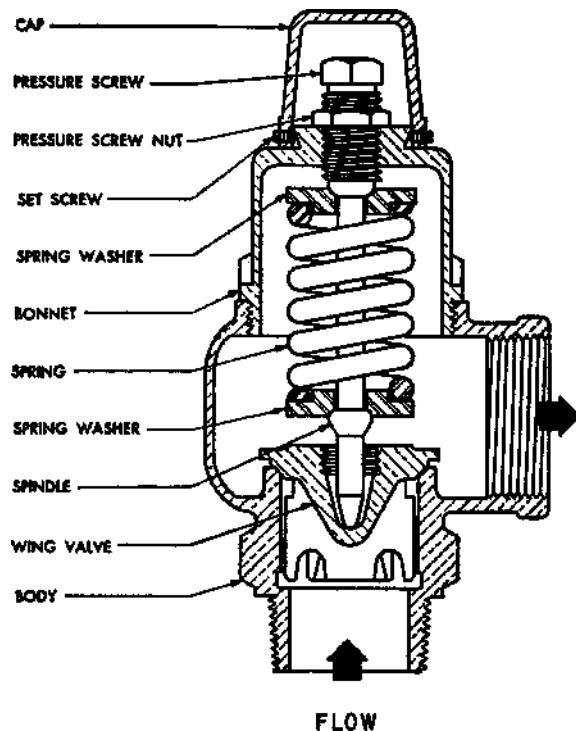


Figure 9-1a Relief valve. (Courtesy of Crosby-Ashton Valve Co.)

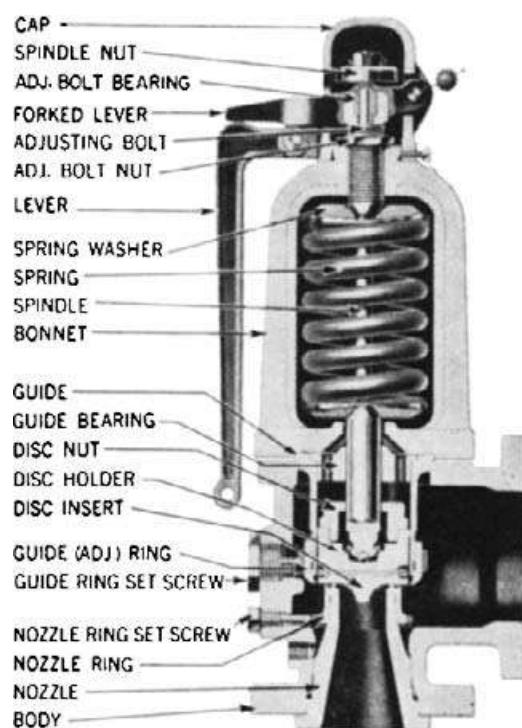


Figure 9-2 Safety valve. (Courtesy of Crosby-Ashton Valve Co.)

#### Pressure-Relieving Devices

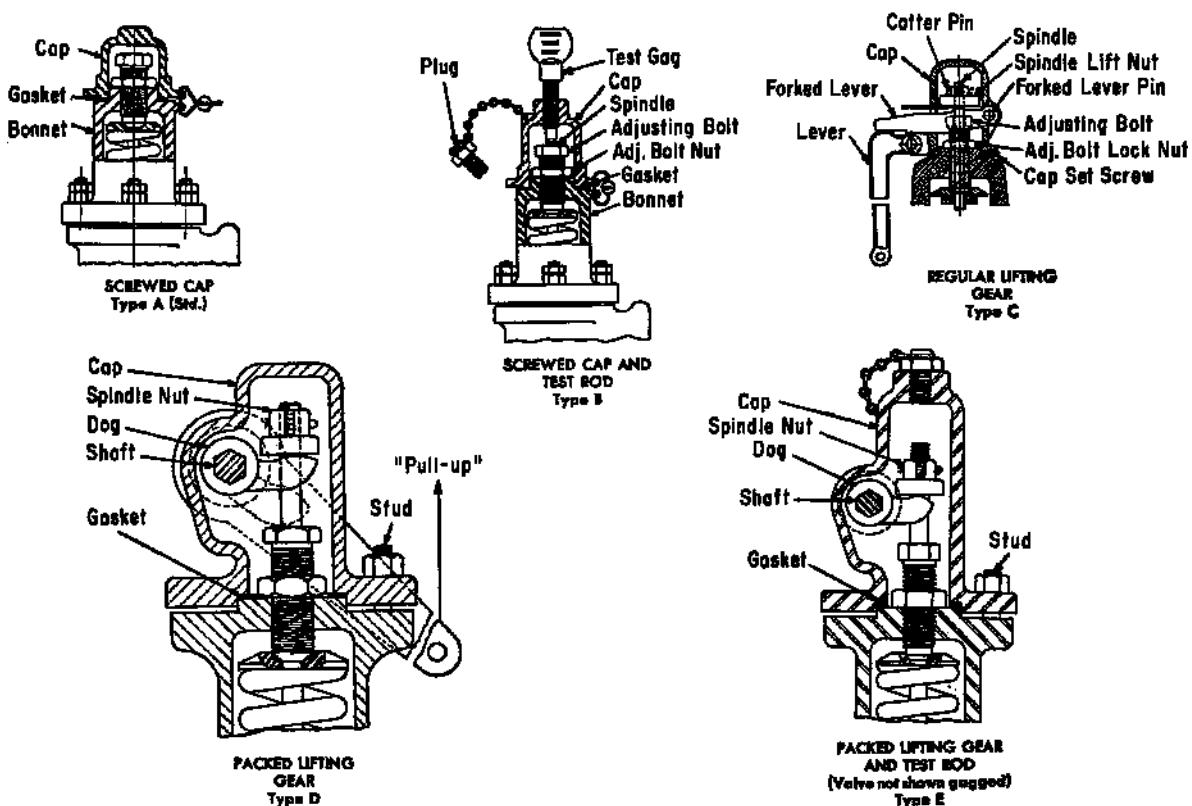
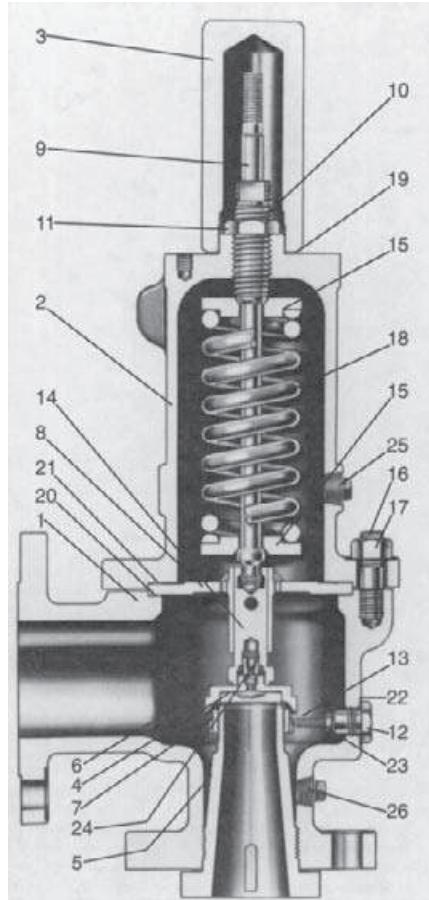


Figure 9-1b Accessories for all types of safety relieving valves. (Courtesy of Crosby-Ashton Valve Co.)



**Figure 9-3a** Conventional or unbalanced nozzle safety relief valve. (By permission from Teledyne Farris Engineering Co.)

rated capacity is reached at 3 or 10% overpressure, depending upon code and/or process conditions. It is suitable for use either as a safety or as a relief valve [1, 8]. It opens in proportion to an increase in internal pressure.

**Pressure-relief Valve:** The term "Pressure-relief valve" applies to relief valves, safety valves, or safety relief valves [8].

**Pilot-operated Safety Valves:** When properly designed, this type of valve arrangement conforms to the ASME code. It is a pilot-operated pressure-relief valve in which the major relieving device is combined with and is controlled by a self-activating auxiliary pressure-relief valve (see Figures 9-5a and b).

## 9.2 TYPES OF VALVES/RELIEF DEVICES

There are many design features and styles of safety relief valves, such as flanged ends, screwed ends, valves fitted internally for corrosive service, high temperature service, cryogenic service/low temperatures, with bonnet or without, nozzle entrance or orifice entrance, and resistance to discharge piping strains on body. Yet most of these variations have little, if anything to do with the actual performance to relieve overpressure in a system/vessel.

A few designs are important to the system arrangement and relief performance. They are as follows.

### CONVENTIONAL SAFETY RELIEF VALVE

This valve design has the spring housing vented to the discharge side of the valve. The performance of the valve upon relieving

### Bill of Materials-Conventional

ITEM	PART NAME	MATERIAL
1	Body	SA-216 GR. WC6, Carbon Steel 26( )A10 thru 26( )A26 26( )A32 thru 26( )A36
	Bonnet	SA-216 GR. WC6, Alloy St. (1½ CR-½ Moly) 26( )A10 thru 26( )A26 26( )A32 thru 26( )A36
3	Cap, Plain Screwed	Carbon Steel
	4 Disc	Stainless Steel
	5 Nozzle	316 St. St.
	6 Disc Holder	300 Series St. St.
	7 Blow Down Ring	300 Series St. St.
	8 Sleeve Guide	300 Series St. St.
	9 Stem	Stainless Steel
	10 Spring Adjusting Screw	Stainless Steel
	11 Jam Nut (Sp. Adj. Scr.)	Stainless Steel
	12 Lock Screw (R.D.R)	Stainless Steel
	13 Lock Screw Stud	Stainless Steel
	14 Stern Retainer	Stainless Steel
	15 Spring Button	Carbon Steel Rust Proofed
	16 Body Stud	ASTM A193 Gr. B7, Alloy St.
	17 Hex Nut (Body)	ASTM A194 Gr. 2H, Alloy St.
	18 Spring	Carbon Steel Rust Proofed 26( )A10 thru 26( )A16 26( )A20 thru 26( )A36
	19 Cap Gasket	Soft Iron or Steel
	20 Body Gasket	Soft Iron or Steel
	21 Bonnet Gasket	Soft Iron or Steel
	22 Lock Screw Gasket	Soft Iron or Steel
	23 Hex Nut (B.D.R.L.S.)	Stainless Steel
	24 Lock Screw (D.H.)	Stainless Steel
	25 Pipe Plug (Bonnet)	Steel
	26 Pipe Plug (Body)	Steel

Also suitable for liquid service where ASME Code certification is not required.

overpressures is directly affected by any changes in the back pressure on the valve (opening pressure, closing pressure, and relieving capacity referenced to opening pressure) (see Figures 9-3, 9-6a, and 9-6b) [11]. When connected to a multiple relief valve manifold, the performance of the valve can be somewhat unpredictable from a relieving capacity standpoint due to the varying back pressure in the system.

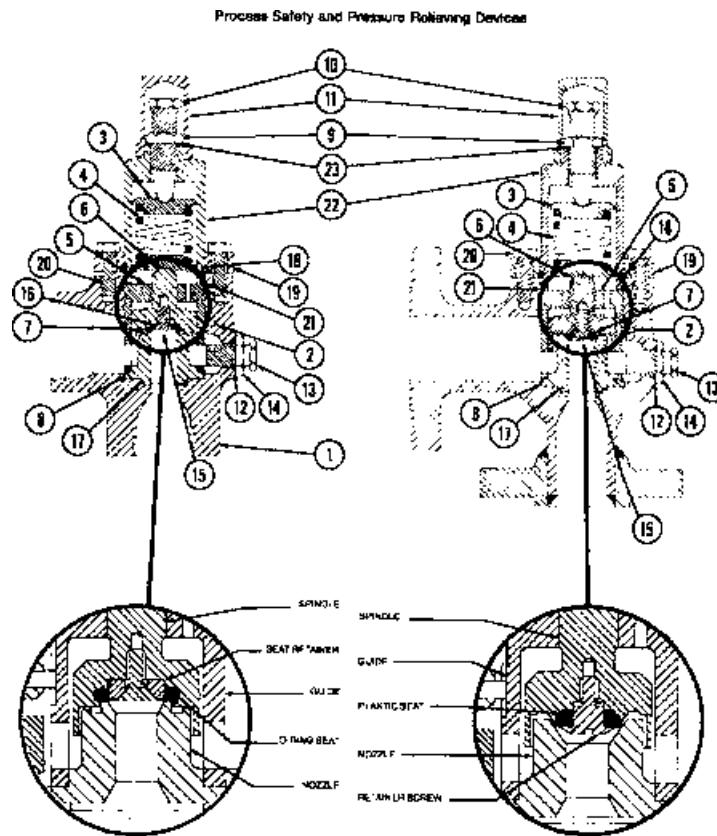
### BALANCED SAFETY RELIEF VALVE

This valve provides an internal design (usually bellows) above/on the seating disk in the huddling chamber that minimizes the effect of back pressure on the performance of the valve (opening pressure, closing pressure, and relieving capacity) [11]. (see Figures 9-4, 9-6a and b).

### SPECIAL VALVES

- a) internal spring safety relief valve
- b) power-actuated pressure relief valve
- c) temperature-actuated pressure-relief valve.

These last three are special valves from the viewpoint of chemical and petrochemical plant applications, but they can be designed by the major manufacturers and instrumentation manufacturers as these are associated with instrumentation controls (Figure 9-7a). Care must be taken in the system design to make certain it meets all ASME code requirements.



**Figure 9-3b** Safety relief valve with rubber or plastic seats. (By permission from Anderson, Greenwood and Co.)

### RUPTURE DISK

A rupture disk is a non-reclosing thin diaphragm (metal, plastic, or carbon/graphite (non-metallic)) held between flanges and designed to burst at a predetermined internal pressure. Each bursting requires the installation of a new disk. It is used in corrosive service, toxic or “leak-proof” applications, and for required bursting pressures not easily accommodated by the conventional valve such as explosions. It is applicable to steam, gas, vapor, and liquid systems (Figures 9-7b, 9-8a–r, and 9-9a–f). There are at least four basic types of styles of disks, and each requires specific design selection attention.

An explosion rupture disk is a special disk or (disc) designed to rupture at high rates of pressure rise, such as runaway reactions. It requires special attention from the manufacturer [11].

Other rupture devices suitable for certain applications are as follows [11]:

- breaking pin device
- shear pin device
- fusible plug device.

**Set pressure:** The set pressure, in pounds per square inch gauge (bars), is the inlet pressure at which the safety or relief valve is adjusted to open [8, 9]. This pressure is set regardless of any back pressure on the discharge of the valve, and is not to be confused with a manufacturer’s spring setting.

**Overpressure:** Pressure increase over the set pressure of the primary relieving device is overpressure. It is the same as accumulation when the relieving device is set at the maximum allowable working pressure (MAWP) of the vessel [8].

**Accumulation:** Pressure increase over the MAWP of the vessel during discharge through the safety or relief valve, expressed as a percent of that pressure, pounds per square inch (bar), is called accumulation [8].

**Blowdown:** Blowdown is the difference between the set pressure and the reseating pressure of a safety or relief valve, expressed as a percent of the set pressure, or pounds per square inch (bar) [8].

**Back pressure:** This is the pressure existing at the outlet or discharge connection of the pressure-relieving device, resulting from the pressure in the discharge system of the installed device [11]. This pressure may be only atmospheric if discharge is directly to atmosphere, or it may be some positive pressure due to pressure drops of flow of discharging vapors/gases (or liquids where applicable) in the pipe collection system or a common header, which in turn may be connected to a blowdown or flare system with definite back pressure conditions during flow, psig (gauge). The pressure drop during flow discharge from the safety relief valve is termed “built-up back pressure.”

**Burst pressure:** This is the inlet static pressure at which a rupture disk pressure-relieving device functions or opens to release internal pressure.

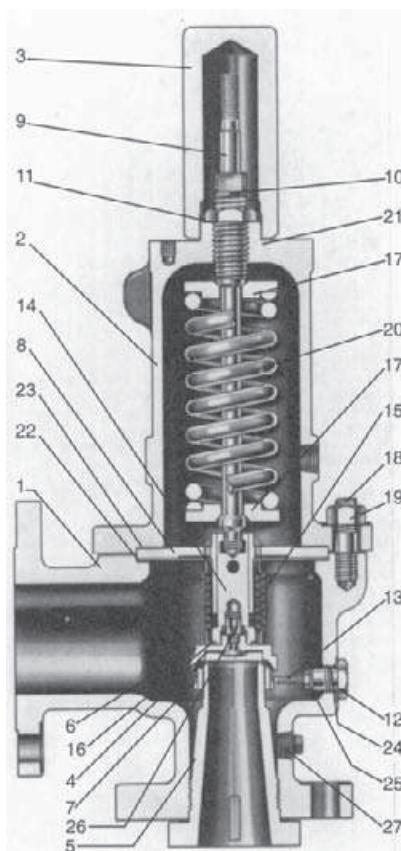
**Design pressure:** This is the pressure used in the vessel design to establish the minimum code permissible thickness for containing the pressure in pounds per square inch gauge (bars).

**Maximum allowable working pressure (MAWP):** This is the maximum pressure in pounds per square inch gauge (bars) permissible at the top of a completed vessel in its operating position for a specific designated temperature. This pressure is calculated in

**Bill of Materials-BalanSeal**

ITEM	PART NAME	MATERIAL
1	26( )810 thru 26( )826	SA-216 GR. WCB, Carbon Steel
	26( )832 thru 26( )836	SA-217 GR. WC6, Alloy St. (1½ CR—½ Moly)
2	26( )810 thru 26( )826	SA-216 GR. WCB, Carbon Steel
	26( )832 thru 26( )836	SA-217 GR. WC6, Alloy St. (1½ CR—½ Moly)
3	Cap Plain Screwed	Carbon Steel
4	Disc	Stainless Steel
5	Nozzle	316 St. St.
6	Disc Holder	300 Series St. St.
7	Blow Down Ring	300 Series St. St.
8	Sleeve Guide	300 Series St. St.
9	Stem	Stainless Steel
10	Spring Adjusting Screw	Stainless Steel
11	Jam Nut (Spc. Adj. Scr.)	Stainless Steel
12	Lock Screw (B.O.R.)	Stainless Steel
13	Lock Screw Ssd	Stainless Steel
14	Stem Retainer	Stainless Steel
15	Bellows	316L St. St.
16	Bellows Gasket	Flexible Graphite
17	Spring Button	Carbon Steel Rust Proofed
18	Body Stud	ASTM A193 Gr. B7, Alloy St.
19	Hex Nut (Body)	ASTM A194 Gr. 2H, Alloy St.
20	26( )810 thru 26( )816	Carbon Steel Rust Proofed
	26( )820 thru 26( )836	High Temp. Alloy Rust Proofed
21	Cap Gasket	Soft Iron or Steel
22	Body Gasket	Soft Iron or Steel
23	Bonnet Gasket	Soft Iron or Steel
24	Lock Screw Gasket	Soft Iron or Steel
25	Hex Nut (B.O.R L.S.)	Stainless Steel
26	Lock Screw (D.M.)	Stainless Steel
27	Pipe Plug (Body)	Steel

Also suitable for liquid service where ASME Code certification is not required.



**Figure 9-4** Balanced nozzle safety relief valve, Balanseal®. (By permission from Teledyne Farris Engineering Co.)

accordance with the ASME code (Par. UG-98) [1] for all parts or elements of the vessel using closest, next larger to calculated value, nominal thickness (closest standard for steel plate) (see Par. UG-A22) but exclusive of any corrosion allowance or other thickness allowances for loadings (see ASME Par. UG-22) on vessels other than pressure (e.g., extreme wind loadings for tall vessels).

The MAWP is calculated using nominal standard steel plates (but could be other metal-use code stresses) thickness, using maximum vessel operating temperature for metal stress determinations. See [1] Par. UG-98.

For the above example, assume calculated MAWP (above) = 80 psig. This is the maximum pressure that any safety relief valve can be set to open.

#### EXAMPLE 9-1 Hypothetical Vessel Design, Carbon Steel Grade A-285, Gr C

Normal operating condition: 45 psig at 600°F

Design pressure: 65 psig at 700°F corresponding to the 65 psig.

Assume calculated thickness per ASME code Par. UG-27: 0.43 in. Closest standard plate thickness to fabricate vessel is 0.50 in. with -0.01 in. and +0.02 in. tolerances at mill.

Then

1. Using 0.50 in. -0.01 in. (tolerance) = 0.49 in. minimum thickness.
2. Using 0.50 in. +0.02 in. (tolerance) = 0.52 in. maximum thickness.

Generally, for design purposes, with this type of tolerance, nominal thickness = 0.50 in. can be used for calculations.

Now, using Par. UG-27, 0.50 in. thickness, and ASME code stress at 750°F (estimated or extrapolated) per Par. UCS-23 at 750°F, the maximum allowable stress in tension is 12,100 psi.

Recalculate pressure (MAWP) using Par. UG-27 [1]

For cylindrical shells under internal pressure:

- (1) Circumferential stress (longitudinal joint):

$$P_d = SEt / (R_i + 0.6t) \quad (\text{psi} = \text{psig}) \quad (9-1)$$

$$t = PR / [SE - 0.6P] \quad (9-2)$$

(continued)

**EXAMPLE 9-1—(continued)**

where

$t$  = minimum actual plate thickness of shell, no corrosion  
 $= 0.50"$

$P_d$  = design pressure, for this example equals the MAWP, psi

$R_i$  = inside radius of vessel, no corrosion allowance added, in.

$S$  = maximum allowable stress, psi, from Table UCS-23

$E$  = joint efficiency for welded vessel joint, plate to plate to heads. See ASME Par. UW-12, nominal = 85% = 0.85.

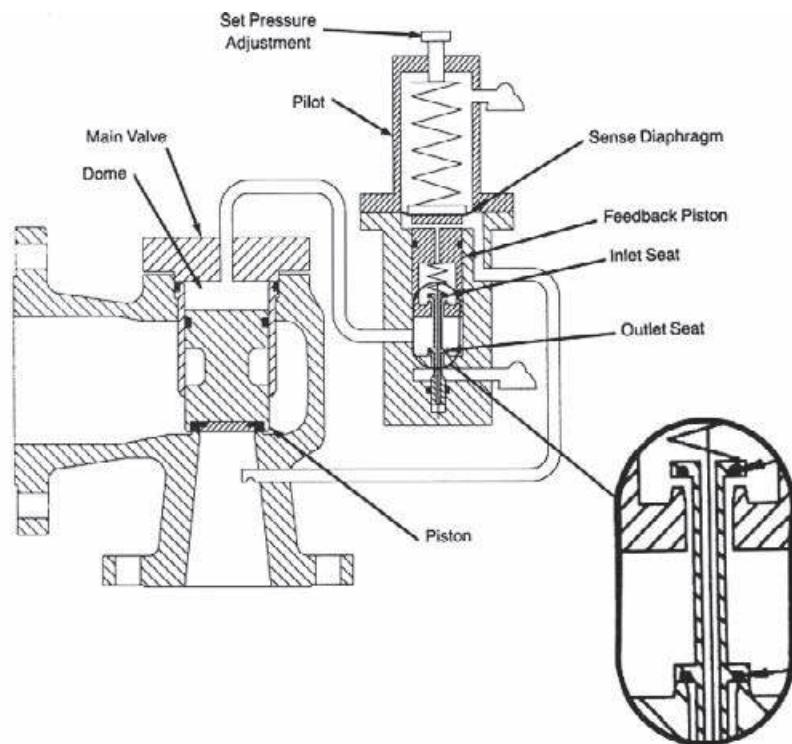
$t$  = required thickness of shell, exclusive of corrosion allowance, inches

**(2) Longitudinal stress (circumferential joints):**

$$P_d = 2SEt/(R - 0.4t) \quad (9-3)$$

$$t = PR/[2SE + 0.4P] \quad (9-4)$$

The vessel shell wall thickness shall be the greater of Eq. (9-2) or (9-4), or the pressure shall be the lower of Eq. (9-1) or (9-3) [1].



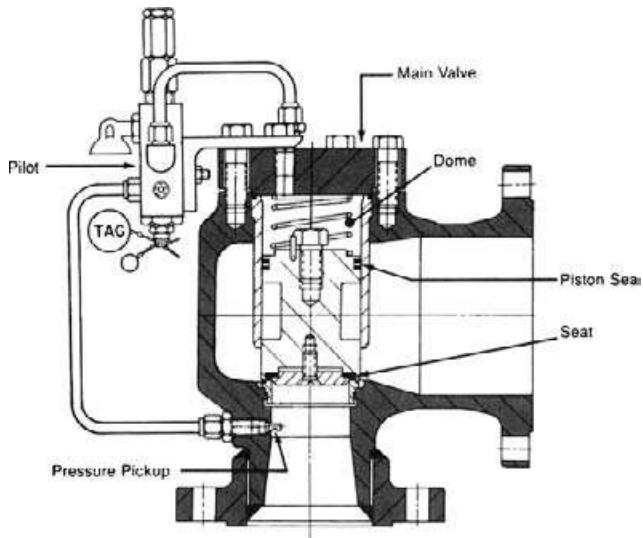
With no system pressure, the pilot inlet seat is open and outlet seat is closed. As pressure is admitted to the main valve inlet, it enters the pilot through a filter screen and is transmitted through passages in the feedback piston, past the inlet seat, into the main valve dome to close the main valve piston.

As system pressure increases and approaches valve set pressure, it acts upward on the sense diaphragm, with the feedback piston moving upward to close the inlet seat, thus sealing in the main valve dome pressure, as the outlet seat is also closed. A small, further increase in system pressure opens the outlet seat, venting the main valve dome pressure. This reduced dome pressure acts on the unbalanced feedback piston to reduce feedback piston lift, tending to "lock in" the dome pressure. Thus, at any stable inlet pressure there will be no pilot flow (i.e. zero leakage).

As inlet pressure rises above set pressure, dome pressure reduction will be such as to provide modulating action of the main valve piston proportional to the process upset. The spool/feedback piston combination will move, responding to system pressure, to alternately allow pressure in the main valve dome to increase or decrease, thus moving the main valve piston to the exact lift that will keep system pressure constant at the required flow. Full main valve lift, and therefore full capacity, is achieved with 5% overpressure. As system pressure decreases below set pressure, the feedback piston moves downward and opens the inlet seat to admit system pressure to the dome, closing the main valve.

Due to the extremely small pilot flow, the pilot on gas/vapor valves normally discharge to atmosphere through a weather and bug-proof fitting. Pilots for liquid service valves have their discharge piped to the main valve outlet.

Figure 9-5a Pilot-operated safety relief valve. (By permission from Anderson, Greenwood and Co.)



**Figure 9-5b** Safety relief valve mechanism as connected to a non-flow (zero flow) pilot safety valve. (By permission from Anderson Greenwood and Co.)

For pressure levels of pressure-relief valves referenced to this MAWP, see Figures 9-7a and b.

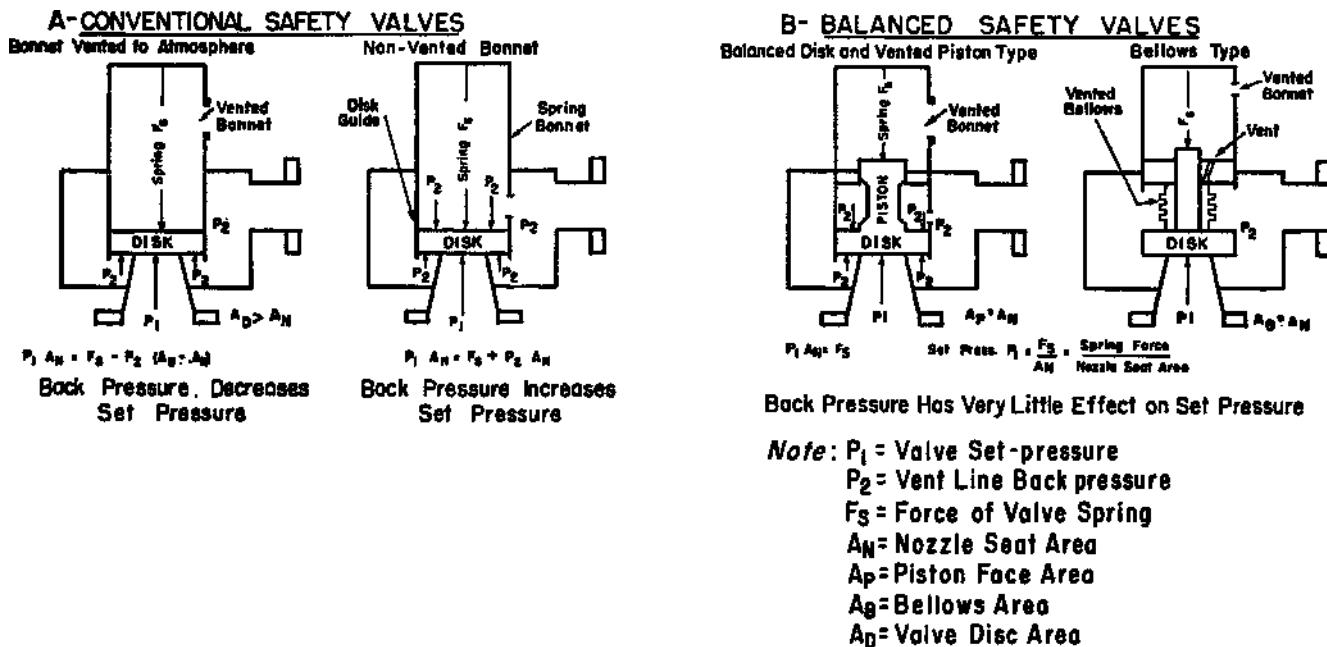
**Operating pressure:** This is the pressure, psig (barg), to which the vessel is expected to be subjected during normal or the maximum probable pressure during upset operations. There is a difference between a pressure generated internally due to controlled rising vapor pressure (and corresponding temperature) and that generated due to an unexpected runaway reaction, where reliance must depend on the sudden release of pressure at a code conformance pressure/temperature. In this latter case, careful examination of the possible conditions for a runaway reaction should be made.

This examination is usually without backup data or a firm basis for calculating possible maximum internal vessel pressure to establish a maximum operating pressure and, from this, a design pressure.

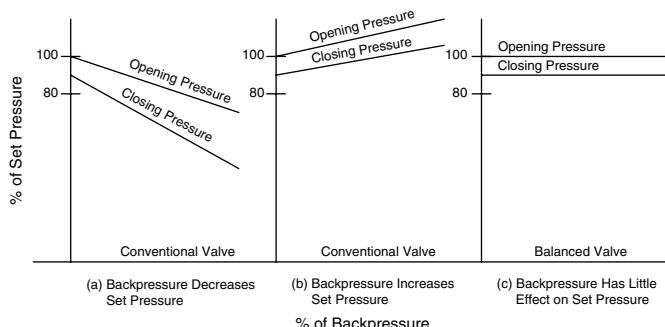
**Design pressure of a vessel:** This is the pressure established as a nominal maximum above the expected process maximum operating pressure. This design pressure can be established by reference to the chart in Chapter 1 (Figure 1-40), which is based on experience/practice and suggests a percentage increase of the vessel design pressure above the expected maximum process operating pressure level. There is no code requirement for establishing the design pressure. (see Figure 1-40.) Good judgment is important in selecting each of these pressures. See operating pressure description in above paragraph. Depending on the actual operating pressure level, the increase usually varies from a minimum of 10% higher or 25 psi, whichever is greater, to much higher increases. For instance, if the maximum expected operating pressure in a vessel is 150 psig, then experience might suggest that the design pressure be set for 187–200 psig. Other factors known regarding the possibility of a runaway reaction might suggest setting it at 275 psig. A good deal of thought needs to enter into this pressure level selection. (Also see Sections 9.52 and 9.91 [12, 13].)

**Relieving pressure:** This is the pressure-relief device's set pressure plus accumulation or overpressure (see Figures 9-7a and 9-7b). For example, at a set pressure equal to the maximum allowable at the MAWP of the vessel of 100 psig, and for process internal vessel pressure, the pressure-relief device would begin relieving at nominal 100 psig (actually begins to open at 98 psig, see figures above) and the device (valve) would be relieving at its maximum conditions at 110 psig (the 10 psig is termed the "accumulation pressure") for a single valve installation, or 116 psig for a multiple valve installation on the same vessel. These are all process situations, which do not have an external fire around the vessel (see Section 9.26 and Figures 9-7b, 9-32a, b for these allowable pressure levels) and in no case do the figures apply to a sudden explosion internally.

**Reseating pressure:** This is the pressure level after valve opening under pressure that the internal static pressure falls to



**Figure 9-6a** Effect of back pressure on set pressure of safety or safety relief valves. (By permission from *Recommended Practice for Design and Construction of Pressure-Relieving Systems in Refineries*, API RP-520, 5th ed., American Petroleum Institute (1990) (also see [5a]).)



**Figure 9-6b** Diagram of approximate effects of back pressure on safety relief valve operation. (Adapted by permission from Teledyne Farris Engineering Co.)

when there is no further leakage through the pressure-relief valve (see Figure 9-7a).

**Closing pressure:** This is the pressure established as decreasing inlet pressures when the disk of the valve seats and there is no further tendency to open or close.

**Simmer:** This is the audible or visible escape of fluid between the seat and disk of a pressure-relieving valve at an inlet static pressure below the popping pressure, but at no measurable capacity of flow (for compressible fluid service).

**Popping pressure:** This is the pressure at which the internal pressure in a vessel rises to a value that causes the inlet valve seat to begin to open and to continue in the opening direction to relieve the internal overpressure greater than the set pressure of the device (for compressible fluid service).

### 9.3 MATERIALS OF CONSTRUCTION

#### SAFETY AND RELIEF VALVES: PRESSURE-VACUUM-RELIEF VALUES

For most process applications, the materials of construction can be accommodated to fit both the corrosive-erosive and mechanical strength requirements. Manufacturers have established standard materials, which will fit a large percentage of the applications, and often only a few parts need to be changed to adapt the valve to a corrosive service. Typical standard parts are (see Figures 9-3a, 9-3b, and 9-4) as follows:

Option 1 (typical only)	
Body	carbon steel, SA 216, gr. WCB
Nozzle	316 stainless steel
Disk/Seat	stainless steel
BlowDown Ring	300 Ser. stainless steel
Stem or Spindle	stainless steel
Spring	C.S. rust proof or high temp. alloy, rust proof
Bonnet	SA-216, gr. WCB carbon steel
Bellows	316L stainless steel
Option 2 (typical only)	
Body	316 stainless steel
Nozzle	17-4 stainless steel or 316 stainless steel
Disk/Seat	Teflon, Kel-F, Vespel or Buna-N
BlowDown Ring	316 stainless steel
Stem or Spindle	17-4 stainless steel or 316 SS
Spring	316 stainless steel
Bonnet	316 stainless steel
Bellows	-

For pressure and temperature ratings, the manufacturers' catalogs must be consulted. In high pressure and/or temperature, the materials are adjusted to the service.

For chemical service the necessary parts are available in 3.5% nickel steel; Monel; Hastelloy C; Stainless Type 316, 304, and so on.; plastic-coated bellows; nickel; silver; nickel-plated springs; and other workable materials.

The designer must examine the specific valve selected for a service and evaluate the materials of construction in contact with the process as well as in contact with or exposed to the vent or discharge system. Sometimes the corrosive nature of the materials in the vent system presents a serious corrosion and fouling problem on the back or discharge side of the valve while it is closed.

For these special situations, properly designed rupture disks using corrosion-resistant materials can be installed both before the valve inlet as well as on the valve discharge. For these cases, refer to both the valve manufacturers and the rupture disk manufacturers. (See Section 9.4 for this condition.)

### RUPTURE DISKS

Rupture disks are available in

1. practically all metals that can be worked into thin sheets, including lead, Monel, nickel, aluminum, silver, Inconel, 18-8 stainless steel, platinum, copper, Hastelloy, and others;
2. plastic-coated metals, lead-lined aluminum, and lead-lined copper;
3. plastic seals of polyethylene, Kel-F®, and Teflon®;
4. graphite, impregnated graphite, or carbon.

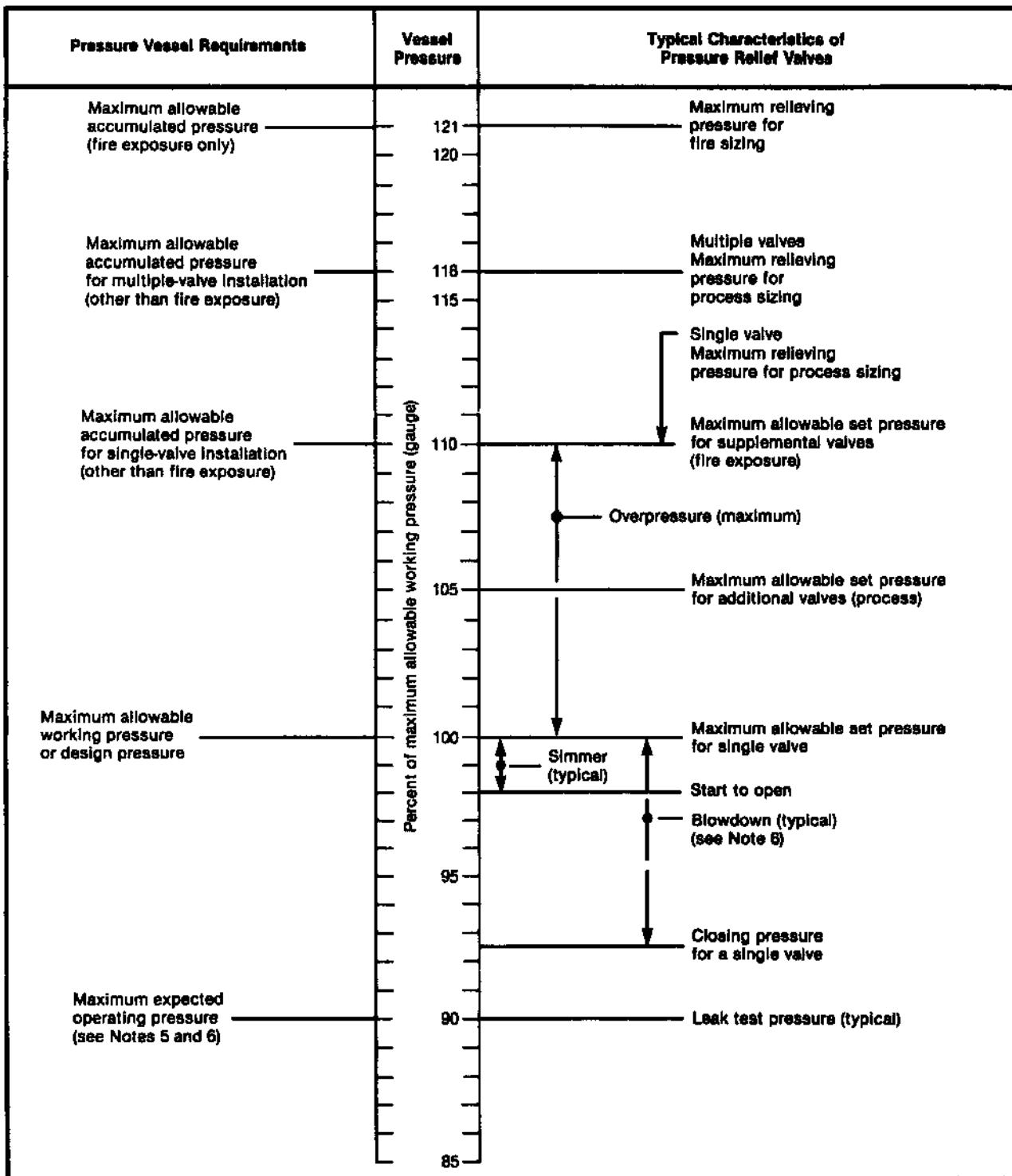
The selection of the material suitable for the service depends upon the corrosive nature of the fluid and its bursting characteristics in the pressure range under consideration. For low pressure, a single standard disk of some materials would be too thin to handle and maintain its shape, as well as give a reasonable service life from the corrosion and fatigue standpoints (see Section 9.9).

### 9.4 GENERAL CODE REQUIREMENTS [1]

It is essential that the ASME code requirements be understood by the designer and the individual rating and specifying the installation details of the safety device. It is not sufficient to merely establish an orifice diameter, since process considerations which might cause overpressure must be thoroughly explored in order to establish the maximum relieving conditions.

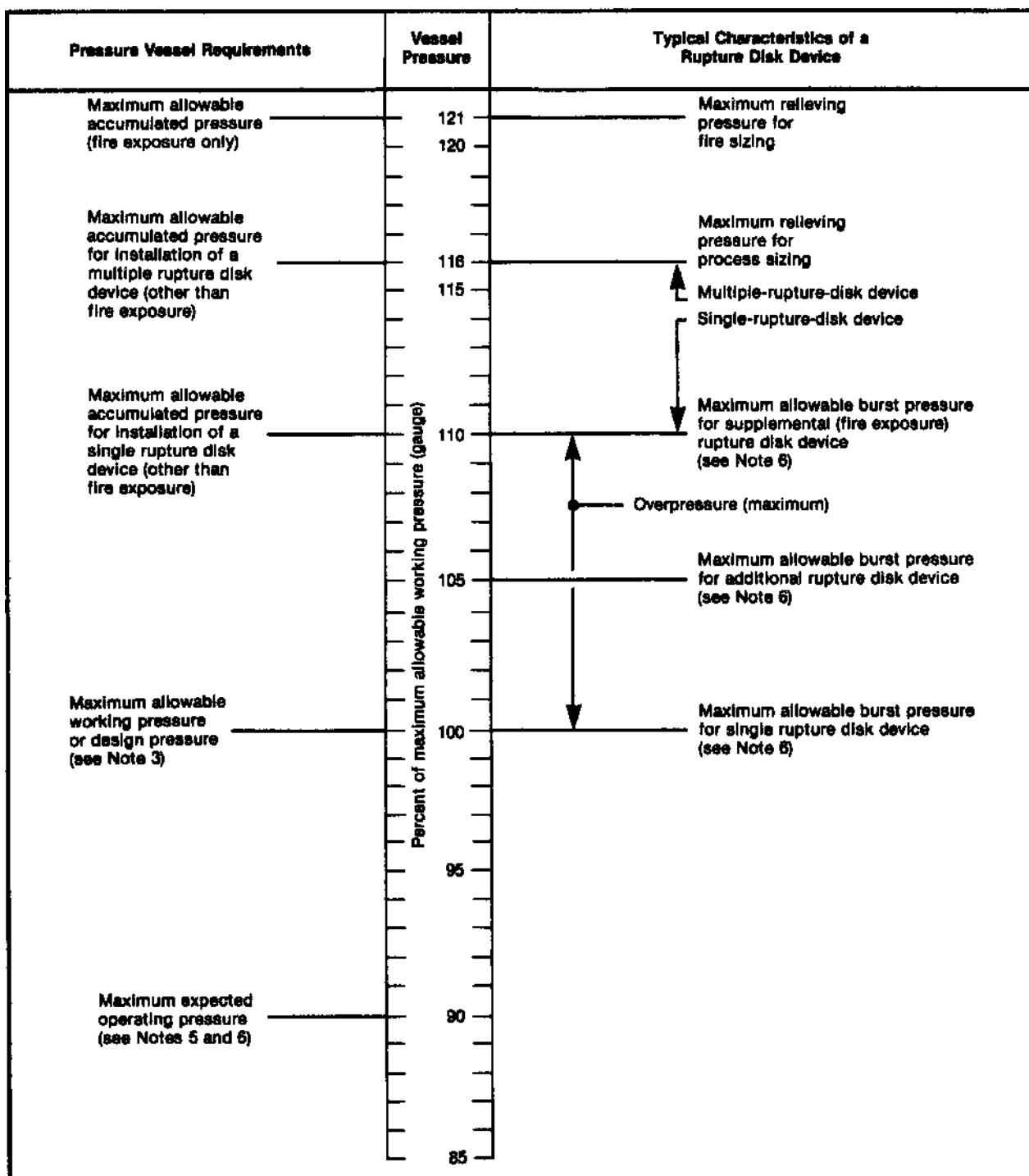
An abbreviated listing of the key rating provisions is given in paragraphs UG-125–135 of the ASME code, Section 8, Division 1, for unfired pressure vessels [1].

1. All pressure vessels covered by Division 1 or 2 of Section VIII are to be provided with protective overpressure devices. There are exceptions covered by paragraph U-1 of the code, and in order to omit a protective device this paragraph should be examined carefully. For example, vessels designed for pressure tolerances above 3000 psi are not covered; also vessels with <120 gallons of water, vessels with inside diameter not over 6 in. (at any pressure), vessels having internal or external operating pressures not over 15 psig (regardless of size), and a few other conditions may not be subject to this code.
2. Unfired steam boilers must be protected.
3. Pressure relief must be adequate to prevent internal pressures from rising over 10% above the MAWP, except when the excess pressure is developed by external fire or other unforeseen heat source. (see design details in later paragraph.) Papa [14]

**Notes:**

1. This figure conforms with the requirements of Section VIII of the ASME Boiler and Pressure Vessel Code.
2. The pressure conditions shown are for pressure-relief valves installed on a pressure vessel.
3. Allowable set-pressure tolerances will be in accordance with the applicable codes.
4. The maximum allowable working pressure is equal to or greater than the design pressure for a coincident design temperature.
5. The operating pressure may be higher or lower than 90.
6. Section VIII, Division 1, Appendix M, of the ASME Code should be referred to for guidance on blowdown and pressure differentials.

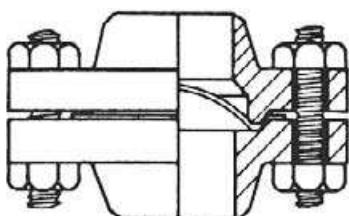
Figure 9-7a Pressure level relationship for pressure-relief valve installed on a pressure vessel (vapor phase). Single valves (or more) used for process or supplemental valves for external fire (see labeling on chart). (Reprinted by permission from *Sizing, Selection and Installation of Pressure Relieving Devices in Refineries*, Part 1 "Sizing and Selection", API RP-520, 5th ed., Jul 1990, American Petroleum Institute.)

**Notes:**

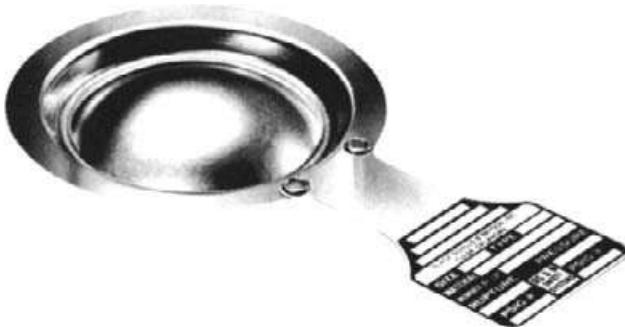
- This figure conforms with the requirements of Section VIII of the ASME Boiler and Pressure Vessel Code.
- The pressure conditions shown are for rupture disk devices installed on a pressure vessel.
- The margin between the maximum allowable working pressure and the operating pressure must be considered in the selection of a rupture disk.

- The allowable burst-pressure tolerance will be in accordance with the applicable code.
- The operating pressure may be higher or lower than 90 depending on the rupture disk design.
- The stamped burst pressure of the rupture disk may be any pressure at or below the maximum allowable burst pressure.

**Figure 9-7b** Pressure level relationships for rupture disk devices. (Reprinted by permission from *Sizing, Selection and Installation of Pressure Relieving Devices in Refineries*, Part 1 "Sizing and Selection", API RP-520, 5th ed., Jul 1990, American Petroleum Institute.)



**Figure 9-8a** Metal type frangible disk (above) with cross section (below). (Courtesy of Black, Sivalis and Bryson Safety Systems, Inc.)



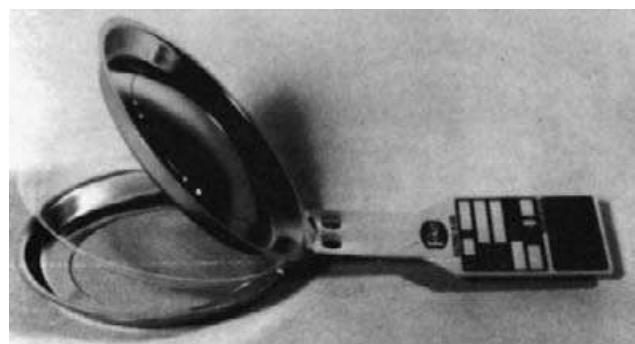
**Figure 9-8b** Standard rupture disk. A prebulged rupture disk available in a broad range of sizes, pressures, and metals. (By permission from B.S. & B. Safety Systems, Inc.)



**Figure 9-8c** Disk of Figure 9-8b after rupture. Note: 30° angular seating in holder is standard for prebulged solid metal disk. (By permission from B.S. & B. Safety Systems, Inc.)



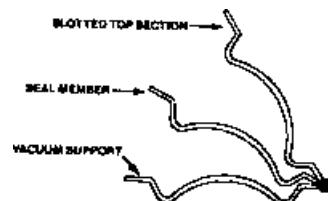
**Figure 9-8d** Disk of Figure 9-8b with an attached (underside) vacuum support to prevent premature rupture in service with possible less than atmospheric pressure on underside and/or pulsation service. (By permission from B.S. & B. Safety Systems, Inc.)



**Figure 9-8e** Rupture disk (top) with Teflon® or other corrosion-resistance film/sheet seal, using an open retaining ring. For positive pressure only. (By permission from Fike Metal Products Div., Fike Corporation, Blue Springs, MO.)



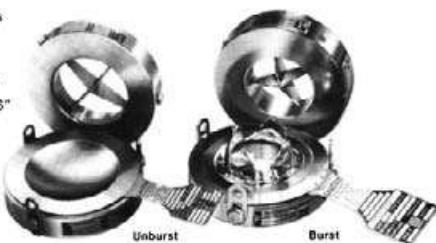
**Figure 9-8f(a)** Rupture disk (top), similar to Figure 9-8e, except a metal vacuum support is added (see Figure 9-8f(b)). (By permission from Fike Metal Products Div., Fike Corporation, Blue Springs, MO.)



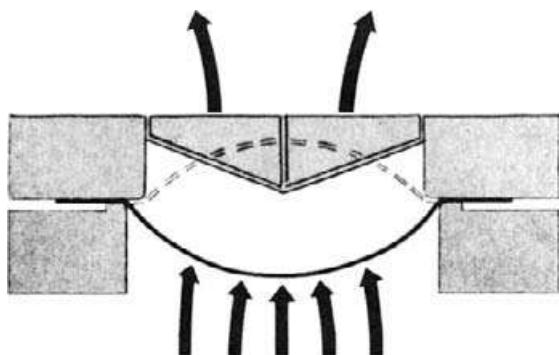
**Figure 9-8f(b)** Cross section of disk assembly for Figure 9-8f(a). (By permission from Fike Metal Products Div., Fike Corporation, Blue Springs, MO.)

**FEATURES:**

- Isolates Safety Relief Valves
- No Fragmentation
- Operates up to 90% Rated Pressure
- Can Withstand Full Vacuum without Supports
- Available in Sizes 1" thru 36"
- Wide Material Availability
- U.S. Patent Number 3,294,277

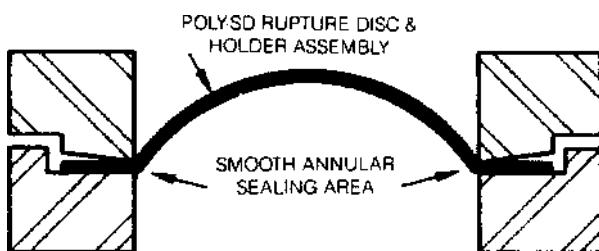


**Figure 9-8g(a)** Reverse buckling® disk, showing top holder with knife blades (underside) that cut the disk at time of rupture. (By permission, B.S. & B. Safety Systems, Inc.)



**RB-90 reverse buckling® disk.** Pressure on convex side of disk and patented sealing design puts compression loading on disk metal.

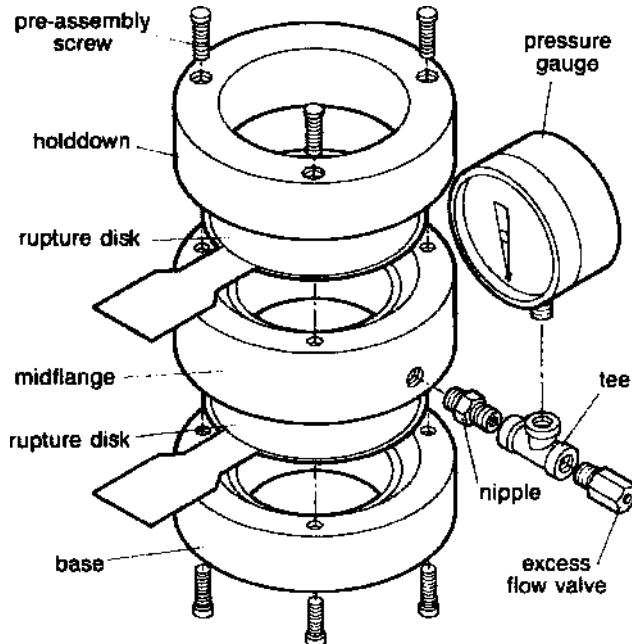
**Figure 9-8g(b)** Reverse buckling® disk. Pressure on convex side of disk and patented sealing design puts compression loading on disk metal. (By permission from B.S. & B. Safety Systems, Inc.)



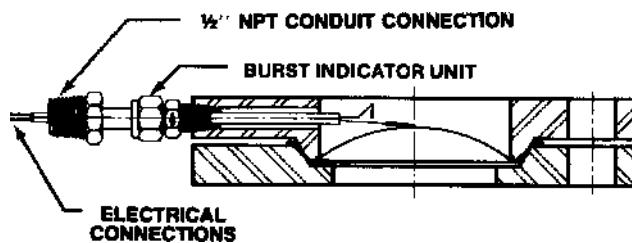
**Figure 9-8h** Special metal disk holder for polymer systems using a smooth disk surface to reduce polymer adherence and a smooth annular sealing area. Usually thick to avoid need for vacuum support and to allow for corrosion attack. (By permission from Fike Metal Products Co. Div., Fike Corporation, Inc.)



**Figure 9-8i** Flat disk used for low pressure and for isolation of corrosive environments. Usual pressure range is 2–15 psig with  $\pm 1$  psi tolerance. Stainless steel disk with Teflon® seal usually standard. (By permission from Fike Metal Products Div., Fike Corporation. Catalog 73877-1, p. 35.)



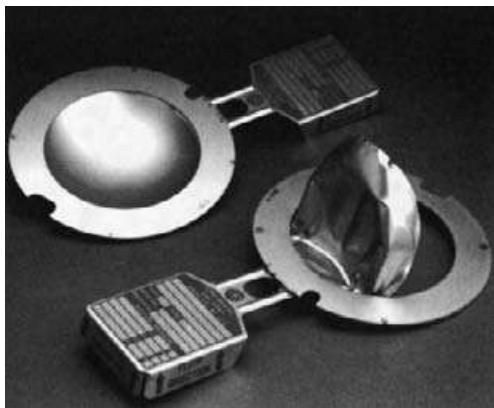
**Figure 9-8j** Exploded view of double disk assembly. Usually burst pressure is same for each disk. Used for corrosive/toxic conditions to avoid premature loss of process and at remote locations. Note the use of tell-tale between disks. (By permission from Fike Metal Products Div., Fike Corporation, Inc.)



**Figure 9-8k** Rupture disk with burst indicator. Several other techniques available. (By permission from Fike Metal Products Div., Fike Corporation Inc.)



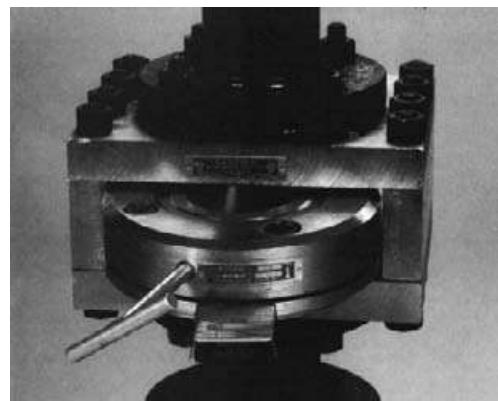
**Figure 9-8l** Rupture disk indicator alarm trip breaks when rupture disk breaks and alarms to monitor. FM system approved. (By permission from Continental Disc Corporation.)



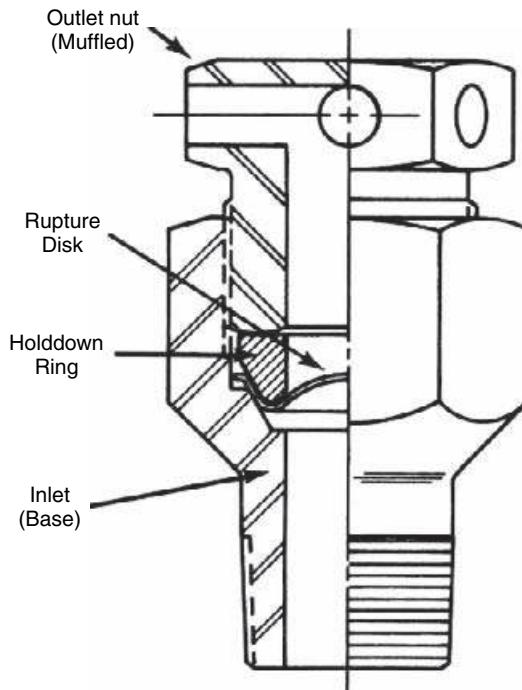
**Figure 9-8m** Ultrex® reverse acting rupture disk providing instantaneous full opening, non-reclosing. (By permission from Continental Disc Corporation.)

proposes an improved technique for relief sizing. (Also see Figures 9-7a and b.)

4. When a pressure vessel is exposed to external heat or fire, supplemental pressure-relieving devices are required for this excessive pressure. These devices must have capacity to limit the overpressure to not more than 21% above the MAWP of the vessel (see Figures 9-7a and b). A single relieving device may be used to handle the capacities of paragraph UG-125 of the code, provided it meets the requirements of both conditions described.
5. Rupture disks may be used to satisfy the requirements of the code for conditions such as corrosion and polymer formations,



**Figure 9-8n** Quick change™ for quick rupture disk changeout. (By permission from Continental Disc Corporation.)



**Figure 9-8o** Reusable screw type holder (30° seat) for smaller disks. (By permission from Fike Metal Products Div., Fike Corporation, Inc.)

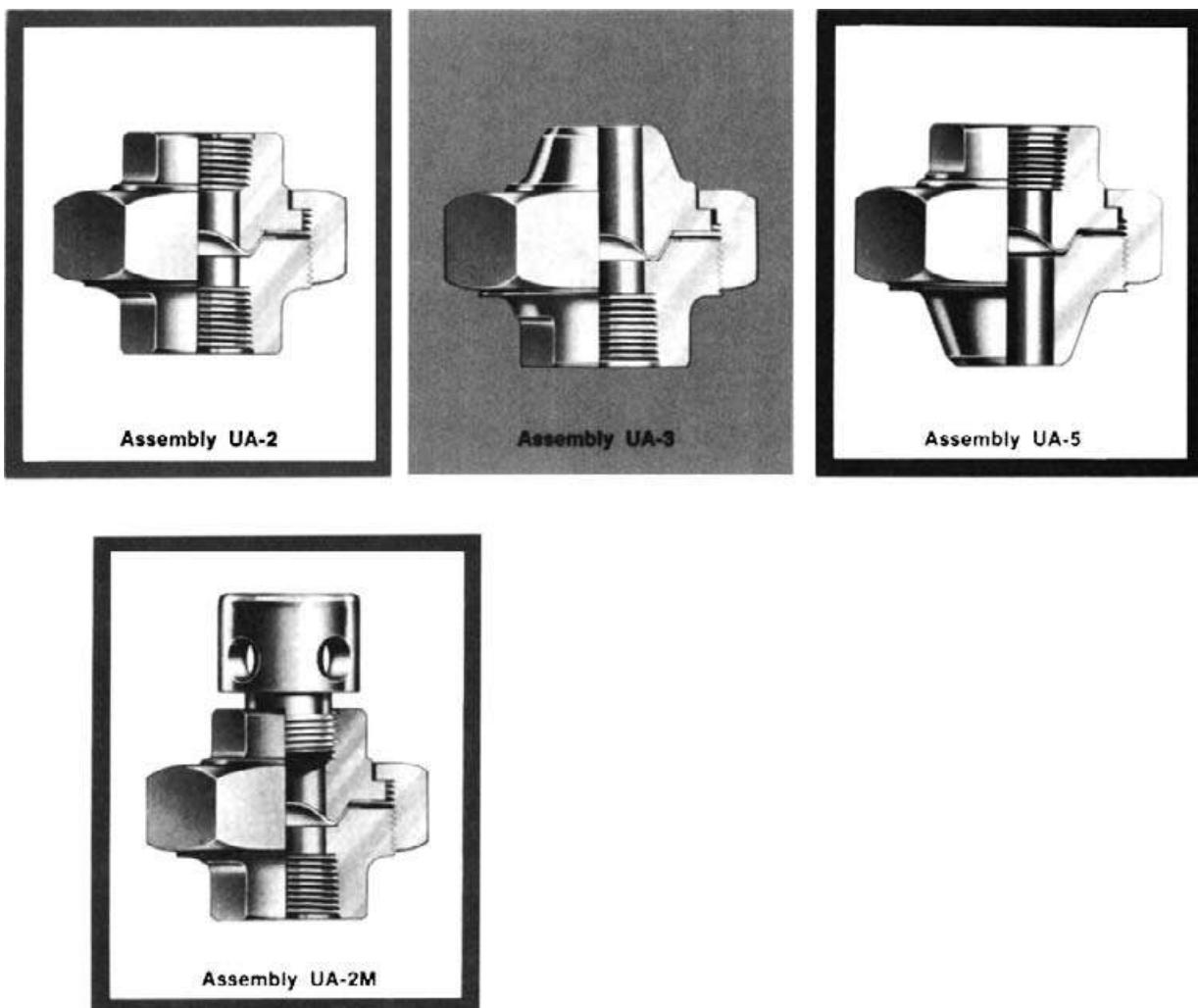
which might make the safety/relief valve inoperative, or where small leakage by a safety valve cannot be tolerated. They are particularly helpful for internal explosion pressure release.

6. Liquid relief valves should be used for vessels that operate full of liquid.

## 9.5 RELIEF MECHANISMS

### RECLOSING DEVICES, SPRING LOADED

Safety and relief valves must be the direct spring-loaded type, and for pressure ranges noted below the code [1] requires the following:



**Figure 9-8p** Typical union type disk holders. They all are not available. (By permission from B.S & B. Safety Systems, Inc.)

Set Pressure	Maximum Spring Reset Referenced to Set Pressure*
$\leq 250 \text{ psig}$	$\pm 10\%$
$\geq 250 \text{ psig}$	$\pm 5\%$

\*Marked on value

The set pressure tolerances of pressure-relief valves are not to exceed  $\pm 2 \text{ psi}$  for pressures up to and including 70 psig and  $\pm 3\%$  for pressures above 70 psig. Indirect operation of safety valves, for example, by pilot valve, is not acceptable unless the primary unloading valve will automatically open and will operate fully in accordance with design relieving capacity conditions if some essential part of the pilot or auxiliary device should fail [1].

The pilot valve is a self-actuated pressure-relief valve that controls the main valve opening.

#### NON-RECLOSED PRESSURE-RELIEVING DEVICES

Rupture disks must have a specified bursting pressure at a specified temperature. There must be complete identification of the metal-

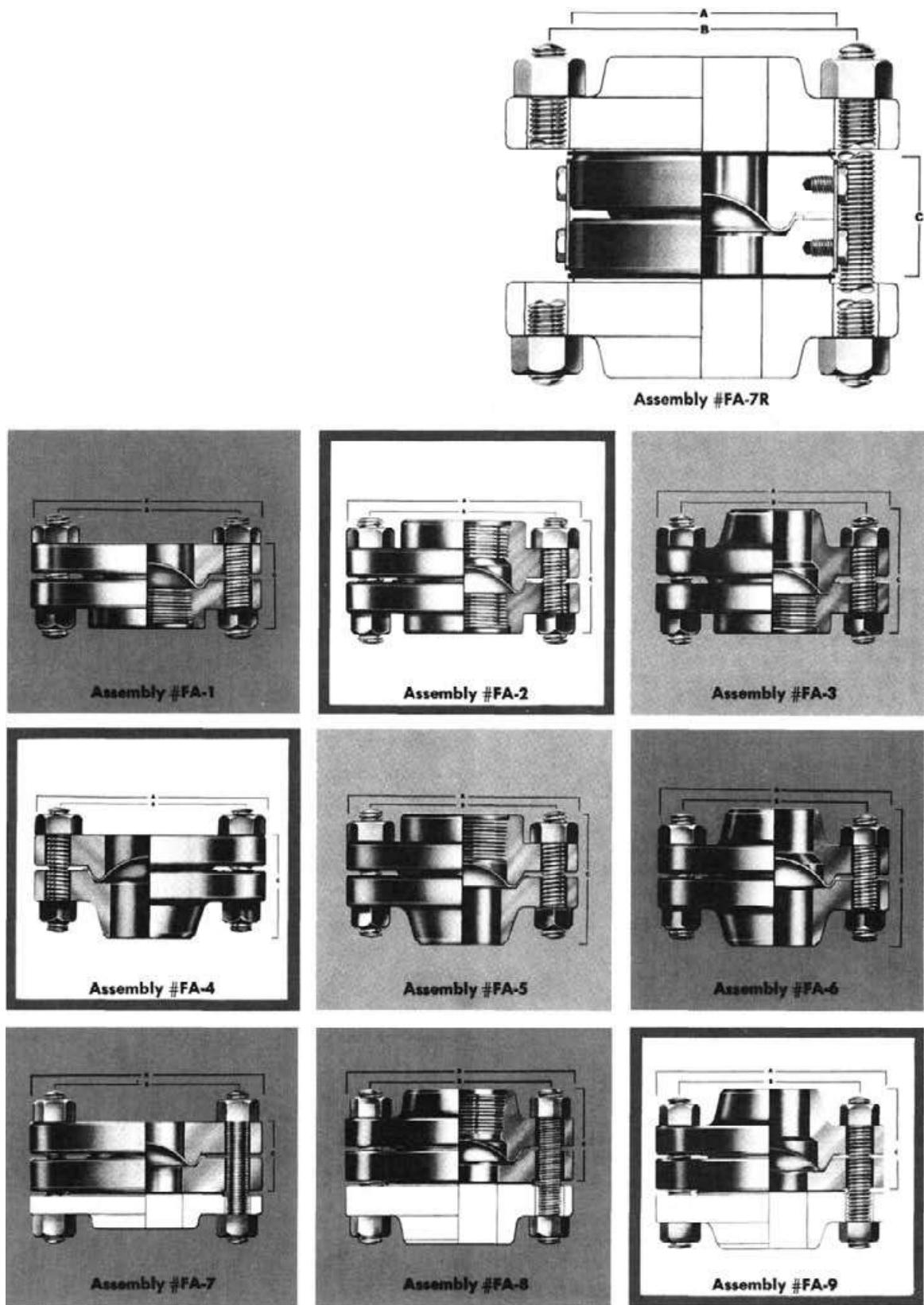
lurgy (if metal) or other properties if graphite or plastic, and the disk must be guaranteed by the manufacturer to burst within  $\pm 5\%$  of the specified bursting pressure at the rated temperature.

The connection nozzle holding the disk must have a net cross-sectional area no less than that required for the design rated conditions of the disk.

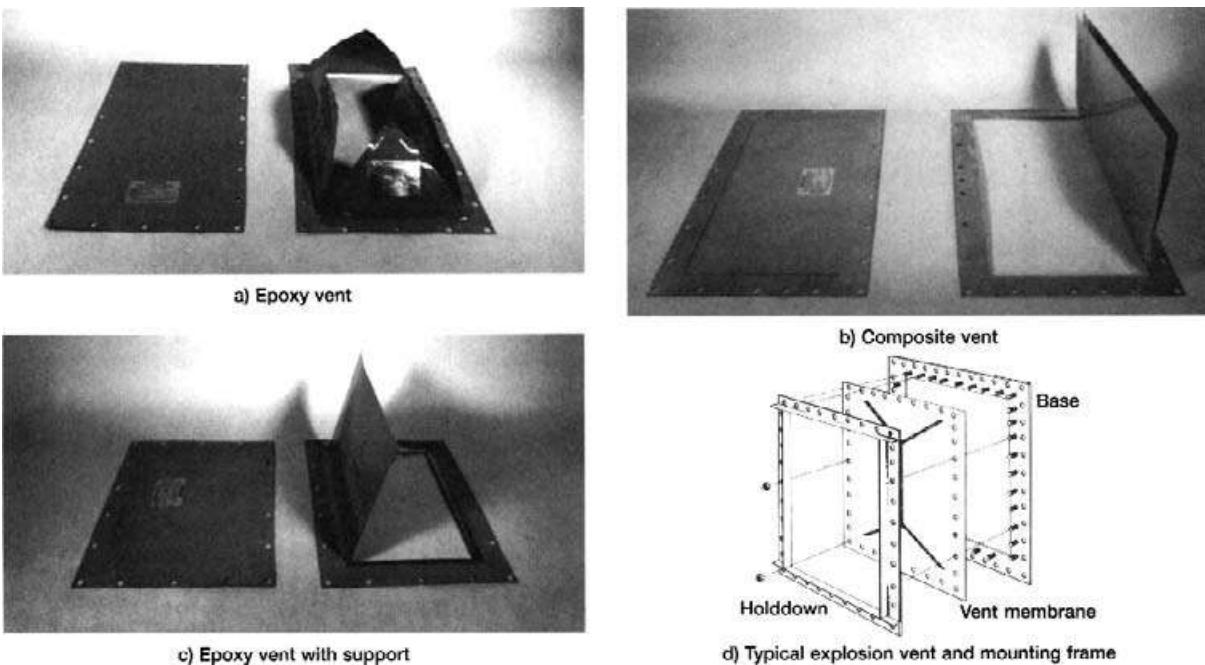
The certification of disk performance is to be based on actual bursting tests of two or more disks from a lot of the same material of the same size as the disk to be sold by the manufacturer. The holder for the test disks must be identical to the design, dimensions, and so on for the disk being certified. (for details, see ASME code, Par. UG-127 [1]).

#### 9.6 PRESSURE SETTINGS AND DESIGN BASIS

*Unfired* steam boilers, that is, nominally termed “waste heat boilers” or “heat exchangers”, which generate steam by heat interchange with other fluids (see ASME code [1] Par. U-1 (g)), should be equipped with pressure-relieving devices required by the ASME code, Section 1, as far as applicable; otherwise, refer Par. UG-125.



**Figure 9-8q** Typical bolted type safety head assemblies with angular seat design. (By permission from B.S. & B. Safety Systems, Inc.)



**Figure 9-8r** Low pressure bursting vents for explosion relief on storage silos, buildings, and so on. Usually burst 1.0–8.0 psig depending on design. Round vents also available. (By permission from Fike Metal Products Div., Fike Corporation, Inc.)



**Figure 9-9a** Non-metal frangible disk. Ruptured disk showing complete breakout of membrane. (Courtesy of Fails Industries, Inc.)

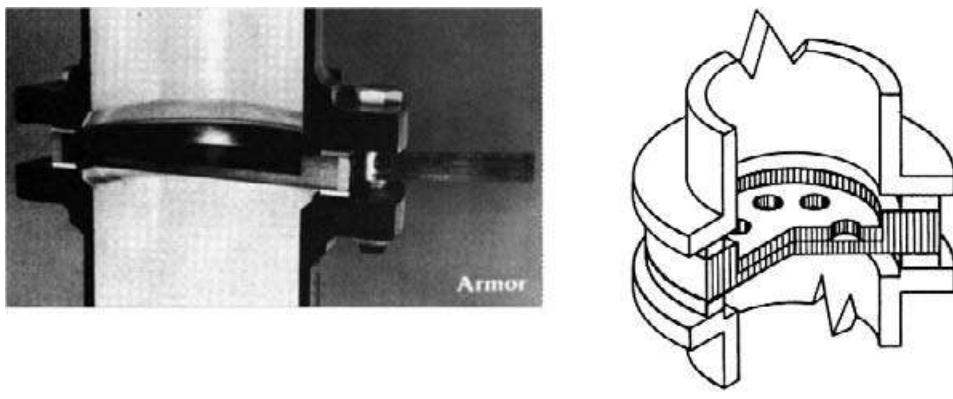


**Figure 9-9b** Standard non-metal frangible disk (graphite); Teflon® coatings or linings are available on entire disk. (By permission from Zook enterprises.)

Vessels, as per Par. U-1(g), which follow Par. UG-125ff are as follows:

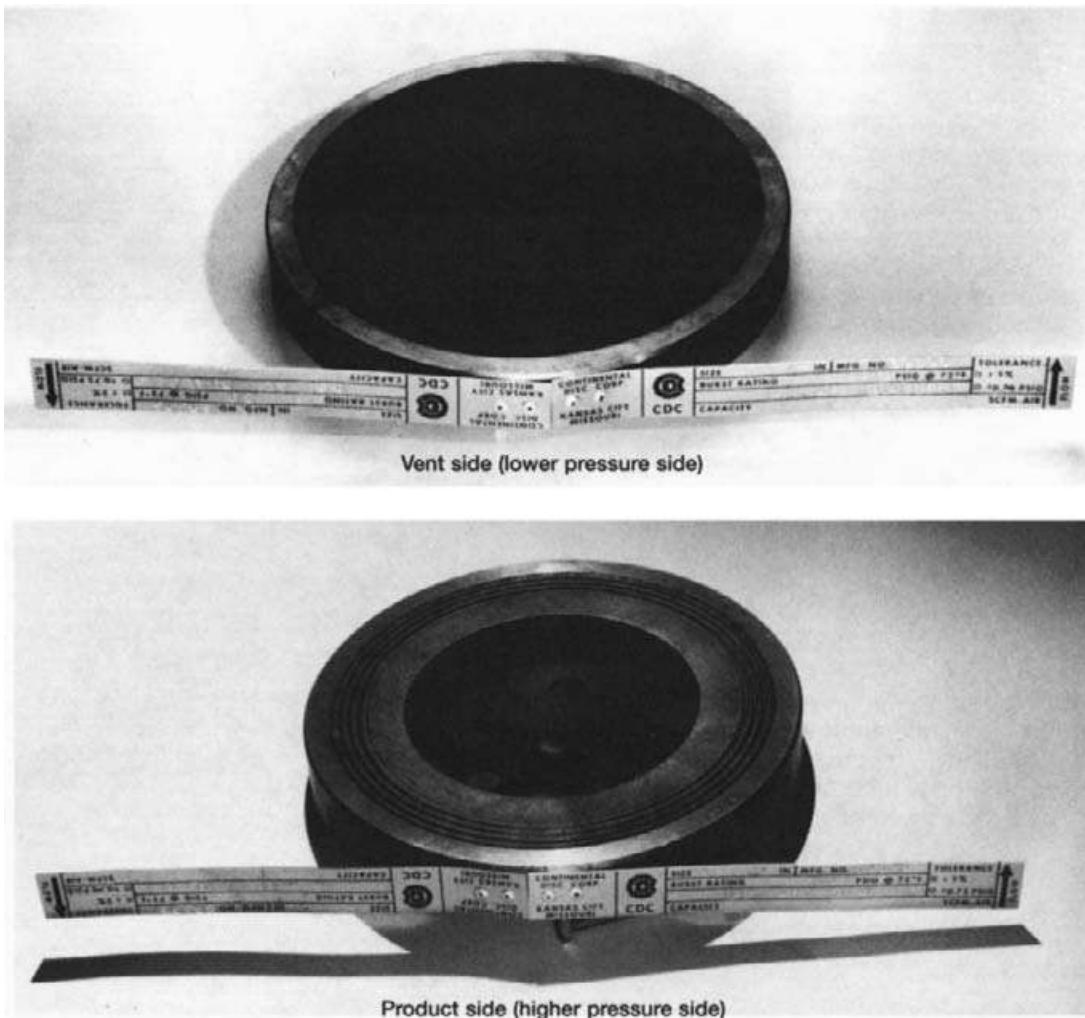
1. Evaporators or heat exchangers.
2. “Vessels in which steam is generated by the use of heat resulting from operation of a processing system containing a number of pressure vessels such as used in the manufacturer of chemical and petrochemical products” [1].
3. Par. U-1 (h) “Pressure vessels or parts subject to direct firing from the combustion of any fuel, which are not within the scope of Sections I, III or IV, may be constructed in accordance with the rules of Section VIII, Division I, Par. UW-2 (d)” [1].

To meet code requirements, the relieving device must be directly open to the system to be relieved, see Figures 9-10, 9-11, and 9-12. For Figures 9-10, 9-11, and 9-12, the rupture disk and the relief valve must be designed to handle the relieving capacity at the relieving temperature without allowing more than a 10% pressure buildup above the MAWP of the unfired pressure vessel (or corresponding overpressure for other code requirements). Figure 9-11 requires that the rupture disk be designed the same as for Figures 9-10, 9-13a and b and Figure 9-12 requires that the relief valve be the primary device and meets the process relief requirements; it may have additional capacity to accommodate such conditions as external fire, or this additional requirement may be installed in a separate relief valve or rupture disk as shown. Also the separate rupture disk may be in a secondary function not covered by the code for such conditions as runaway reactions and internal explosion. For these conditions the setting of the rupture disk is left up to the designer, and may be higher than that for the usual relief. Of course, it should be set sufficiently below the rupture condition for the vessel or component in order to avoid a hazardous condition and meet code requirements.



Rupture disc installation

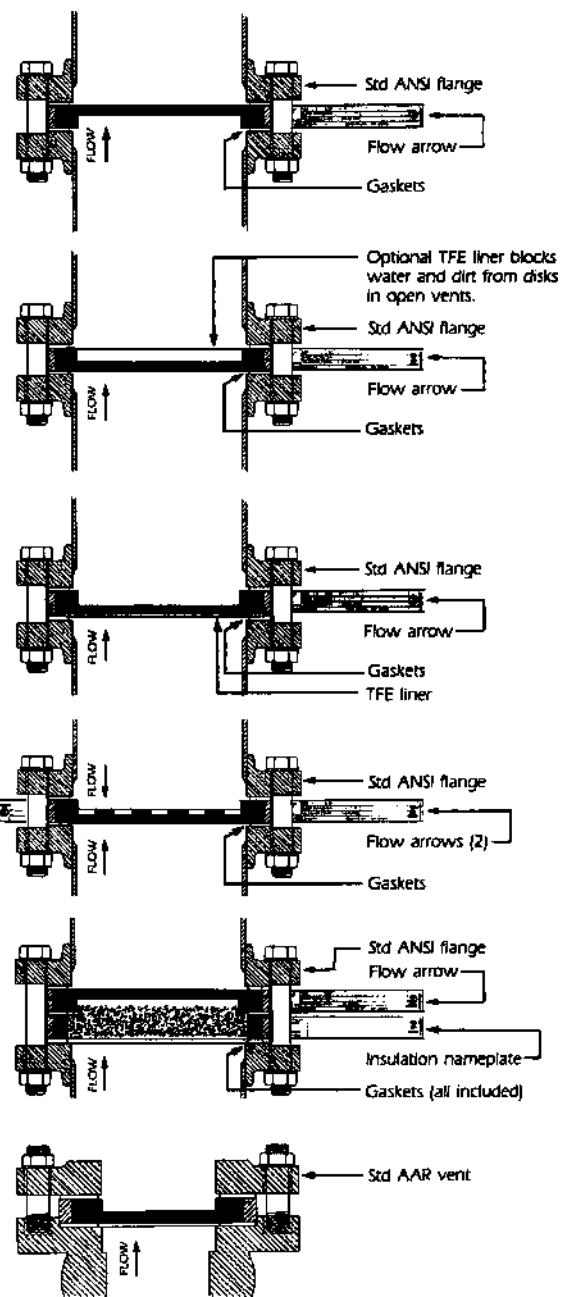
**Figure 9-9c** Armored graphite disk. *Note:* steel ring bonded to circumference of disk to increase safety in toxic or flammable services and improve reliability by preventing unequal piping stresses from reaching the pressure membrane. Teflon® coatings or linings are available on the entire disk. (By permission from Zook Enterprises.)



**Figure 9-9d** Protection against two different pressures from opposite directions using graphite disks, such as in closed storage tanks; particularly API-type to guard against failure of primary breathers, conservation vents, and so on. These require a differential of at least 10 psig between the two burst ratings, depending on diameters of disks. (By permission from Continental Disc Corporation.)

## Duplex

**DUPLEX** Disks extend corrosion resistance to highly oxidizing agents, halogens except free fluorine, and virtually all other corrosives. A sheet of PTFE is used as a barrier on the service side of the disk. Additionally, these disks are processed to accommodate temperatures upto 392°F without insulation.



**Figure 9-9e (a)&(b)** Duplex and insulated disks. (By permission from Zook Enterprises.)



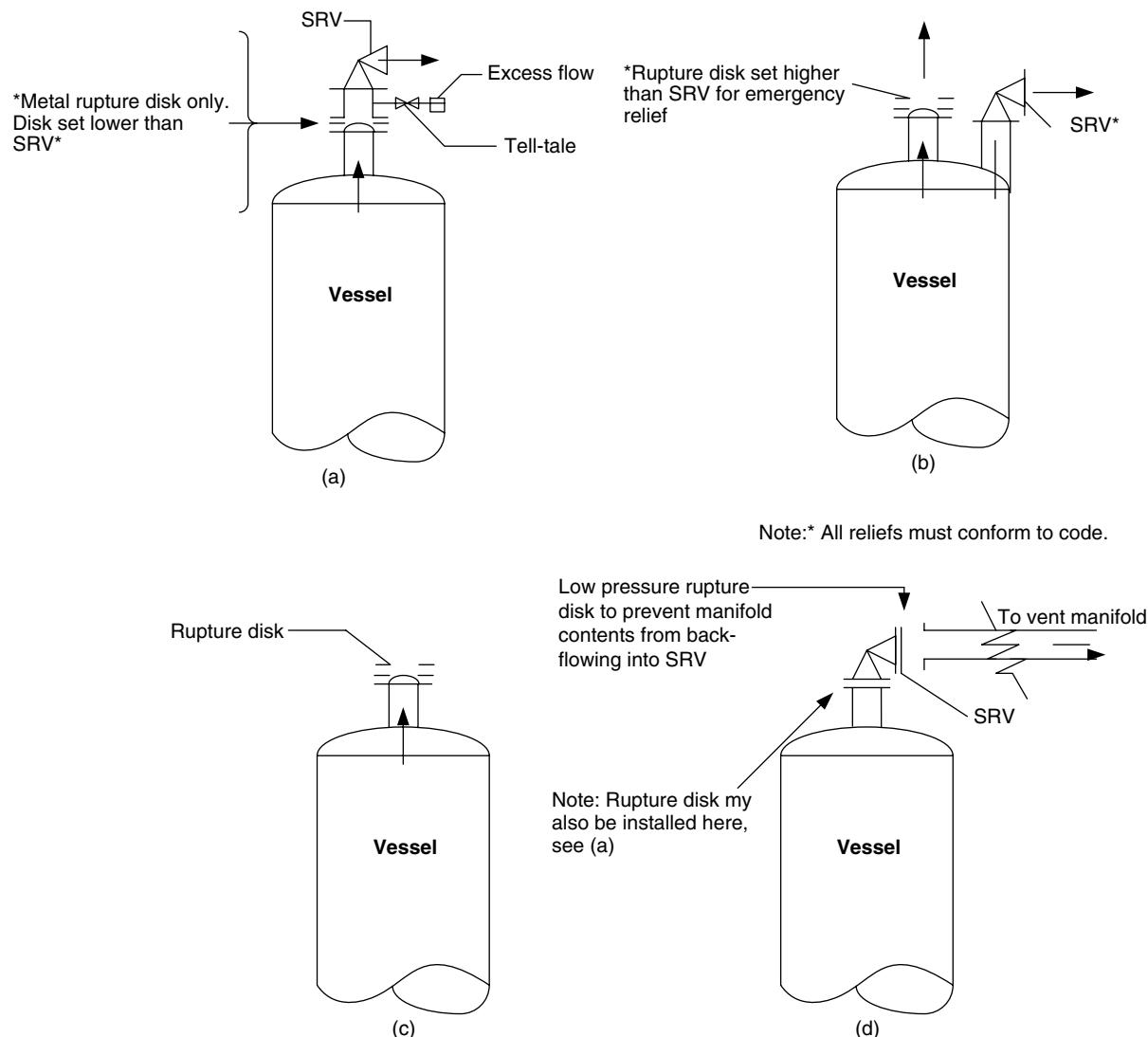
**Figure 9.9f** For pressure ratings of 15 psig or lower, subject to internal vacuum conditions, a vacuum support is required that is an integral part of the rupture disk and cannot be added in the field. (By permission from Falls Industries.)

## 9.7 UNFIRED PRESSURE VESSELS ONLY, BUT NOT FIRED OR UNFIRED STEAM BOILERS

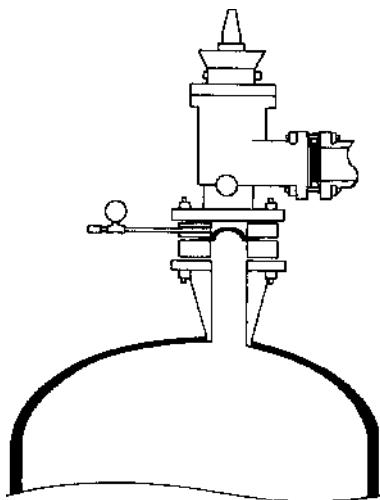
### NON-FIRE EXPOSURE

Single pressure-relief valve installation must be set to operate at a pressure not exceeding the MAWP of the vessel, [1] Par. UG-134, but may be set to operate at pressures below the MAWP. The device must prevent the internal pressure from rising more than 10% above the MAWP.

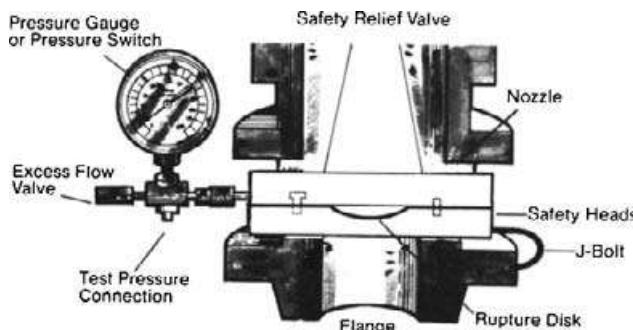
For multiple pressure-relief valves installation, if the required capacity is provided using more than one pressure-relieving device, (i) only one device must be set at or below the MAWP of the vessel, and (ii) the additional device(s) may be set to open at higher pressures, but in no case at a pressure any higher than 105% of the MAWP. The combination of relieving valves must prevent the pressure from rising more than 16% above the MAWP. See ASME [1] Par. UG-125C and G-1 and Par. UG-134a.



**Figure 9-10** Rupture disk installations.



**Figure 9-11** Safety valve and rupture disk installation using pressure rupturing disk on inlet to safety relief valve, and low pressure disk on valve discharge to protect against backflow/corrosion of fluid on valve discharge side, possibly discharge manifold. (By permission from Fike Metal Products Div., Fike Corporation, Inc.)



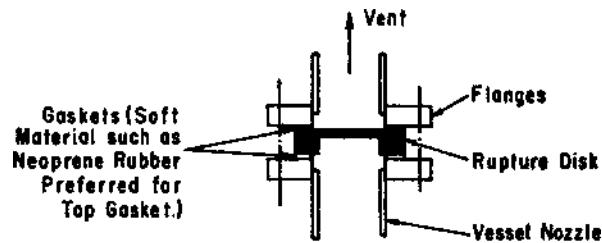
**Figure 9-12** Rupture disk mounted beneath a pressure-relieving spring-loaded valve. A reverse buckling® disk arrangement is often recommended here. (By permission from B.S. & B. Safety Systems, Inc.)

#### EXTERNAL FIRE OR HEAT EXPOSURE ONLY AND PROCESS RELIEF

Valves to protect against excessive internal pressures must be set to operate at a pressure not in excess of 110% of the MAWP of the vessel (ASME Par. UG-134b).

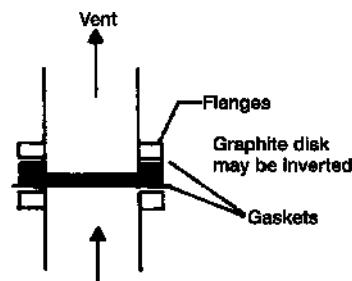
When valves are used to meet the requirements of both Par. UG-125(c) and UG-125c-(2), that is, both internal process pressure and external fire/heat requirements, the valve(s) must "be set to operate not over the MAWP of the vessel. For these conditions of the additional hazard of extreme fire or heat, supplemental" pressure-relieving devices must be installed to protect the vessel. The supplemental devices must be capable of preventing the pressures from rising more than 21% above the MAWP (*Note:* this is not the setting). The same pressure-relieving devices may be used to satisfy the capacity requirements of Par. UG-125c or C(1), and Par. UG-125c-(2) provided the pressure setting requirements of Par. UG-134(a) are met. See Par. (A) 1 and 2 above and see Figure 9-7a.

When pressure-relief devices are intended primarily for protection against overpressure due to external fire or heat, have no



Install Disk with Pressure Membrane up. When inverted, the Disk Bursts at about 65% Increase in Pressure. Disk Must be Positioned True Center of Vent Line and Nozzle. If Eccentric, Burst Characteristics Might Not Hold True.

**Figure 9-13a** Installation of graphite rupture disk. (Adapted by permission from Falls Industries, Inc.)



Inverted graphite disk bursts at higher pressure than with flat surface on top.

**Figure 9-13b** Inverted graphite disk bursts at higher pressure than with flat surface on top. (Adapted by permission from Falls Industries, Inc.)

permanent supply connection, and are used for storage at ambient temperature of non-refrigerated liquefied compressed gases, they are excluded from requirements of Par. UG-125c (1) and C (2), with specific provisions. See ASME code [1] for detailed references and conditions.

- Vessels operating completely filled with liquid must be equipped with liquid relief valves, unless otherwise protected (Par. UG-125-3(g)).
- Safety and safety relief valves for steam service should meet the requirements of ASME Par. UG-131(b), [1]. Note that the requirements for these valves are slightly different than for process type valves.

#### 9.8 RELIEVING CAPACITY OF COMBINATIONS OF SAFETY RELIEF VALVES AND RUPTURE DISKS OR NON-RECLOSURE DEVICES (REFERENCE ASME CODE, PAR. UG-127, U-132).

##### PRIMARY RELIEF

A single rupture disk can be used as the only overpressure protection on a vessel or system (Figure 9-10). The disk must be stamped by the manufacturer with the guaranteed bursting pressure at a specific temperature. The disk must rupture within  $\pm 5\%$  of its stamped bursting pressure at its specified burst temperature of operation. The expected burst temperature may need to be determined by calculation or extrapolation to be consistent with the selected pressure.

The set burst pressure should be selected to permit a sufficiently wide margin between it and the vessels used or design operating pressure and temperature to avoid premature failure due to fatigue or creep of metal or plastic coatings.

#### **SELECTED PORTIONS OF ASME PRESSURE VESSEL CODE (QUOTED BY PERMISSION [1])**

Section VIII, Division I Superscript = Footnote reference July 1, 1989 Edition in Code Figure No., for this text.

##### **Rupture Disk Devices [15], Par UG-127**

###### **1. General**

- a. Every rupture disk shall have a stamped bursting pressure within a manufacturing design range [11] at a specified disk temperature [16] and shall be marked with a lot number, and shall be guaranteed by its manufacturer to burst within 5% (plus or minus) of its stamped bursting pressure at the coincident disk temperature.

###### **2. Capacity rating**

- a. The calculated capacity rating of a rupture disk device shall not exceed a value based on the applicable theoretical formulas (see Par. UG-131) for the various media multiplied by  $K = \text{coefficient} = 0.62$ . The area A (square inches) in the theoretical formula shall be the minimum net area existing after burst [17].

###### **3. Application of rupture disks**

- a. A rupture disk device may be used as the sole pressure-relieving device on a vessel. *Note:* When rupture disk devices are used, it is recommended that the design pressure of the vessel be sufficiently above the intended operating pressure to provide sufficient margin between operating pressure and rupture disk bursting pressure to prevent premature failure of the rupture disk due to fatigue or creep. Application of rupture disk devices to liquid service should be carefully evaluated to assure that the design of the rupture disk device and the dynamic energy of the system on which it is installed will result in sufficient opening of the disk.

- b. A rupture disk device may be installed between a pressure-relief valve [18] and the vessel provided. (see Figure 9-10.)

- i. The combination of the spring loaded safety or safety relief valve and the rupture disk device is ample in capacity to meet the requirements of UG-133 (a) and (b).

- ii. The stamped capacity of a spring loaded safety or safety relief valve (nozzle type) when installed with a rupture disk device between the inlet of the valve and the vessel shall be multiplied by a factor of 0.80 of the rated relieving capacity of the valve alone, or alternatively, the capacity of such a combination shall be established in accordance with Par. 3 below.

- iii. The capacity of the combination of the rupture disk device and the spring loaded safety or safety relief valve may be established in accordance with the appropriate paragraphs of UG-132, Certification of Capacity of Safety Relief Valves in Combination with Non-reclosing Pressure Relief Devices.

- iv. The space between a rupture disk device and a safety or safety relief valve shall be provided with a pressure gauge, a try cock, free vent, or suitable telltale indicator. This arrangement permits detection of disk rupture or leakage [19].

- v. The opening [20] provided through the disk, after burst, is sufficient to permit a flow equal to the capacity of the valve (Par. 2 and 3 above) and there is no chance of

interference with proper functioning of the valve; but, in no case shall this area be less than 80% of the area of the inlet of the valve unless the capacity and functioning of the specific combination of rupture disk and valve been established by test in accordance with UG-132.

Note that in lieu of testing, Par. (b) 2 and (b) 3 above allow the use of a capacity factor of 0.80 as a multiplier on the stamped capacity of the spring loaded safety relief valve (nozzle type). Some manufacturers test specific valve/rupture disk combinations and determine the actual capacity factor for the combination, and then use this for the net capacity determination. See Figures 9-10, 9-11, 9-12, 9-13a and b.

- c. A rupture disk device may be installed on the outlet side [21] of a spring loaded safety relief valve which is opened by direct action of the pressure in the vessel provided (Figure 9-12).
- 1. The valve is so designed that it will not fail to open at its proper pressure setting regardless of any back pressure that can accumulate between the valve disk and the rupture disk. The space between the valve disk and rupture disk shall be vented or drained to prevent accumulation of pressure due to a small amount of leakage from the valve [22].
- 2. The valve is ample in capacity to meet the requirements of UG-133(a) and (b).
- 3. The stamped bursting pressure of the rupture disk at the coincident disk temperature plus any pressure in the outlet piping shall not exceed the design pressure of the outlet portion of the safety or safety relief valve and any pipe or fitting between the valve and the rupture disk device. However, in no case shall the stamped bursting pressure of the rupture disk at the coincident operating temperature plus any pressure in the outlet piping exceed the maximum allowable working pressure of the safety or safety relief valve.
- 4. The opening provided through the rupture disk device after breakage is sufficient to permit a flow equal to the rated capacity of the attached safety or safety relief valve without exceeding the allowable overpressure.
- 5. Any piping beyond the rupture disk cannot be obstructed by the rupture disk or fragment.
- 6. The contents of the vessel are clean fluids, free from gumming or clogging matter, so that accumulation in the space between the valve inlet and the rupture disk (or in any other outlet that may be provided) will not clog the outlet.
- 7. The bonnet of the safety relief valve shall be vented to prevent accumulation of pressure.

#### **FOOTNOTES TO ASME CODE**

- 47. The minimum net flow area is the calculated net area after a complete burst of the disk with appropriate allowance for any structural members which may reduce the net flow through the rupture disk device. The net flow area for sizing purposes shall not exceed the nominal pipe size area of the rupture disk device.
- 48. Use of a rupture disk device in combination with a safety or safety relief valve shall be carefully evaluated to ensure that the media being handled and the valve operational characteristics will result in pop action of the valve coincident with the bursting of the rupture disk.
- 49. Users are warned that a rupture disk will not burst at its design pressure if back pressure builds up in the space between the disk and the safety or safety relief valve which will occur

## 596 PROCESS SAFETY AND PRESSURE-RELIEVING DEVICES

- should leakage develop in the rupture disk due to corrosion or other cause.
50. This use of a rupture disk device in series with the safety or safety relief valve is permitted to minimize the loss by leakage through the valve of valuable or of noxious or otherwise hazardous materials and where a rupture disk alone or disk located on the inlet side of the valve is impracticable, or to prevent corrosive gases from a common discharge line from reaching the valve internals.
51. Users are warned that an ordinary spring loaded safety relief valve will not open at its set pressure if back pressure builds up in the space between the valve and rupture disk. A specially designed valve is required, such as a diaphragm valve or a valve equipped with a bellows above the disk.

(Source: Reprinted with ASME permission. ASME Pressure Vessel Code, Section VIII, Division I, UG-127, 1989 Edition, pp. 86–88.)

### 9.9 ESTABLISHING RELIEVING OR SET PRESSURES

The pressure at which the valve is expected to open (set pressure) is usually selected as high as possible consistent with the effect of possible high pressure on the process as well as the containing vessel. Some reactions have a rapid increase in temperature when pressure increases, and this may fix the maximum allowable working pressure. In other situations the pressure rise above operating pressure must be kept to some differential, and the safety valve must relieve at the peak value. A set pressure at the maximum value (whether MAWP of vessel or other, but insuring protection to the weakest part of the system) requires the smallest valve. Consult manufacturers for set pressure compensation (valve related) for temperatures  $>200^{\circ}\text{F}$  ( $>93.3^{\circ}\text{C}$ ).

When the pressure rise in a system is gradual and not “explosive” in nature, a safety or safety relief valve is the proper device, but when it is critical to completely depressurize a system or the rate of pressure increase might be expected to be rapid, then a rupture disk is the proper device. Properly designed, a pilot-operated valve may be selected after checking its performance with the manufacturer.

Often a system (a group of vessels not capable of being isolated from each other by block valves, or containing restriction to flow and release of pressure) may need a relief valve set reasonably close, set 15–20% when system is below 1000 psig; above, typically use 7–15% above as set criteria related to normal operating pressure to catch any pressure upswing. Then this may have a backup valve set higher (but within code) to handle further pressure increase. Or, the second device may be a rupture disk. It is not unusual to have two relief devices on the same equipment set at different pressures.

For situations where explosion may involve chemical liquid, vapor, or dust, it is generally advisable to obtain rate of pressure rise data and peak explosion pressure data in order to intelligently establish the design parameters. Such data are available [23–30]; however, it is important to evaluate whether the conditions are comparable between the systems when selecting the values for design. In general, the lower the setting for pressure relief, the lower will be the final internal peak pressure in the vessel. It is extremely important to realize that the higher the system pressure before relief, the higher will be the peak pressure attained in the vessel. In some difficult cases it may be advisable to set relief devices at two pressures, one lower than the other. Each must be designed for the conditions expected when it relieves, and one or all must satisfy code requirements or be more conservative than the code.

For pulsating service, the set pressure is usually set greater than the nominal 10% or 25 psig above the average operating pressure of the system in order to avoid unnecessary releases caused by surging pressure peaks, but still not exceeding the MAWP of the vessel/system. Careful analysis must be made of the proper set condition.

Safety relief valves are available for relieving or set pressures as low as 2, 10, and 20 psig, as well as higher pressures. Lower pressures are available on special order. Usually a more accurate relief is obtained from the higher pressures.

Safety relief valves are normally tested in the shop, or even on the equipment at atmospheric temperature. The set tolerances on the valves as manufactured are established by the code as discussed earlier. In order to recognize the difference between the test temperature and the actual operating temperature at actual relief, the corrections shown in Tables 9-1a and b are applied. An increase in temperature above design causes a reduction in valve set pressure due to the effects of temperature on the spring and body.

Testing of pressure-relieving spring-loaded valves at atmospheric temperature requires an adjustment in set pressure at ambient conditions to compensate for higher operating temperatures. For process services, see Table 9-1a and for saturated steam, see Table 9-1b.

#### SAFETY AND SAFETY RELIEF VALVES FOR STEAM SERVICE

Pressure-relieving devices in process plants for process and utility steam systems must conform to the requirements of ASME [1] Par. UG-131b. This is not necessarily satisfactory to meet the ASME Power Boiler Code for applications on power generating equipment.

**TABLE 9-1a Compensation Factors for Safety Relief Valves Between Atmospheric Test Temperatures and Actual Operating Temperature [31]**

Operating Temperature (°F)	Percent Increase in Set Pressure at Atmospheric Temperature
–450–200	None
201–450	2
451–900	3
901–1200	4

By permission from Teledyne Farris Engineering Corp., Cat. FE-316, p. 12.

**TABLE 9-1b Set Pressure Compensation for Saturated Steam Service Safety Relief Valves Between Atmospheric Test Temperature and Actual Operating Temperature**

Saturated Steam Pressure Set Pressure (psig)	% Increase in Spring Settling
10–100	2
101–300	3
301–1000	4
1001–3000	5

By permission from Teledyne Farris Engineering Corp., Cat. FE-316, p. 12.

Vessels or other pressure-containing equipment that operates filled with liquid must be provided with liquid-relief valves, unless protected otherwise [1]. Any liquid-relief valve must be at least 0.5 in. in pipe size. [1] Par UG-128 (see [32]).

## 9.10 SELECTION AND APPLICATION

### CAUSES OF SYSTEM OVERPRESSURE

Figure 9-14a, Operational Check Sheet [33], lists 16 possible causes of overpressure in a process system and Figure 9-14b shows a typical relief valve process data sheet. There are many others, and each system should be reviewed for its peculiarities. System evaluation is the heart of a realistic, safe, and yet economical overpressure protection installation on any single equipment or any group of equipment. Solving formulas with the wrong basis and/or data can be disastrous. The following should be reviewed:

1. the sources of possible overpressure
2. maximum overpressure possible from all sources

3. maximum rate of volume increase at the burst pressure, and temperature at this condition
4. length of duration of overpressure.

## 9.11 CAPACITY REQUIREMENTS EVALUATION FOR PROCESS OPERATION (NON-FIRE)

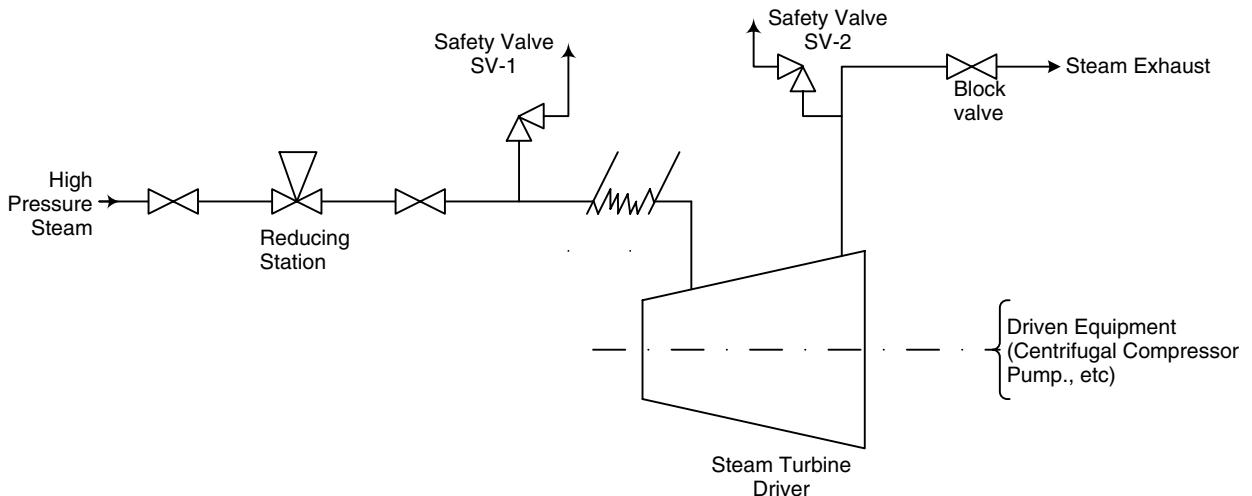
Each system and item of equipment should be examined for operational safety as set forth by specific plant area (and process fluids) requirements and the codes previously cited. The codes particularly [5a-d, 8-10] establish guides based on wide experience, and are sound requirements for design. Relief capacity is based on the most severe requirement of a system, including possible two-phase flow [13]. A system is generally equipment or groups of equipment which is isolated by shut-off valves. Within these isolated systems, a careful examination of the probable causes of overpressure is made [35]. Figures 9-15, 9-16, and 9-17 are suggested guides [33].

SAFETY VALVE DESIGN OPERATIONAL CHECK SHEET					
Date: _____				Job No.: _____	
Checked: _____				By: _____	
Vessel or System: Process Evaporator					
Design Pressure: 75 psig					
Allowable Pressure for Capacity Relief:			$75 + (75 + 10\%) = 82.5 \text{ psig}$		
Operating Conditions:		Fluid: _____	PDC sp.gr. 1.16	Mol Wt. 113.5	
				Temp. Oper.	
Physical Conditions:		Latent Heat 125 BTU/lb.		Corrosive	No
		Vessel Dia. 5 feet		$\times$ length	6 feet
		Insulation : Yes, 2"			
		Fire Control Measures: No Sprinkler Sys.			
Cause of Overpressure			Capacity Requirement, lb./hr.		
1. Failure Cool Water/Elect/Mechanical			-----		
2. Reflux and/or Condensing Failure			-----		
3. Entrance of a Highly Volatile Fuel			-----		
4. Vapor generation, external fire			---2,920---		
5. Excessive Operating Heat Inputs			---22,500---		
6. Accumulation of Non-Condensibles			-----		
7. Closed Outlets			---22,500 or less-----		
8. Failure of Automatic Controls/Instr.			---22,500 or less-----		
9. Internal Explosions (Use Rupture Disk)			-----		
10. Chemical Reaction/Run-a-Way			-----		
11. Two Phase Flow Conditions			-----		
12. Inadvertent opening valve into system			-----		
13. Check Valve failure			-----		
14. Cooling Fans failure			-----		
15. Heat Exchanger Tube rupture/failure			-----		
16. Circulating Pump failure			-----		
Causes that may occur simultaneously:			-----		
----- Any of the four considered may occur simultaneously -----			-----		
Probability of occurrence: --Closed outlet with a failure of automatic control resulting in excessive heat input. -----			-----		
Allowance to make: -----None-----			-----		
Relief capacity used for sizing valve(s) -----22,500 lb./hr.			-----		
Auxiliaries		Cause of overpressure		Capacity Requirement	
1. Exchangers		Split tube(s)		1482 lb. hr.	
		Thermal Vaporization		-----	
2. Pumps		Discharge Restriction		-----	
3. Length of Line		Thermal Vaporization		-----	

**Figure 9-14a** Safety valve design operational check sheet. (Adapted and added to by permission from N.E. Sylvander and D.L. Klatz, *Design and Construction of Pressure Relieving Systems*, Univ. of Michigan, Ann Arbor (1948). Six items of overpressure list above by E.E. Ludwig [34] and from API Rec. Practice 521 (1982).)

		RELIEF DEVICE PHILOSOPHY SHEET DOCUMENT/ITEM REFERENCE					
		EQUIPMENT No.:					
		DATE:	SHEET No.:		OF		
CHECKED BY:	MADE BY:				1		
					2		
DESIGN CODES:	VESSELS	EXCHANGERS		LINES	3		
					4		
OTHER REQUIREMENTS					5		
					6		
BASIS FOR CALCULATION:					7		
					8		
SET PRESSURE, psig:	:MAX. BACK PRESSURE	(a) BEFORE RELIEVING	(b) WHILE RELIEVING		9		
NORMAL CONDITIONS UNDER RELIEF DEVICE:				Calculated		10	
STATE:	TEMPERATURE, °F:	PRESSURE, psig:	POSSIBLE CAUSE?	FLUID RELIEVED	RELIEF RATE, lb/h	ORIFICE AREA, in <sup>2</sup>	11
							12
HAZARDS CONSIDERED							13
1. Outlets blocked							14
2. Control Valve malfunction							15
3. Machine trip/ overspeed/density change							16
4. Exchanger tube rupture							17
5. Power failure/ Voltage dip							18
6. Instrument air failure							19
7. Cooling failure							20
8. Reflux failure							21
9. Abnormal entry of volatile liquid							22
10. Loss of liquid level							23
11. Abnormal chemical reaction							24
12. Boxed in thermal expansion							25
13. External fire							26
14. (specify)							27
15. (specify)							28
16. (specify)							29
SELECTED DESIGN CASE:					30		
					31		
RELIEVED FLUID: STATE	DENSITY / MW:	TEMPERATURE:	Cp/Cv:		32		
COMPOSITION:					FLASHING:	33	
RELIEF RATE REQUIRED, lb/h:	ORIFICE SELECTED:	AREA, in <sup>2</sup> :	TYPE:		34		
ACTUAL CAPACITY, lb/h					35		
REMARKS/SKETCH					36		
					37		
					38		
					39		
					40		
					41		
					42		
					43		
					44		
					45		
					46		
Issue No:	1	Date	2	Date	3	Date	47
Made/Revised by							48
Checked by							49
Approved- Process							50
Approved- by							51

Figure 9-14b Relief Valve Process Data Sheet.



Safety Valve Required to Protect Reducing Station. Discharge Pressure in Case of Valve Failure. SV-1 is Set at Slightly Above Downstream Pressure of Reducing Station, and Protects All Equipment Operating at this Pressure on Steam Header.

Safety Valve SV-2 is Set to Protect Discharge Side of Turbine, as it is Not Designed to Withstand Inlet Steam Pressure on Exhaust Side.

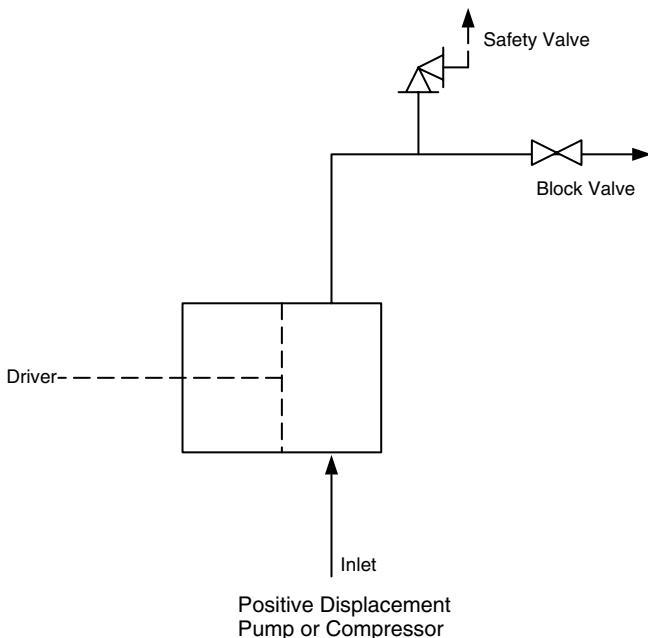
**Figure 9-15** Safety valve protecting specific equipment operating.

Capacities are calculated for conditions of temperature and pressure at actual state of discharge. Final discharge pressure is the set pressure plus overpressure.

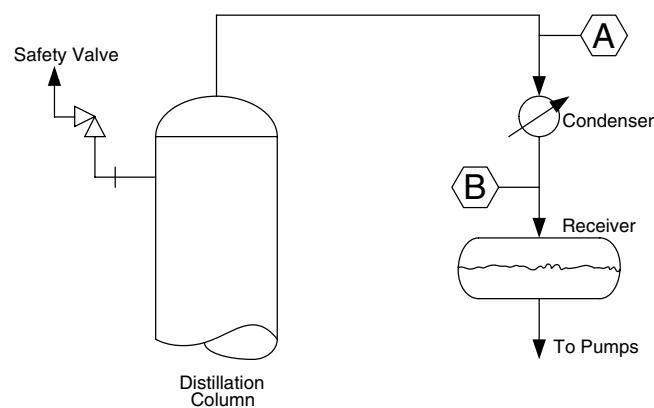
It must be emphasized, that the determination of the anticipated maximum overpressure volume at a specified pressure and temperature is vital to a proper protection of the process system. The safety relief calculations should be performed at the actual worst conditions of the system, for example, at the allowable accumulated pressure and its corresponding process temperature. These can be tedious and perhaps time-consuming calculations, but they

must not be “glossed” over but developed in a manner that accounts for the seriousness of the effort. *They must be documented carefully and preserved permanently.*

The situation is just as critical, if not more so, for runaway reactions or reaction conditions that are not adequately known. They should be researched or investigated by laboratory testing for possible runaway conditions and then the kinetic and heat/pressure rise calculations should be performed, even if some assumptions must be made to establish a basis. Refer to later paragraphs and the American Institute of Chemical Engineers Design Institute for Emergency Relief (AIChE/DIERS) [13]. At the time of a vessel or pressure/vacuum system failure, the calculations for the effected pressure-relief devices are always reviewed by plant management and the Occupational Safety and Health Administration (OSHA)



**Figure 9-16** Safety valve in positive displacement system.



This is acceptable as there is no block valve isolating any item. If a block valve installed at A, a safety valve would be required to protect condenser and receiver. If an additional block valve installed at B, a safety valve would be required for the condenser and also for the receiver.

**Figure 9-17** System protected by safety valve on column.

inspectors. A few notes on causes of process system failures are noted below, with additional comments in API-521 [5a-d], [9].

**Failure of cooling water:** Assume all cooling media fail; determine relief capacity for the total vapors entering the vessel, including recycle streams (see [36] and [8]).

**Reflux failure:** (a) At the top of a distillation column, the capacity is total overhead vapor [8]; (b) when source of heat is in feed stream, the capacity is vapor quantity calculated in immediate feed zone [36]; and (c) when reboilers supply heat to system, the capacity is feed plus reboil vapors [36]. Each situation must be examined carefully.

**Blocked outlets on vessels:** (a) For liquid, the capacity is the maximum pump-in rate. (b) For liquid-vapor system, the capacity is total entering vapor plus any generated in the vessel [8].

**Blocked outlets and inlets:** For systems, lines or vessels capable of being filled with liquid and heated by the sun or process heat require thermal relief to accommodate the liquid expansion (assuming vaporization is negligible).

**Instrument failure:** Assume instrument control valves freeze or fail in open position (or closed, which ever is worse), determine the capacity for relief based on flows, temperatures, or pressures possible under these circumstances. The judicious selection of instrument failure sequence may eliminate or greatly reduce relief valve requirements.

**Equipment Failure:** Pumps, tubes in heat exchangers and furnaces, turbine drivers and governor, and compressor cylinder valves are examples of equipment which might fail and cause overpressure in the process. If an exchanger tube splits or develops a leak, a high pressure fluid will enter the low side, over pressuring either the shell or the channels and associated system as the case may be.

**Vacuum:** (a) Removal of liquid or vapor at greater rate than entering the vessel, the capacity is determined by the volume displaced; (b) Injecting cold liquid into hot (steamed out) vessel,

the condensing steam will create vacuum, and must be relieved. Capacity is equivalent to vapor condensed.

**In-breathing and Out-breathing Pump In and Out:** See Section 9.42.

## INSTALLATION

Never place a block valve on the discharge side of a pressure-relief device of any kind, for exception see [1] Par. U-135 (e).

Never place a block valve on the inlet side of a pressure-relief device of any kind, unless it conforms to the code practice for rupture disks or locking devices. See [1] Par. UG-135 (e) and Appendix M, ASME code.

Note that the intent of the ASME code is to ensure that under those circumstances where a pressure-relieving device can be isolated by a block valve from its pressure, or its discharge, a responsible individual locks and unlocks the block valve to the safe open position and that this individual remains at the block valve the entire time the block valve is closed.

Safety (relief, or safety relief) valves are used for set pressures from 10 to 10,000 psig (0.69–690 bar) and even higher. At the low pressures, the sensitivity to relieving pressure is not always as good as is required for some processes, and for this reason most valve installations start at 15–20 psig (1.03–1.38 bar).

Figures 9-10 and 9-18 illustrate a few typical safety valve installations. Care must be shown in designing any manifold discharge headers collecting the vents from several valves. Sharp bends are to be avoided. Often two or more collection systems are used in order to avoid discharging a high pressure valve into the same header with a low pressure valve. The simultaneous discharge of both valves might create too great a back pressure on the low pressure valve, unless adequate arrangement has been made in the valve design and selection. The balanced safety relief valve can overcome most of the problems of this system.

Whenever possible the individual installation of valves is preferred, and these should be connected directly to the vessel or

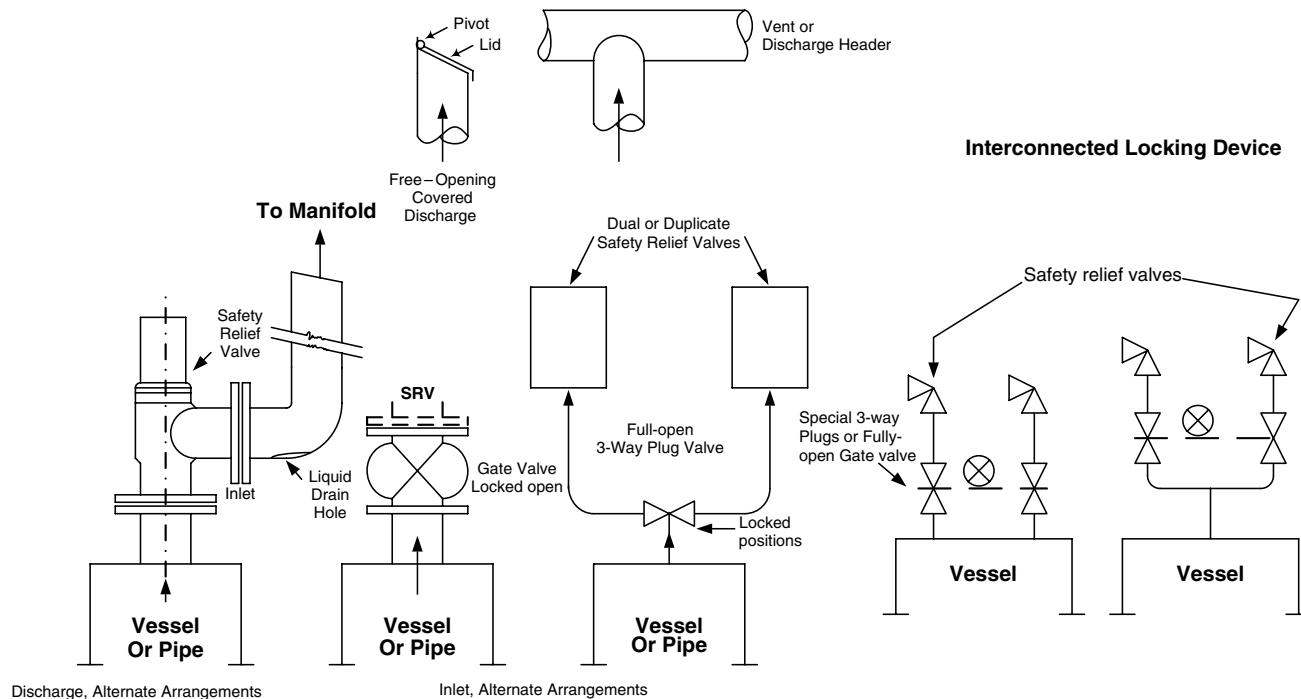


Figure 9-18 Safety relief valve installations.

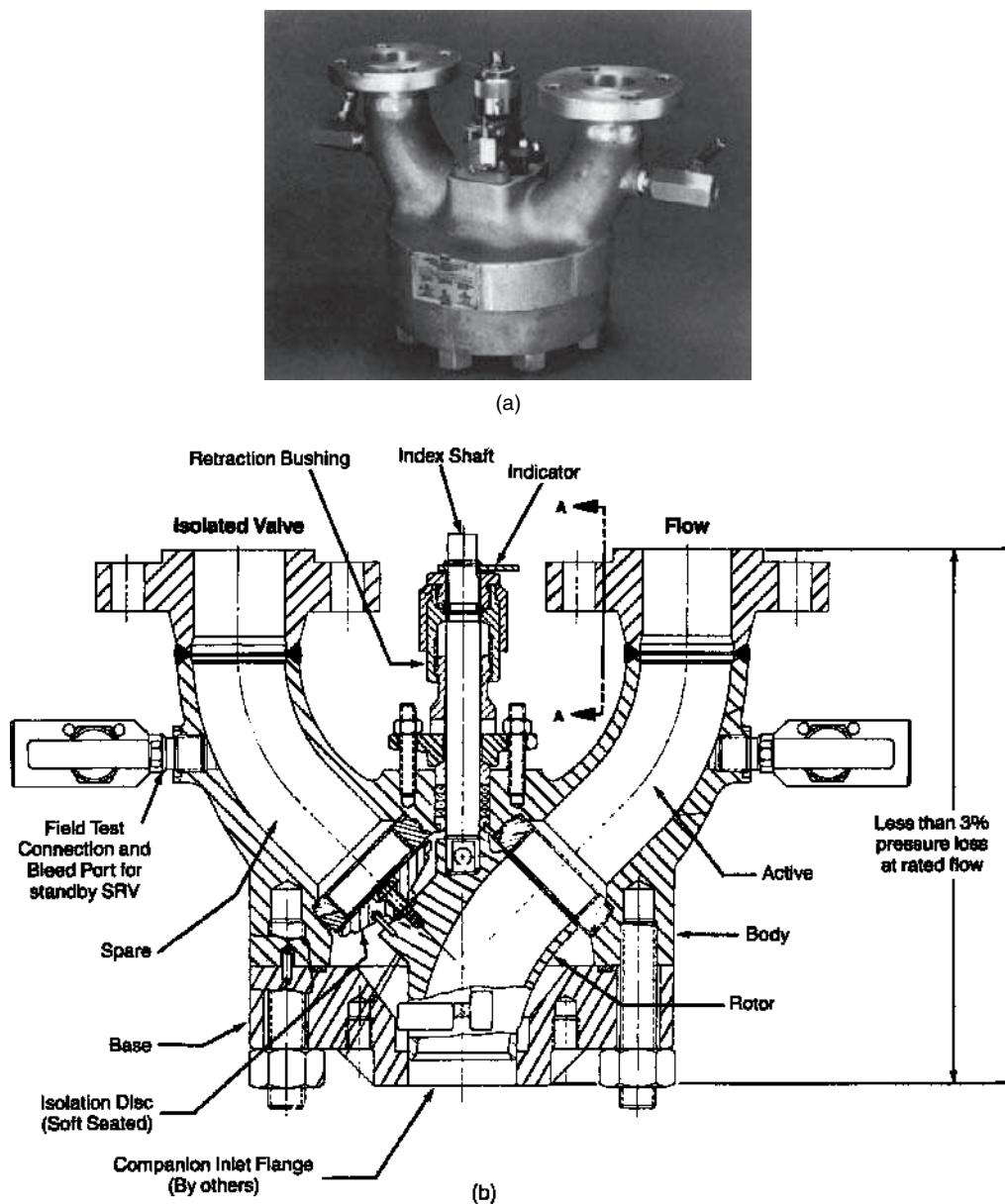
pipe line [1, 37]. If a block-type valve is considered necessary for a single valve installation, it must be of the full open type, and locked open with the key in responsible hands, as stated earlier.

Dual installations are frequently made in continuous processes, to allow switching from one valve to another without shutdown of the pressure system. A special three-way plug of full open type is installed directly on the vessel, and the safety valves are attached to it with short piping (Figure 9-18). The three-way valve insures that one side of the safety valve pair is always connected to the vessel, as this pattern valve does not have a blind point during switching (Figure 9-19). One of the important justifications for this dual arrangement is that safety relief valves may leak on reseating after discharging. This leak may be caused by a solid particle lodged on the seat. This valve can be removed for repair and cleaning after the process has been switched to the second valve. Each valve must be

capable of relieving the full process requirements. Multiple valves may also be individually installed separately on a vessel.

Figure 9-19 illustrates a newer approach at simplifying the dual safety relief valve installation, ASME Section VIII, Division 1, UG-135(b) [1] and API RP-520, Part II Conformance [5]. Note that the safety relief valves (SRV) are mounted on top of each of one dual vertical connections and are bubble tight. Also see cross-section view. The flow  $C_v$  values for each size device are available from the manufacturer.

The Anderson, Greenwood & Co. (AGCO) Safety Selector Valve body houses a uniquely designed switching mechanism. The internal rotor smoothly diverts flow to either safety relief valve. Conventional direct spring-operated valves or pilot-operated valves may be used. The inactive valve is totally isolated by external adjustment. To begin the switchover, the retraction bushing is rotated to its stop. This separates the isolation disk from the standby



**Figure 9-19** Safety selector valve for dual relief valve installation with switching. (By permission from Anderson, Greenwood and Co. © AGGO.)

valve channel and temporarily "floats" it in the main valve cavity. The index shaft is then rotated 180° to the alternate channel. The retraction bushing is then returned to its original position, securely seating the isolation disk beneath the valve taken out of service. A red pointer indicates which valve is in service and double padlocking provisions allow the safety selector valve to be locked in either safety relief valve position. The padlocks or car seals can only be installed with the internals in the proper position. No special tools are necessary for switching.

The alternate concept which has been in use for many years is to fabricate or purchase a Tee connection upon which the two-safety relief valves can be mounted on top of their full-port plug or gate valve with required locking lugs.

Rupture disks are often used in conjunction with safety valves as shown in Figures 9-10, 9-11, 9-12, and 9-18.

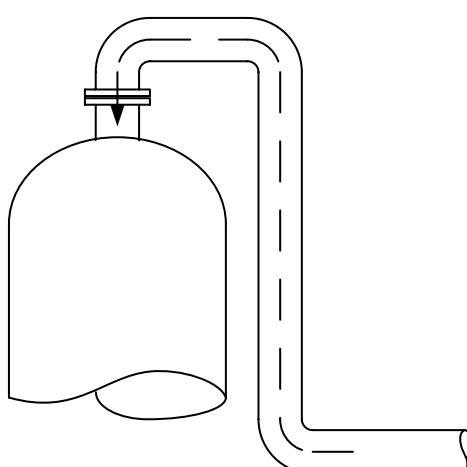
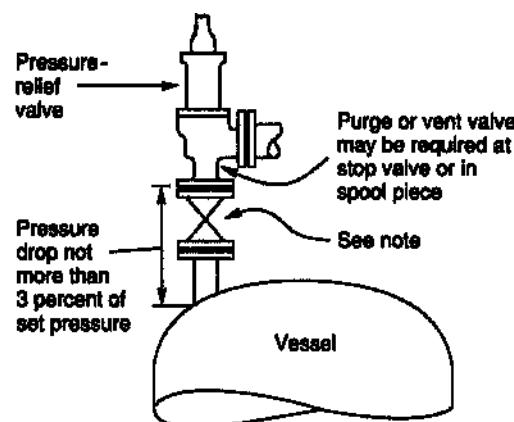
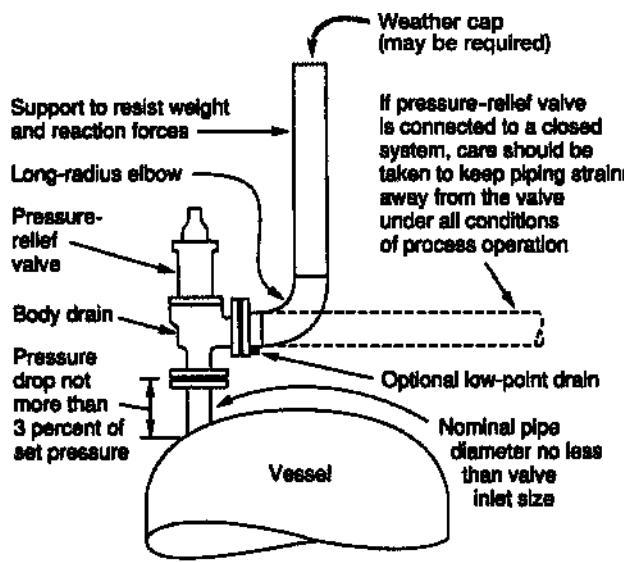
Inlet piping is held to a minimum, with the safety device preferably mounted directly on the equipment and with the total system pressure drop to pressure-relief valve inlet not

exceeding 3% of the set pressure in psig, of maximum relief flowing conditions [8]. To conform to the code (see ASME code, Section VIII, Division 1-UG-127 [1]) avoid high inlet pressure drop and possible valve chatter:

1. Never make pipe connection smaller than valve or disk inlet.
2. Keep friction pressure drop very low, not over 1–2% of allowable pressure for capacity relief [1, 5, 8, 33, 37].
3. Velocity head loss should be low, not over 2% of allowable pressure for capacity relief [33].

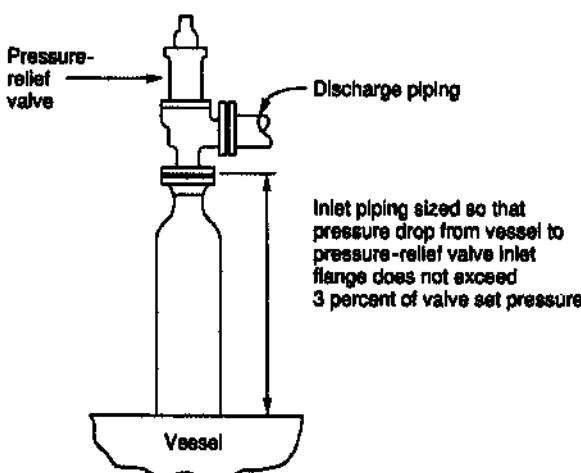
Discharge piping must be sized for low pressure drop at maximum flow not only from anyone valve, but for the combined flow possibilities in the discharge collection manifold all the way to the vent release point, whether it be a flare, incinerator, absorber, or other arrangement [9]. (Figures 9-20a–f).

Conventional safety relief valves, as usually installed, produce unsatisfactory performance when variable back pressure



**Note:** The stop valve must have a full port area greater than or equal to the inlet size of the pressure-relief valve. The stop valve should be used only as permitted by the applicable codes.

**Typical pressure-relief valve with a stop valve**



**Figure 9-20a** Recommended API-520 piping for safety relief valve installations. (Reprinted by permission from American Petroleum Institute, *Sizing, Selection and Installation of Pressure Relieving Devices in Refineries*, Part II-Installation, API RP-520, 3rd ed., Nov 1988.)

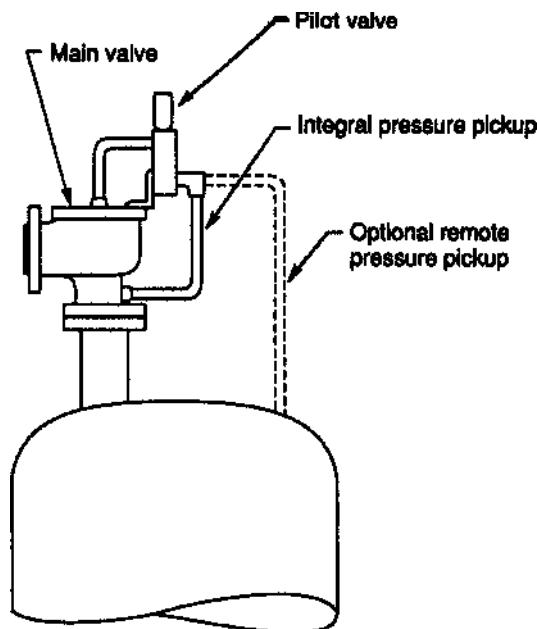


Figure 9-20b Typical pilot-operated pressure-relief valve installation.

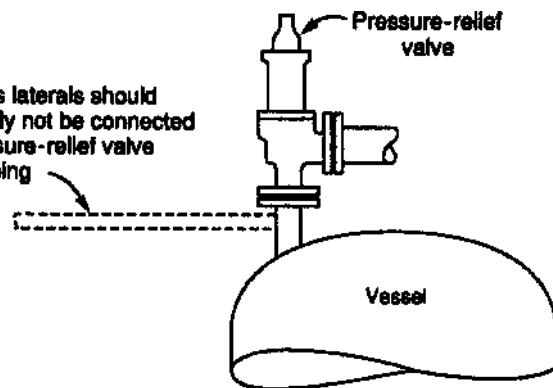


Figure 9-20d Typical installation avoiding process laterals connected to pressure-relief valve inlet piping.

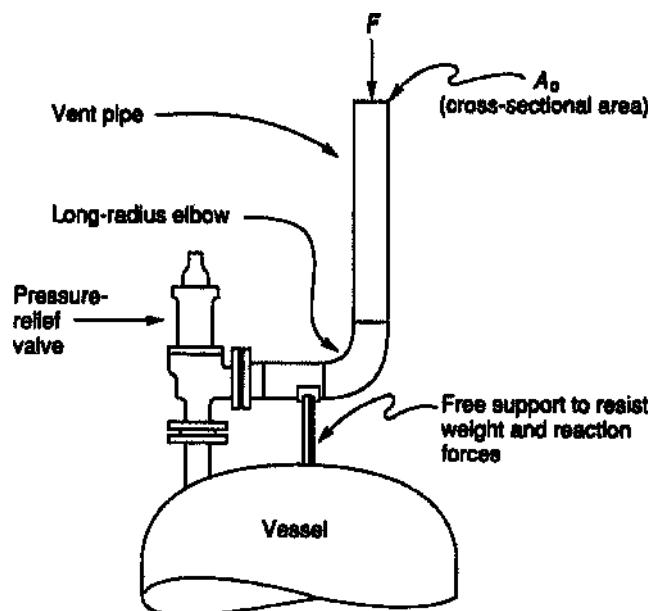


Figure 9-20c Typical pressure-relief valve installation with vent pipe.

exists [5, 8] (see Figure 9-6a). Additionally, the same variable back pressure forces affect the set pressure release. At low back pressures, the valve flow falls rapidly as compared with the flow for a theoretical nozzle. (See Figures 9-19 and 9-20 [5a]).

For conventional valves, pressure drop or variations in back pressure should not exceed 10% of set pressure. Because most process safety valves are sized for critical pressure conditions, the piping must accommodate the capacity required for valve relief and not have the pressure at the end of vent or manifold exceed the critical pressure. Designing for pressure from 30 to 40% of critical

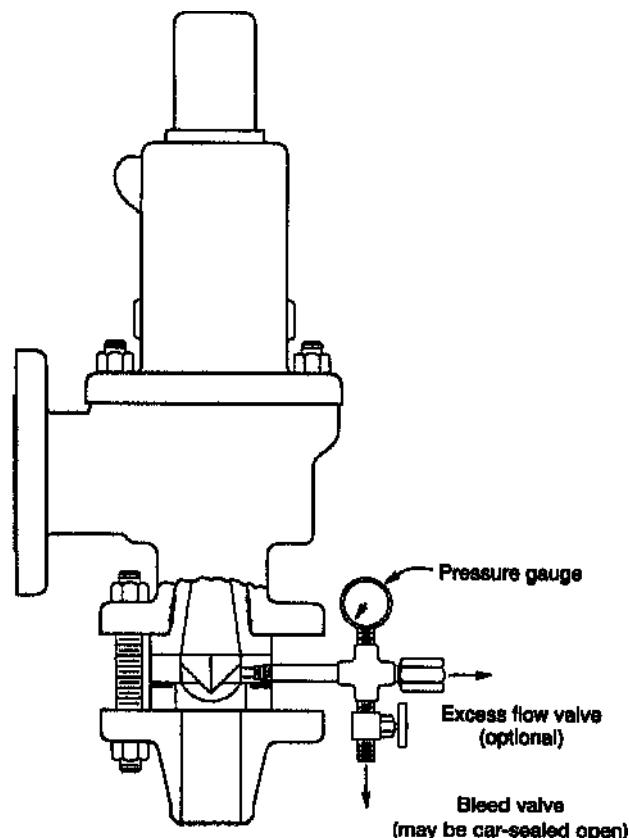
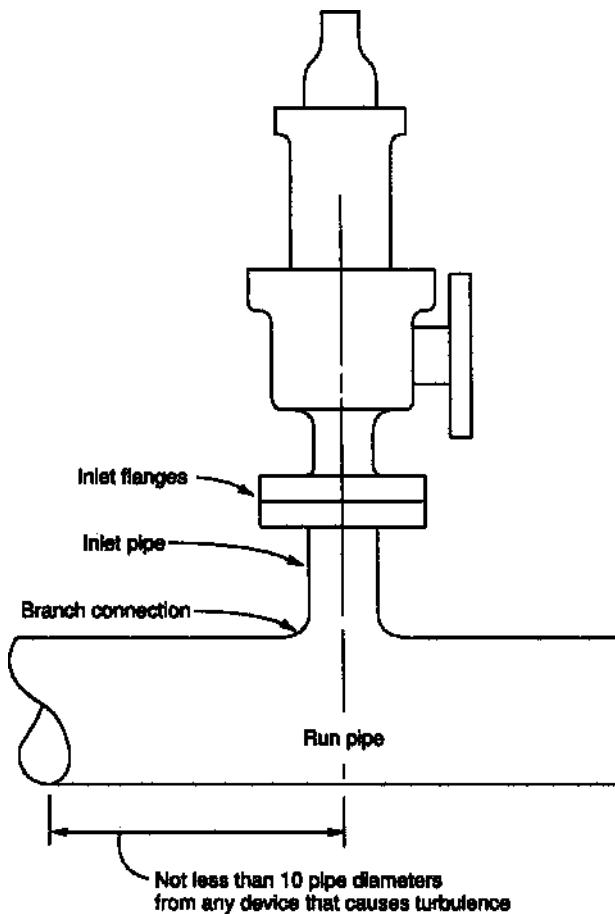


Figure 9-20e Typical rupture disk assembly installed in combination with a pressure-relief valve.

with balanced valves yields smaller pipes yet allows proper functioning of the valve. The discharge line size must not be smaller than the valve discharge. Check the manufacturer for valve performance under particular conditions, especially with balanced valves which can handle 70–80% of set pressure as back pressure.

For non-critical flow the maximum back pressure must be set and pressure drop calculated by the usual friction equations.

When process conditions permit, the low pressure range is handled by bursting disks which will relieve down to 2 psig. These



**Figure 9-20f** Typical installation avoiding excessive turbulence at pressure-relief valve inlet.

disks are also used up to 100,000 psig and above. The rupture pressures and manufacturing ranges of metal disks are given in Tables 9-2 and 9-3. For non-metallic materials such as graphite, bursting pressures are available from the manufacturers. From these manufacturing tolerances it can be seen that the relation of disk bursting pressure to the required relieving pressure must be carefully considered. Manufacturing practice is to furnish a disk, which will burst within a range of pressures and tolerances, and whose rated pressure is the result of bursting tests of representative sample disks which burst within the range specified. The engineer should specify only ASME code certified disks. It is not possible to obtain a disk for the usual process application set to burst at a given pressure, as is the relieving pressure of a safety valve. An increase in temperature above the disk rating temperature (72° F) decreases the bursting pressure 70–90% depending upon the metal and temperature (Tables 9-4a and b).

The minimum rupture pressure of disks of various metals and combinations varies so widely that individual manufacturer must be consulted.

For the usual installation, the rupture disk is installed as a single item between special flanges which hold the edges securely and prevent pulling and leakage. If the system is subjected to vacuum or pressure surges, a vacuum support must be added to prevent collapse of the sealing disk. The flanges which hold the disk may be slip-on, weld neck, and so on. Disks to fit screwed and union-type connections are also available (see Figures 9-8 and 9-9).

The service life of a rupture disk is difficult to predict, since corrosion, cycling pressures, temperature, and other process conditions can all affect the useful life and cause premature failure. A graphite-type disk is shown in Figure 9-9. In some processes, it is safer to replace disks on a schedule after the life factor has been established, as a planned shutdown is certainly less costly than an emergency one.

Rupture disks are often placed below a safety valve to prevent corrosive, tarring, or other material from entering the valve nozzle. Only disks which do not disintegrate when they burst (Figures 9-10, 9-11, 9-12, and 9-18) can be used below a safety valve, as foreign pieces which enter the valve might render it useless. This is acceptable to certain code applications [1]. These disks are also used to provide secondary relief when in parallel with safety valves set at lower pressures. They can also be installed on the discharge of a safety valve to prevent loss of hazardous vapors, but caution should be used in any serious situation.

## 9.12 SELECTION FEATURES: SAFETY, SAFETY RELIEF VALVES, AND RUPTURE DISKS

Referring to the description and definitions in the introduction for this chapter, it is important to recognize that in order to accomplish the required pressure relief the proper selection and application of device type is essential.

**Safety valve:** This is normally used for steam service, but suitable for gases or vapors. When used in steam generation and process steam service the valves conform to the *ASME Power Boiler Code* as well as the *ASME Pressure Vessel Code*, Section VIII, and are tested at capacity by the National Board of Boiler and Pressure Vessel Inspectors. This type of valve characteristically "pops" full open and remains open as long as the overpressure exists.

**Relief Valve:** This is normally selected for liquid relief service such as hydraulic systems, fire and liquid pumps, marine services, liquefied gases, and other total liquid applications. The valve characteristically opens on overpressure to relieve its rated capacity, and then reseats.

**Safety Relief Valve:** This is normally selected for vapors and gases as may be found in all types of industrial processes. Characteristically, this valve will open only enough to allow the pressure to drop below the set pressure, and then it will reseat until additional overpressure develops. If the pressure persists or increases, then the valve will remain open or increase its opening up to the maximum design, but as the pressure falls the valve follows by closing down until it is fully reseated. However, as in any installation of any "safety" type valve, the valve may not reseat completely gas tight. In such cases, it may be necessary to switch to a stand-by valve and remove the leaking valve for repair (see Figures 9-7a and b).

**Special Valves:** Because of the difficult and special sealing requirements of some fluids such as chlorine and Dowtherm, special valves have been developed to handle the requirements.

**Vacuum Relief and Combined Pressure-Vacuum Relief for Low Pressure Conditions:** This is normally used for low pressures such as 1 ounce water to 1.5 psig above atmospheric by special spring or dead weight loading; and for vacuum protection such as 0.5 psi below atmospheric. Usually these conditions are encountered in large process – that is, processing of crude oil, ammonia, and so on – storage tanks. See Section 9.42.

**Rupture disks:** This is used for low as well as high pressure protection of vessels and pipelines where sudden and total release of overpressure is required. Once the disk has ruptured, the process system is exposed to the environment of the back pressure of the discharge system, whether atmospheric or other. The process

**TABLE 9-2 Typical Prebulged Solid Metal Disk Manufacturing Ranges and Tolerances at 72° F**

Specified Burst Pressure Rating (psig)	Manufacturing Range (%)		Rated (Stamped) Burst Tolerance (%)
	Under	Over	
2–5	−40	+40	±25
6–8	−40	+40	±20
9–12	−30	+30	±15
13–14	−10	+20	±10
15–19	−10	+20	±5
20–50	−4	+14	±5
51–100	−4	+10	±5
101–500	−4	+7	±5
>501	−3	+6	±5

**Note:**

1. Special reduced manufacturing ranges can be obtained for the STD prebulged metal disk;  $\frac{1}{4}$ ,  $\frac{1}{2}$ , and  $\frac{3}{4}$  ranges are available upon request. Please consult your representative or the factory for additional information.
2. Burst tolerances are the maximum expected variation from the disk's rated (stamped) burst pressure.
3. Standard-type rupture disk comply with ASME code requirements.

**Manufacturing Range**

The manufacturing range is defined as the allowable pressure range within which a rupture disk is rated. It is based upon the customer-specified burst pressure. The manufacturing ranges for Continental's standard rupture disk are outlined in this table.

**Burst Tolerance**

After the disk has been manufactured and tested, it is stamped with the rated burst pressure. The rated (stamped) burst pressure is established by bursting a minimum of two disks and averaging the pressures at which the disks burst. This average is the rated (stamped) burst pressure of the disk. Standard rupture disks above 15 psig at 72° F are provided with a burst tolerance of ±5% of the rated (stamped) burst pressure. This is in accordance with the ASME code. Burst tolerances for disks below 15 psig at 72° F are outlined in this table. Burst tolerance applies only to the rated (stamped) burst pressure of the disk. Burst certificates are provided with each disk lot.

By permission from Continental Disk Corporation, Catalog STD-1184.

system is depressurized and the disk must be replaced before the process can be restarted. Typically, the types of disks available are as follows.

1. Solid metal rupture disk (Figure 9-8). This is the original type of rupture device, available in various metals and non-metals. It should normally not be used for operating pressures greater than 70% of rupture pressure in a non-corrosive environment. The metal disks are designed with a domed or hemispherical shape, with pressure on the concave side. As the pressure internally increases, the metal wall thins as the metal stretches to achieve a smaller radius of curvature. After the wall has thinned sufficiently, it will burst to relieve the pressure and tension loading on the metal. The accuracy of metal disks is ±5%, except for the Reverse Buckling Disk Assembly, which is ±2%. The usual recommended maximum operating temperatures for metal rupture disks is given in Tables 9-4a and b (also see Figures 9-8g and n).
- a. **Solid metal disk with vacuum support:** When vacuum can occur internally in the system, or when external pressure on the convex side of the disk can be greater than the pressure on the concave side of the disk, a vacuum support is necessary to prevent reversal of the disk, Figure 9-8D.

- b. **Solid metal disk with rating near minimum for size:** With a rating pressure near the minimum available for the size and material of construction, a special thin disk is attached to the lower and possibly the upper sides of the rupture disk to ensure freedom from deformity caused by the condition of the disk holding surfaces, Figure 9-8e. There are several versions of what to include under such conditions; therefore, it is advisable to clearly explain the installation conditions and application to the manufacturer.
- c. **Composite rupture disk:** This type consists of a metal disk (not necessarily solid, it may have slots) protected by an inner and/or an outer membrane seal, Figure 9-8e. There are several possible arrangements, including vacuum support, as for the styles of paragraph (b) above. This general class has the same use-rating limitation as for the solid disk.

- d. **Reverse acting or buckling disk assembly:** This design allows the disk to be operated in a system at up to 90% of its rated burst pressure. The pressure is operating on the convex side of the disk, and when bursting pressure is reached the disk being in compression reverses with a snap action at which time the four knife edges (Figure 9-8g) cut the metal and it clearly folds back without fragmentation. There is another version of the same concept of reverse buckling, but it uses a pre-scored disk and thereby omits the knife blades. These

**TABLE 9-3 Typical Metal Disk (Single) Bursting Pressures at 72° F Using Different Metals**

Size (in.)	Disk Minimum Burst Pressure (psig) (Without Liners)					
	Alum	Silver	Nickel	Monel	Inconel	316SS
1/4	160	450	600	700	1120	1550
1/2	65	220	300	350	560	760
1	29	120	150	180	250	420
1 1/2	22	80	100	116	160	275
2	13	48	60	70	110	150
3	10	35	45	50	80	117
4	7	26	35	40	70	90
6	5	20	25	30	47	62
8	4	15	20	23	34	51
10	4	—	16	17	30	43
12	3	—	13	15	25	36
14	3	—	11	13	21	31
16	3	—	10	12	19	28
18	3	—	9	11	17	24
20	3	—	8	9	16	22
24	3	—	—	—	—	—
30	—	—	—	—	—	—
36	—	—	—	—	—	—

(-) = Consult factory

\* Special designs of some manufacturers may exceed 150,000 psi for small sizes. The pressures listed are generally typical but certainly not the only ones available for the size shown.

*Note:*

1. Maximum burst pressure depends upon disk size and application temperature. Pressures to 80,000\* psig are available.
2. Other materials and sizes are available upon request.
3. Other liner materials are available upon request. Minimum burst pressures will change with change in liner material.
4. For larger sizes or sizes not shown, consult your representative, or the factory.

Courtesy of Continental Disk Corp., Bul. 1184, pp. 4–5.

**TABLE 9-4a Typical Recommended Maximum Temperatures for Metals Used in Disks**

Metals	Maximum Temperature (°F)
Aluminum	250–260
Silver	250–260
Nickel	750–800
Monel	800
Inconel	900–1000
316 Stainless Steel	900

*Source:* Various manufacturers' technical catalogs.

- types of disks do not need vacuum supports, unless there is unusually high differential pressure across the disk.
2. Graphite rupture disk (Figure 9-9). There are special designs of disks and disk assemblies for specific applications, and the manufacturer should be consulted for his recommendation. Disks are available for pressure service, pressure–vacuum applications, high temperature conditions, and close tolerance bursting conditions. The bursting accuracy of most designs is  $\pm 5\%$  for rated pressures above 15 psig and  $\pm 0.75\%$  for rated pressures 14 psig and below. It should be noted that these ratings are not affected by temperature up to 300° F.

A new concept in graphite disks includes addition of a fluorocarbon film barrier between the process and the disk, and is termed

**TABLE 9-4b Typical Recommended Maximum Temperatures for Linings and Coatings with Metals Used with Disks**

Metal	Maximum Temperature (°F)
Teflon® FEP Plastic	400
Polyvinylchloride	150–180
Lead	250

*Source:* Various manufacturers' technical catalogs.

a “duplex disk.” These disks are suitable for temperatures to 392° F, with accuracies as just mentioned.

Graphite disks are normally used in corrosive services and/or high temperature situations where metal wall thickness and corrosion rates make the metal units impractical because of unpredictable life cycles. The disks are available down to  $1 \pm 0.75$  psig, and are not affected by fatigue cracking. An interesting feature is the use of standard ASA (ANSI) flanges rather than special flanges (see Figure 9-9c).

It is important to recognize here also that once the disk bursts the system is depressured, and there will be fragments of graphite blown out with the venting system. Special discharge designs are often used to prevent plugging of discharge pipe and fragments from being sprayed into the surrounding environment.

### **9.13 CALCULATIONS OF RELIEVING AREAS: SAFETY AND RELIEF VALVES**

References to the ASME code [1] and the API code [5, 8] are recommended in order that the design engineer may be thoroughly aware of the many details and special situations that must be recognized in the final sizing and selection of a pressure-relieving device. All details of these codes cannot be repeated here; however, the usually important requirements are included for the typical chemical and petrochemical application for the guidance of the engineer.

Before performing sizing/design of relief valves calculations, a thorough examination of the possible causes and flow conditions of temperature and pressure should be determined. From this list, select the most probable and perhaps the worst case possibility and establish it as a design basis, Figures 9-14a and b [38].

When the possibilities of internal explosion or a runaway chemical reaction exists, or are even suspected, they must also be rigorously examined and calculations performed to establish the magnitude of the flow, pressure, and temperature problems. Select the worst condition and plan to provide for its proper release to prevent rupture of equipment. This latter situation can only be handled by application of rupture disks and/or remote sensing and predetermined rupture of the disks (see Figures 9-5a, 9-8k and 9-8l) or remote sensing and application of quenching of the reaction/developing explosive condition by automatic process action and/or commercial application of quenching medium. See later discussion in Section 9.52.

### **9.14 STANDARD PRESSURE-RELIEF VALVES – RELIEF AREA DISCHARGE OPENINGS**

The “orifice” area of these devices (see table below) is at the outlet end of the SRV nozzle through which the discharging vapor/gases/liquids must pass. These values are identified in industry as: (valve body inlet size, in.) × (orifice letter) × (valve body outlet size, in.). For example, a valve would be designated 3E4.

The standard orifice area designations are (also refer to mechanical illustrations of valves, previously shown in this chapter):

Orifice letter	D	E	F	G	H	J
Area, in. <sup>2</sup>	0.11	0.196	0.307	0.503	0.785	1.287
Area, cm <sup>2</sup>	0.71	1.27	1.98	3.25	5.06	8.30
Orifice letter	K	L	M	N	P	Q
Area, in. <sup>2</sup>	1.838	2.853	3.600	4.340	6.380	11.05
Area, cm <sup>2</sup>	11.85	18.4	23.2	28.0	41.20	71.30
Orifice letter	R	T	V*	W	W2*	X*
Area, in. <sup>2</sup>	16.0	26.0	42.19	57.26	93.6	101.8
Area, cm <sup>2</sup>	103.20	167.70		60.75		
Orifice letter	Y*	Z*	Z <sub>2</sub> *	AA	BB	BB2
Area, in. <sup>2</sup>	128.8	159.0	–	–	–	–
	82.68	90.95	108.86	136.69	168.74	185.00

\* Note: These letters and orifice areas are not consistent for these large orifices between various manufacturers. Some sizes go to 185 in.<sup>2</sup>, which is a very large valve. When two valves are shown, they represent two different published values by manufacturers.

### **9.15 SIZING SAFETY RELIEF TYPE DEVICES FOR REQUIRED FLOW AREA AT TIME OF RELIEF\***

Before initiating any calculations, it is necessary to establish the general category of the pressure-relief valve being considered. This section covers conventional and balanced spring-loaded types.

Given the rate of fluid flow to be relieved, the usual procedure is to first calculate the minimum area required in the valve orifice for the conditions contained in one of the following equations. In the case of steam, air, or water, the selection of an orifice may be made directly from the capacity tables if so desired. In either case, the second step is to select the specific type of valve that meets the pressure and temperature requirements.

General equations are given first to identify the basic terms which correlate with ASME Pressure Vessel Code, Section VIII.

It is recommended that computations of relieving loads avoid cascading of safety factors or multiple contingencies beyond the reasonable flow required to protect the pressure vessel.

### **9.16 EFFECTS OF TWO-PHASE VAPOR-LIQUID MIXTURE ON RELIEF VALVE CAPACITY**

Many process systems where conditions for safety relief valve discharge are not single phase of all liquid (through the valve) or all vapor, but a mixture either inside the “containing” vessel or quite often as the fluid passes through the valve orifice; the liquid flashes to partial vapor, or the flashing starts just ahead of the orifice. Here, a mixture attempts to pass through the orifice, and the size must be sufficient or a restriction will exist and pressure will build up in the vessel due to inadequate relief. This problem was of considerable concern to the Design Institute for Emergency Relief of the AIChE during their studies [13]. As a result, considerable research was performed leading to design techniques to handle this problem. A review of two-phase relief shall be presented; meanwhile the designer is referred to the references in the bibliography of this chapter. Also see Leung [39] for detailed procedure and additional references.

### **9.17 SIZING FOR GASES, VAPORS, OR LIQUIDS FOR CONVENTIONAL VALVES WITH CONSTANT BACK PRESSURE ONLY**

This type of valve may be used when the variations in back pressure on the valve discharge connection do not exceed 10% of the valve set pressure, and provided this back pressure variation does not adversely affect the set pressure.

#### **PROCEDURE**

- For a new installation, establish pressure vessel normal maximum operating pressure and temperature, and then the safe increment above this for vessel design conditions and determine the MAWP of the new vessel. (Have a qualified fabricator or designer establish this (see Section 9.15)).
- Establish the maximum *set* pressure for the pressure-relieving valves as the MAWP, or lower, but *never* higher.
- Establish actual *relieving* pressure (and corresponding temperature) from Figure 9-7a (at 110% of set pressure for non-fire and non-explosive conditions). Explosive conditions may require total separate evaluation of the set pressure (*never above the MAWP*), which should be lower or staged; or, most likely, will not be satisfied by a standard SRV due to the extreme rapid

\* Extracted by permission from Teledyne-Farris Engineering Catalog.

response needed. The capacity for flow through the valve is established by these conditions.

4. For *existing vessel* and re-evaluation of pressure relieving requirements, start with the known MAWP for the vessel, recorded on the vessel drawings and on its ASME certification papers. Then follow steps 2 and 3 above.

### ESTABLISH CRITICAL FLOW FOR GASES AND VAPORS

Critical or sonic flow will usually exist for most (compressible) gases or vapors discharging through the nozzle orifice of a pressure-relieving valve. The rate of discharge of a gas from a nozzle will increase for a decrease in the absolute pressure ratio  $P_2/P_1$  (exit/inlet) until the linear velocity in the throat of the nozzle reaches the speed of sound in the gas at that location. Thus, the critical or sonic velocity or critical pressures are those conditions that exist when the gas velocity reaches the speed of sound. At that condition, the actual pressure in the throat will not fall below  $P_1/r_c$  even if a much lower pressure exist downstream [16]. The maximum velocity at the outlet end (or restriction) in a pipe or nozzle is sonic or critical velocity. This is expressed [40] as:

English Engineering units

$$\begin{aligned} v_s &= \sqrt{kgRT} = \sqrt{kg(144)P'\bar{V}} = 68.1\sqrt{kP'\bar{V}} \\ &= 223 \left( \frac{kT}{M} \right)^{0.5} \end{aligned} \quad (9-5)$$

In metric units

$$\begin{aligned} v_s &= \sqrt{\gamma RT} = \sqrt{\gamma P'\bar{V}} = 316.2\sqrt{\gamma p'\bar{V}} \\ &= 91.2 \left( \frac{\gamma T}{M} \right)^{0.5} \end{aligned} \quad (9-5a)$$

where

$k$  = ratio of specific heats at constant pressure/constant volume,  $C_p/C_v$  (see Table 9-5).

$v_s$  = sonic velocity of gas, ft/s (m/s)

$g$  = acceleration of gravity, 32.2 ft/s<sup>2</sup>

$R$  = individual gas constant =  $(MR/M) = 1545/M$ .

In metric units

$$R = R_o/M \text{ J/kg K}$$

where  $R_o = 8314 \text{ J/k mol K}$ ,  $M$  = molecular weight of the gas

$MR$  = universal gas constant = 8314 J/k mol K

$M$  = molecular weight

$T$  = upstream absolute temperature, °R = °F + 460  
(K = °C + 273.15)

$\bar{V}$  = specific volume of fluid, ft<sup>3</sup>/lb(m<sup>3</sup>/kg)

$P_1$  =  $P'$  = upstream pressure, psia (N/m<sup>2</sup> abs (pascal))

$p'$  = pressure, bar

$d$  = pipe inside diameter, in.

$W$  = gas rate, lb/h

$Z$  = gas compressibility factor

$P_c$  =  $P_{crit}$  = critical pressure, psia

$\gamma$  = ratio of specific heat at constant pressure to specific heat at constant volume =  $C_p/C_v$ .

The critical pressure at a pipe outlet is [5c]

$$P_{crit} = \left[ \frac{W}{(408d^2)} \right] \left( \frac{ZT}{M} \right)^{1/2}, \text{ psia} \quad (9-6)$$

The velocity  $v_s$  will occur at the outlet end or in a restricted area [40] when the pressure drop is sufficiently high. The condition of temperature, pressure, and specific volume are those occurring at the point in question.

Critical pressure will normally be found between 53 and 60% of the upstream pressure,  $P'$ , at the time of relief from overpressure, including accumulation pressure in psia. That is,  $P'$  represents the actual pressure at which the relief device is "blowing" or relieving, which is normally above the set pressure by the amount of the accumulation pressure (see Figure 9-7A).

Thus, if the downstream or back pressure on the valve is less than 53–60% (should be calculated) of the values of  $P'$ , note above, critical (sonic) flow will usually exist. If the downstream pressure is over approximately 50% of the relief pressure,  $P'$ , the actual critical pressure should be calculated to determine the proper condition. Calculation of critical pressure [41]:

$$P_c = P_1 \left[ \frac{2}{(k+1)} \right]^{\frac{k}{(k-1)}} \quad (9-7)$$

$$\frac{P_c}{P_1} = r_c = \left[ \frac{2}{(k+1)} \right]^{\frac{k}{(k-1)}} \quad (9-8)$$

for critical flow conditions at  $\beta \leq 0.2$ . This equation is conventionally solved by Figure 9-21.

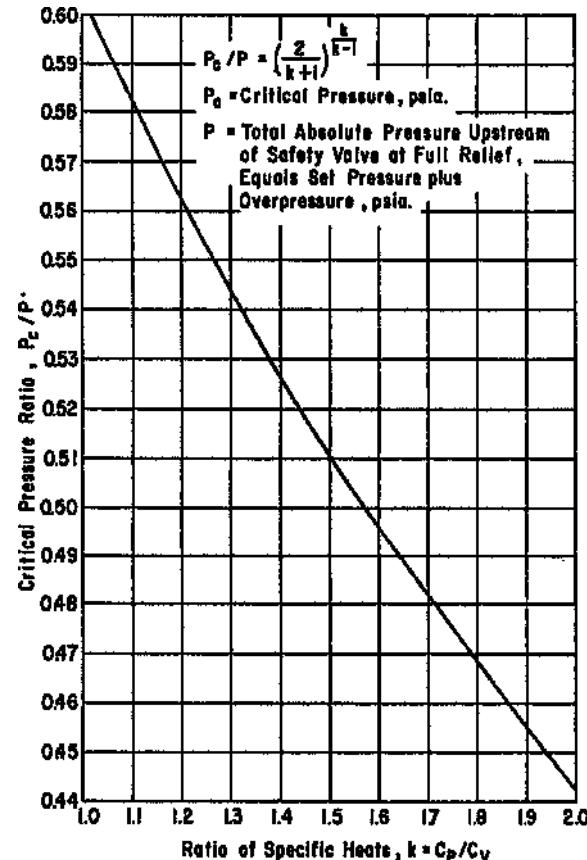


Figure 9-21 Critical back pressure ratio for vapors and gases.

**TABLE 9-5 Properties of Gases and Vapors**

Gases and Vapors	Hydrocarbons Reference Symbols	Chemical Formula	Molecular Weight	R = 1545/Molecular Weight	Critical Conditions		Boiling Point (°F) at 14.7 psia	Specific Volume ft /lb at 14.7 psia & 60°F (Z Factor accounted For)	Latent Heat of Vaporization (Btu/lb at 14.7 psia)	Specific Heat Constant Pressure (C <sub>p</sub> at 60°F)	Specific Heat Constant Volume (C <sub>v</sub> @ 60°F)	Specific Heat Ratio K = C <sub>p</sub> /C <sub>v</sub>
					Pressure (psia)	Temperature (°R)						
1. Acetylene	C <sub>2</sub>	C <sub>2</sub> H <sub>2</sub>	26.04	59.5	905	557	-118.7	14.37	256.0	0.397	0.320	1.24
2. Air	N <sub>2</sub> +O <sub>2</sub>	28.29	53.3	547	239	-317.7	13.09	91.8	0.240	0.171	1.40	
3. Ammonia	NH <sub>3</sub>	17.03	90.8	1657	731	-28.1	22.10	590.0	0.523	0.399	1.31	
4. Argon	Ar	39.94	38.7	705	272	-30.3	9.50	71.7	0.125	0.075	1.66	
5. Benzene	C <sub>6</sub> H <sub>6</sub>	78.11	19.8	714	1013	176.2	*	169.3	0.240	0.215	1.12	
6. Iso-Butane	iC <sub>4</sub>	C <sub>4</sub> H <sub>10</sub>	58.12	26.6	529	735	10.9	6.26	157.8	0.387	0.352	1.10
7. n-Butane	nC <sub>4</sub>	C <sub>4</sub> H <sub>10</sub>	58.12	26.6	551	766	31.1	6.25	165.9	0.397	0.363	1.09
8. Iso-Butylene	iC <sub>4</sub>	C <sub>4</sub> H <sub>10</sub>	56.10	27.5	580	753	19.6	6.54	169.5	0.368	0.333	1.10
9. Butylene	nC <sub>4</sub>	C <sub>4</sub> H <sub>8</sub>	56.10	27.5	583	756	20.7	6.54	167.9	0.327	0.292	1.11
10. Carbon Dioxide	CO <sub>2</sub>	44.01	35.1	1073	548	-109.3	8.53	248.8 <sup>(1)</sup>	0.199	0.153	1.30	
11. Carbon Monoxide	CO	28.01	55.1	514	242	-313.6	13.55	91.0	0.248	0.177	1.40	
12. Carbureted Water Gas (3)			19.48	79.5	454	235	-	19.60	-	0.281	0.208	1.35
13. Chlorine	Cl <sub>2</sub>	70.91	21.8	1119	751	-29.6	5.25	123.8	0.115	0.084	1.36	
14. Coke Oven Gas (3)	-	11.16	138.5	407	197	-	34.10	-	0.679	0.514	1.32	
15. n-Decane	nC <sub>10</sub>	C <sub>10</sub> H <sub>22</sub>	142.28	10.9	312	1115	345.2	*	120.0	0.401	0.387	1.03
16. Ethane	C <sub>2</sub>	C <sub>2</sub> H <sub>6</sub>	30.07	51.5	708	550	-127.5	12.52	210.7	0.410	0.343	1.19
17. Ethyl Alcohol	C <sub>2</sub> H <sub>5</sub> OH	46.07	33.5	927	930	172.9	*	368.0	0.370	0.328	1.13	
18. Ethyl Chloride	C <sub>2</sub> H <sub>5</sub> Cl	64.52	23.9	764	829	54.4	5.59	168.5	0.274	0.230	1.19	
19. Ethylene	C <sub>2</sub>	C <sub>2</sub> H <sub>4</sub>	28.05	55.1	749	510	-154.7	13.40	207.6	0.361	0.291	1.24
20. Flue Gas (2)	-	30.00	51.5	563	265	-	12.63	-	0.240	0.174	1.38	
21. Helium	He	4.00	386.0	33	9	-450.0	94.91	9.9	1.24	0.748	1.66	
22. n-Heptane	nC <sub>7</sub>	C <sub>7</sub> H <sub>16</sub>	100.20	15.4	397	973	209.2	*	136.2	0.399	0.379	1.05
23. n-Hexane	nC <sub>6</sub>	C <sub>6</sub> H <sub>14</sub>	86.17	17.9	434	915	155.7	*	144.8	0.398	0.375	1.06
24. Hydrogen	H <sub>2</sub>	2.02	765.0	188	60	-423.0	187.80	194.0	3.41	2.42	1.41	
25. Hydrogen Sulphide	H <sub>2</sub> S	34.08	45.3	1306	673	-76.5	11.00	236.0	0.254	0.192	1.32	
26. Methane	C	CH <sub>4</sub>	16.04	96.4	673	344	-258.8	23.50	219.7	0.526	0.402	1.31
27. Methyl Alcohol	CH <sub>3</sub> OH	32.04	48.3	1157	924	148.1	*	473.0	0.330	0.275	1.20	
28. Methyl Chloride	CH <sub>3</sub> Cl	50.49	30.6	968	750	-10.8	6.26	184.2	0.200	0.167	1.20	
29. Natural Gas (3)	-	18.82	82.1	675	379	-	20.00	-	0.485	0.382	1.27	
30. Nitrogen	N <sub>2</sub>	28.02	55.1	492	228	-320.0	13.53	85.8	0.248	0.177	1.40	
31. n-Nonane	nC <sub>9</sub>	C <sub>9</sub> H <sub>20</sub>	128.25	12.0	335	1073	303.4	*	125.7	0.400	0.385	1.04
32. Iso-Pentane	iC <sub>5</sub>	C <sub>5</sub> H <sub>12</sub>	72.15	21.4	483	830	82.1	*	145.7	0.388	0.361	1.08
33. n-Pentane	nC <sub>5</sub>	C <sub>5</sub> H <sub>12</sub>	72.15	21.4	485	847	96.9	*	153.8	0.397	0.370	1.07
34. Pentylenne	C <sub>5</sub>	C <sub>5</sub> H <sub>10</sub>	70.13	22.0	586	854	86.0	*	149.0	0.382	0.353	1.08
35. n-Octane	nC <sub>8</sub>	C <sub>8</sub> H <sub>18</sub>	114.22	13.5	362	1025	258.2	*	131.7	0.400	0.382	1.05
36. Oxygen	O <sub>2</sub>	32.00	48.3	730	278	-297.4	11.85	92.0	0.219	0.156	1.40	
37. Propane	C <sub>3</sub>	C <sub>3</sub> H <sub>8</sub>	44.09	35.1	617	666	-43.7	8.45	183.5	0.388	0.342	1.13
38. Propylene	C <sub>3</sub>	C <sub>3</sub> H <sub>6</sub>	42.08	36.7	668	658	-53.9	8.86	188.2	0.354	0.307	1.15
39. Refinery Gas (High Paraffin) (4)	-	28.83	53.6	674	515	-	13.20	-	0.395	0.33	1.20	
40. Refinery Gas (High Olefin) (4)	-	26.26	58.8	639	456	-	14.40	-	0.397	0.33	1.20	
41. Sulphur Dioxide	SO <sub>2</sub>	64.06	24.1	1142	775	14.0	5.80	168	0.147	0.118	1.24	
42. Water Vapor	H <sub>2</sub> O	18.02	85.8	3208	1166	212.0	*	970.3	0.445	0.332	1.33	

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\*These substances are not in a vapor state at 14.7 psia and 60 °F and therefore sp. Vol. values are not listed.

Notes: Most values taken from *Natural Gasoline Supply Men's Association Engineering Data Book*, 1951 – 6th Ed.

1. Heat of Sublimation.

2. Flue gas – Approximate values of based on 80.5% N<sub>2</sub>, 16% CO<sub>2</sub>, 3.5% O<sub>2</sub>. Actual properties depend on exact composition. Reference: *Mark's Engineering Handbook*.

3. Carbureted Water Gas, Coke Oven Gas, and Natural Gas. Based on average compositions. Actual properties will differ depending on exact compositions.

Reference: *Perry's Handbook* (3rd ed.).4. Refinery gas (High Paraffin) – Has a greater mol percent of saturated hydrocarbons (example C<sub>2</sub>H<sub>6</sub>).Refinery gas (High Olefins) – Has a greater mol percent of unsaturated hydrocarbons (example C<sub>2</sub>H<sub>4</sub>).Reference: *Perry's Handbook* (3rd ed.).

At critical conditions, the maximum flow through the nozzle or orifice is [41]

$$W_{\text{max(critical flow)}} = C_o A P_o \sqrt{\frac{k g_c M}{R_g T_o} \left( \frac{2}{k+1} \right)^{(k+1)/(k-1)}} \quad (9-9)$$

where

M = molecular weight of vapor or gas lbm/lb mol

T<sub>o</sub> = temperature of service, °R = (°F + 460)R<sub>g</sub> = ideal universal gas constant = 1545, ft lbf/lb mol °R,  
also = MR

## 610 PROCESS SAFETY AND PRESSURE-RELIEVING DEVICES

$C_o$	= discharge coefficient for sharp-edged orifice = 0.61 for Reynolds Number > 30,000 and not sonic = 1.0 for sonic flow, $C_o$ increases from 0.61 to 1.0 (use 1.0 to be conservative [16, 41], 5th ed.)
$A$	= area of opening, orifice, or hole, or nozzle, $\text{ft}^2$
$P_1 = P_o$	= upstream pressure, $\text{lb}/\text{ft}^2$ abs (psfa)
$g_c$	= conversion factor $32.174 \frac{16\text{m}}{6\text{f}} \frac{\text{ft}}{\text{s}^2}$
$\beta$	= ratio of orifice diameter to pipe diameter (or nozzle inlet diameter)
$W_{\max}$	= maximum mass flow at critical or choked conditions, $\text{lb}/\text{s}$
$P_c = P_{\text{crit}}$	= critical flow throat pressure, psia = sonic = choked pressure = maximum downstream pressure producing maximum flow.

If the downstream pressure exceeds the critical flow pressure, then sub-critical pressure will occur and the equations for subcritical flow should be used.

When the downstream pressure is less than (or below) the critical or choked pressure, the velocity and fluid flow rate at a restriction or throat will not/cannot be increased by lowering the downstream pressure further, and the fluid velocity at the restriction or throat is the velocity of sound at the conditions [41].

The critical or sonic ratio is conveniently shown in Figure 9-21, but this does not eliminate the need for calculating the  $P_c/P_1$  ratio for a more accurate result.

### EXAMPLE 9-2 Flow Through Sharp Edged Vent Orifice (adapted after [41])

A small hole has been deliberately placed in a vessel near the top to provide a controlled vent for a nitrogen purge/blanket. The hole is 0.2 in. diameter with the vessel operating at 150 psig at 100° F. Determine the flow through this vent hole. Assume it acts as a sharp edged orifice.

$$k \text{ (for nitrogen)} = 1.4$$

From Eq. (9-8),

$$P_c = (150 + 14.7) \left[ \frac{2}{(1.4 + 1)} \right]^{\frac{1.4}{(1.4 - 1)}} = 87.0 \text{ psia, critical pressure}$$

$$\text{Hole area } A = \pi d^2/4 = \pi(0.2)^2/4 = 0.0314 \text{ in.}^2 = 0.0002182 \text{ ft}^2$$

Discharge coefficient  $C_o$  = assumed = 1.0 (Note: Could calculate  $Re$  to verify)

$$\text{Inside pressure} = 150 \text{ psig} + 14.7 = 164.7 \text{ psia}$$

$$T_o = 100 + 460 = 560^\circ \text{R}$$

$$W_{\max} = [1.0(0.0002182)(164.7)(144 \text{ in.}^2/\text{ft}^2)] \left[ \sqrt{\left[ \frac{(1.4)(32.174)(28)}{(1545)(560)} \left[ \frac{2}{1.4 + 1} \right]^{(1.4+1)/(1.4-1)} \right]} \right] \\ = 0.1143 \text{ lb/s}$$

$$\text{Critical flow rate, } W_{\max} = 0.1143 \text{ lb/s}$$

## 9.18 ORIFICE AREA CALCULATIONS [42]

Calculations of orifice flow area for conventional pressure-relieving valves, and flow is critical (sonic) through part of relieving system,

that is back pressure is less than 55% of the absolute relieving pressure (including set pressure plus accumulation). See Figure 9-7a, use  $K_b = 1.0$  (Figure 9-26), constant back pressure with variation not to exceed 10% of the set pressure.

- a) For vapors and gases, in  $\text{lb}/\text{h}$ ,  $K_b = 1.0$ ; "C" from Figure 9-25,  $P$  is the relieving pressure absolute, psia

$$A = \frac{W\sqrt{TZ}}{CK_d P_1 K_b \sqrt{M}}, \text{ in.}^2 \quad (9-10)$$

(effective net discharge area)

where

$C$  = gas or vapor flow constant

$K_b = 1$ , when back pressure is below 55% of absolute relieving pressure

$K_d$  = coefficient of discharge (0.953)

$M$  = molecular weight of gas or vapor  $\text{lbm}/\text{lb mole}$

$P_1$  = relieving pressure, psia = set pressure + overpressure + 14.7

$W$  = required vapor or gas capacity,  $\text{lb}/\text{h}$

$T$  = inlet temperature,  ${}^\circ\text{R} = {}^\circ\text{F} + 460$

$Z$  = compressibility factor corresponding to  $T$  and  $P$ .

Metric units in  $\text{kg}/\text{h}$ :

$$A = \frac{1.317W\sqrt{TZ}}{CK_d P_1 K_b \sqrt{M}} \text{ cm}^2 \quad (9-10a)$$

where

$A$  = required orifice area in  $\text{cm}^2$

$C$  = gas or vapor flow constant

$W$  = required vapor capacity,  $\text{kg}/\text{h}$

$T$  = inlet temperature,  $K = ({}^\circ\text{C} + 273.15)$

$K_b$  = vapor or gas flow correction factor for constant back pressure above critical pressure

= 1, when back pressure is below 55% of absolute relieving pressure.

$K_d$  = coefficient of discharge (0.953 for vapors, gases)

$P_1$  = relieving pressure in bara = set pressure + overpressure + 1.013

$Z$  = compressibility factor corresponding to  $T$  and  $P$  ( $Z = 1.0$ ).

- b) For vapors and gases, in  $\text{scfm}$ ,  $K_b = 1.0$

$$A = \frac{V\sqrt{GTZ}}{1.175CP_1K_dK_b}, \text{ in.}^2 \quad (9-11)$$

where

$G$  = specific gravity of gas (molecular weight of gas/molecular weight of air = 29.0)

$T$  = inlet temperature,  ${}^\circ\text{R} = {}^\circ\text{F} + 460$

$P_1$  = relieving pressure, psia = set pressure + overpressure + 14.7

$V$  = required gas capacity,  $\text{scfm}$

$Z$  = compressibility factor corresponding to  $T$  and  $P$  ( $Z = 1.0$ ).

Metric units in Normal  $\text{m}^3/\text{h}$ :

$$A = \frac{V\sqrt{GTZ}}{3.344CK_dP_1K_b} \text{ cm}^2 \quad (9-11a)$$

where

$A$  = required orifice area in  $\text{cm}^2$

$C$  = gas or vapor flow constant

$G$  = specific gravity of gas (molecular weight of gas/molecular weight of air = 29.0)

$V$  = required gas capacity,  $\text{m}^3/\text{h}$

$T$  = inlet temperature,  $K = (\text{°C} + 273.15)$

$K_d$  = coefficient of discharge (0.953 for vapors, gases)

$K_b$  = vapor or gas flow correction factor for constant back pressure above critical pressure ( $K_b = 1.0$ )

$P_1$  = relieving pressure in bara = set pressure + overpressure + 1.013

$Z$  = compressibility factor corresponding to  $T$  and  $P$  ( $Z = 1.0$ ).

Pressure relief devices in gas or vapor service that operate at critical conditions may be sized using the following equations [5d]:

US customary units:

$$A = \frac{W}{CK_d P_1 K_b K_c} \sqrt{\frac{TZ}{M}} \quad (9-11b)$$

or

$$A = \frac{V\sqrt{TZM}}{6.32CK_d P_1 K_b K_c} \quad (9-11c)$$

or

$$A = \frac{V\sqrt{TZG}}{1.175CK_d P_1 K_b K_c} \quad (9-11d)$$

SI Units:

$$A = \frac{13,160 W}{CK_d P_1 K_b K_c} \sqrt{\frac{TZ}{M}} \quad (9-11e)$$

or

$$A = \frac{35,250 V\sqrt{TZM}}{CK_d P_1 K_b K_c} \quad (9-11f)$$

or

$$A = \frac{189,750 V\sqrt{TZG}}{CK_d P_1 K_b K_c} \quad (9-11g)$$

where

$A$  = required effective discharge area of the device,  $\text{in.}^2$  ( $\text{mm}^2$ )

$W$  = required flow through the device,  $\text{lb/h}$  ( $\text{kg/h}$ )

$C$  = coefficient determined from an expression of the ratio of the specific heats ( $k = C_p/C_v$ ) of the gas or vapor at inlet relieving conditions (see Figure 9-25) or

$$C = 520 \sqrt{k \left( \frac{2}{k+1} \right)^{\frac{(k+1)}{(k-1)}}} \quad (9-11h)$$

$K_d$  = effective coefficient of discharge. For preliminary sizing, use the following values:

(i) 0.975, when a pressure-relief valve is installed with or without a rupture disk in combination;

(ii) 0.62, when a pressure-relief valve is not installed and sizing is for a rupture disk

$P_1$  = upstream relieving pressure, psia (kPa). This is the set pressure plus the allowable overpressure plus atmospheric pressure

$K_b$  = capacity correction factor due to back pressure.

The back pressure correction factor applies to balanced bellows valves only. For conventional and pilot-operated valves,  $K_b = 1.0$

$K_c$  = combination correction factor for installation with a rupture disk upstream of the pressure-relief valve

$T$  = relieving temperature of the inlet gas or vapor,

$^{\circ}\text{R} = ^{\circ}\text{F} + 460$  ( $K = ^{\circ}\text{C} + 273$ )

$Z$  = compressibility factor for the deviation of the actual gas from a perfect gas, a ratio evaluated at inlet relieving conditions.

$M$  = molecular weight of the gas or vapor at inlet relieving conditions

$V$  = required flow through the device, scfm at 14.7 psia and 60° F ( $\text{Nm}^3/\text{min}$  at 0° C and 101.3 kPa)

$G$  = specific gravity of gas at standard conditions referred to air at standard conditions (normal conditions).

$G = 1.00$  for air at 14.7 psia and 60° F (101.3 kPa and 0° C).

- c) For steam, in lb/h;  $K_b = 1.0$  and  $K_{sh} = 1.0$  for saturated steam when back pressure is below 55% of absolute relieving pressure [33d].

$$A = \frac{W_s}{51.5P_1 K_d K_b K_c K_n K_{sh}}, \text{ in.}^2 \quad (9-12)$$

SI units:

$$A = \frac{190.4 \times W_s}{P_1 K_d K_b K_c K_n K_{sh}} \quad (9-12a)$$

where

$W_s$  = required steam capacity in  $\text{lb/h}$  ( $\text{kg/h}$ )

$K_d$  = effective coefficient of discharge. For preliminary sizing, use the following values:

(i) 0.975, when a pressure-relief valve is installed with or without a rupture disk in combination;

(ii) 0.62 when a pressure-relief valve is not installed and sizing is for a rupture disk

$K_b = 1$ , when back pressure is below 55% of absolute relieving pressure. The back pressure correction factor applies to balanced bellows valves only. For conventional valves, use a value for  $K_b = 1.0$

$K_c$  = combination correction factor for installation with a rupture disk upstream of the pressure-relief valve = 1.0, when a rupture disk is not installed = 0.9, when a rupture disk is installed in combination with a pressure-relief valve and the combination does not have a published value

$K_n$  = Napier steam correction factor for set pressures between 1500 and 2900 psig = 1, when  $P_1 \leq 1500$  psia (10,339 kPa)

$$= \frac{0.1906 \times P_1 - 1000}{0.2292 \times P_1 - 1061} \quad (\text{US customary units})$$

$$= \frac{0.02764 \times P_1 - 1000}{0.03324 \times P_1 - 1061} \quad (\text{SI units})$$

where  $P_1 \geq 1500$  psia (10,339 kPa) and  $\leq 3200$  psia (22,057 kPa)

$K_{sh} = 1$  for saturated steam

$P_1$  = relieving pressure, psia = set pressure + overpressure + 14.7

= relieving pressure, kPa = set pressure + overpressure + 101.3.

Metric units in kg/h:

$$A = \frac{W_s}{52.49 P_1 K_d K_b K_{sh}}, \text{ cm}^2 \quad (9-12b)$$

where

$W_s$  = required steam capacity, kg/h.

$K_b$  = 1, when back pressure is below 55% of absolute relieving pressure.

$K_d$  = coefficient of discharge ( $K_d = 0.953$ )

$K_{sh}$  = 1 for saturated steam

$P_1$  = relieving pressure, bara = set pressure + overpressure + 1.013.

d) For air, in scfm;  $K_b = 1.0$ , when the back pressure is below 55% of absolute relieving pressure

$$A = \frac{V_a \sqrt{T}}{418 K_d P_1 K_b} \quad (9-13)$$

where

$V_a$  = relieving air capacity, scfm

$K_b$  = 1, when back pressure is below 55% of absolute relieving pressure

$K_d$  = coefficient of discharge ( $K_d = 0.953$ )

$P_1$  = relieving pressure, psia = set pressure + overpressure + 14.7

$T$  = inlet temperature, °R = °F + 460.

Metric units in m<sup>3</sup>/h:

$$A = \frac{V_a \sqrt{T}}{1189.3 K_d P K_b} \text{ cm}^2 \quad (9-13a)$$

where

$V_a$  = Required air capacity in m<sup>3</sup>/h

$K_b$  = 1, when back pressure is below 55% of absolute relieving pressure

$K_d$  = coefficient of discharge ( $K_d = 0.953$ )

$P$  = relieving pressure, bara = set pressure + overpressure + 1.013

$T$  = inlet temperature,  $K = (\text{°C} + 273.15)$ .

e) For liquids, GPM

$K_p$  = 1.0 at 10% overpressure

$K_u$  = 1.0 at normal viscosities

$\Delta P = P_1 - P_2$  = upstream pressure, psig (set + overpressure) – total back pressure, psig.

### 9.19 SIZING VALVES FOR LIQUID RELIEF: PRESSURE-RELIEF VALVES REQUIRING CAPACITY CERTIFICATION [5D]

Section VIII, Division I, of the ASME code requires that capacity certification be obtained for pressure-relief valves designed for liquid service. The procedure for obtaining capacity certification includes testing to determine the rated coefficient of discharge for the liquid-relief valves at 10% overpressure.

ASME code valves: Board Certified for liquids only [5d].

$$A = \frac{V_L \sqrt{G}}{38 K_d K_w K_c K_u \sqrt{P_1 - P_2}} \text{ in.}^2 \quad (9-14)$$

SI units:

$$A = \frac{11.78 \times V_L \sqrt{G}}{K_d K_w K_c K_u \sqrt{P_1 - P_2}} \text{ mm}^2 \quad (9-14a)$$

where

$A$  = required effective discharge area, in.<sup>2</sup> (mm<sup>2</sup>)

$G$  = specific gravity of the liquid at the flowing temperature referred to water at standard conditions (density of liquid/density of water ≈ 62.3 lb/ft<sup>3</sup>)

$V_L$  = required liquid capacity, US gpm (l/min)

$K_d$  = rated coefficient of discharge that should be obtained from the valve manufacturer.

For a preliminary sizing,

= 0.65, when a pressure-relief valve is installed with or without a rupture disk in combination

= 0.62, when a pressure-relief valve is not installed and sizing is for a rupture disk

$K_w$  = variable or constant back pressure sizing factor, balanced valves, liquid only (Figure 9-28). If the back pressure is atmospheric,  $K_w = 1.0$ . Balanced bellows valves in back pressure service will require the correction factor. Conventional and pilot-operated valves require no special correction

$K_c$  = combination correction factor for installation with a rupture disk upstream of the pressure-relief valve

= 1.0, when a ruptured disk is not installed

= 0.9, when a rupture disk is installed in combination with a pressure-relief valve and the combination does not have a published value

$K_u$  = viscosity correction factor

( $K_u = 1$  at normal viscosities)

$$= \left( 0.9935 + \frac{2.878}{Re^{0.5}} + \frac{342.75}{Re^{1.5}} \right)^{-1.0}$$

$P_1$  = upstream relieving pressure, psig (kPag). This is the set pressure plus allowable overpressure

$P_2$  = back pressure at outlet, psig (kPag).

### 9.20 SIZING VALVES FOR LIQUID RELIEF: PRESSURE-RELIEF VALVES NOT REQUIRING CAPACITY CERTIFICATION [5D]

Before the ASME code incorporated requirements for capacity certification, valves were generally sized for liquid service, which assumes an effective coefficient of discharge,  $K_d = 0.62$ , and 25% overpressure.

This method will typically result in an oversized design where a liquid valve is used for an application with 10% overpressure. A  $K_p$  correction factor of 0.6 is used for this situation [5d].

$$A = \frac{V_L \sqrt{G}}{38 K_d K_w K_c K_p K_u \sqrt{1.25 P_1 - P_2}} \text{ in.}^2 \quad (9-15)$$

SI Units

$$A = \frac{11.78 \times V_L \sqrt{G}}{K_d K_w K_c K_u K_p \sqrt{1.25 P_1 - P_2}} \text{ mm}^2 \quad (9-15a)$$

where

$A$  = required effective discharge area, in.<sup>2</sup> (mm<sup>2</sup>)

$G$  = specific gravity of liquid at flowing temperature referred to water at standard conditions

$V_L$  = required liquid capacity, US gpm (l/min).

$K_d = 0.62$  for a preliminary sizing estimation; otherwise, rated coefficient of discharge should be obtained from the valve manufacturer

$K_w$  = variable or constant back pressure sizing factor, balanced valves, liquid only (Figure 9-28). If the back pressure is atmospheric,  $K_w = 1.0$ . Balanced bellows valves in back pressure service will require the correction factor. Conventional and pilot-operated valves require no special correction.

$K_c$  = combination correction factor for installation with a rupture disk upstream of the pressure relief valve  
 = 1.0, when a ruptured disk is not installed  
 = 0.9, when a rupture disk is installed in combination with a pressure-relief valve and the combination does not have a published value

$K_p$  = liquid capacity correction factor for overpressures. At 25% overpressure,  $K_p = 1.0$ . For overpressure other than 25%,  $K_p$  is determined from Figure 9-22

$K_u$  = viscosity correction factor

( $K_u = 1$  at normal viscosities)

$$= \left( 0.9935 + \frac{2.878}{Re^{0.5}} + \frac{342.75}{Re^{1.5}} \right)^{-1.0}$$

$P_1$  = set pressure, psig (kpag)

$P_2$  = total back pressure, psig (kPag).

Metric units in dm<sup>3</sup>/min:

$$A = \frac{V_L \sqrt{G}}{84.89 K_d K_p K_u \sqrt{1.25 P_1 - P_2}} \text{ cm}^2 \quad (9-15b)$$

where,

$A$  = required effective discharge area, cm<sup>2</sup>

$G$  = specific gravity of liquid at flowing temperature referred to water at standard conditions

$V_L$  = required liquid capacity, US gpm (L/min)

$K_d = 0.62$  for a preliminary sizing estimation; otherwise, rated coefficient of discharge should be obtained from the valve manufacturer

$K_p$  = liquid capacity correction factor for overpressures. At 25% overpressure,  $K_p = 1.0$ . For overpressure other than 25%,  $K_p$  is determined from Figure 9-22

$K_u$  = viscosity correction factor ( $K_u = 1$  at normal viscosities)

$P_1$  = set pressure at inlet, barg

$P_2$  = back pressure at outlet, barg.

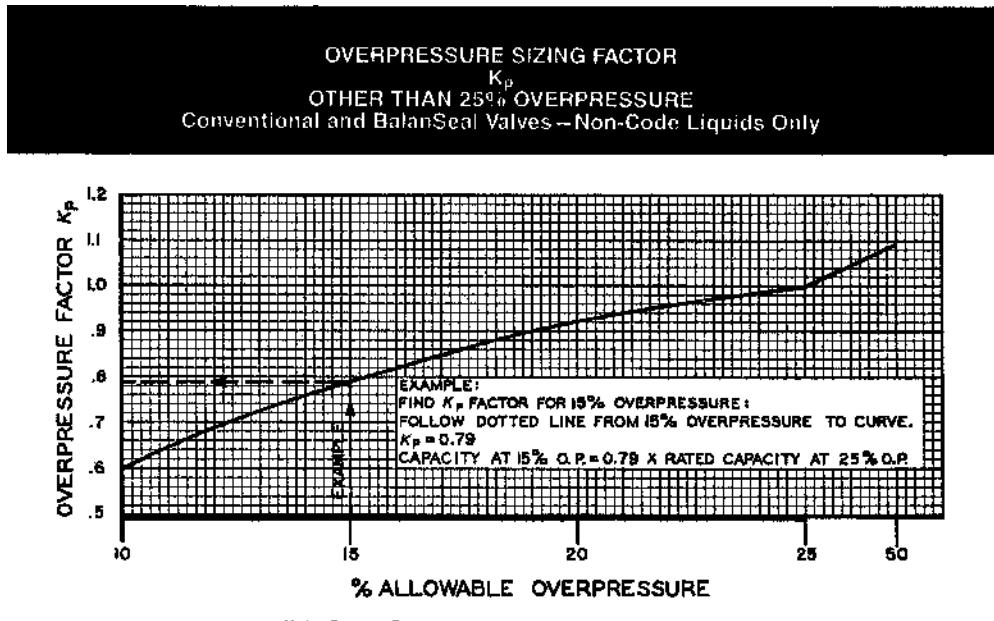
When sizing a relief valve for viscous liquid service, the orifice area is first calculated for non-viscous service to obtain a preliminary discharge area. The next standard orifice size is used to calculate the Reynolds number.

To apply the viscosity correction  $K_u$ , a preliminary or trial calculation should be made for the areas required using the above equation of paragraph (e) or the modified equation (still ASME conformance [5] but not capacity certified). A simplified equation based on the ASME Pressure Vessel Code equations, Section VIII, Division 1, Mandatory Appendix XI uses  $K$  coefficient of discharge in the equations, where  $K$  is defined as 90% of the average  $K_d$  of certified tests with compressible or incompressible fluids, see [42], pg. 40.

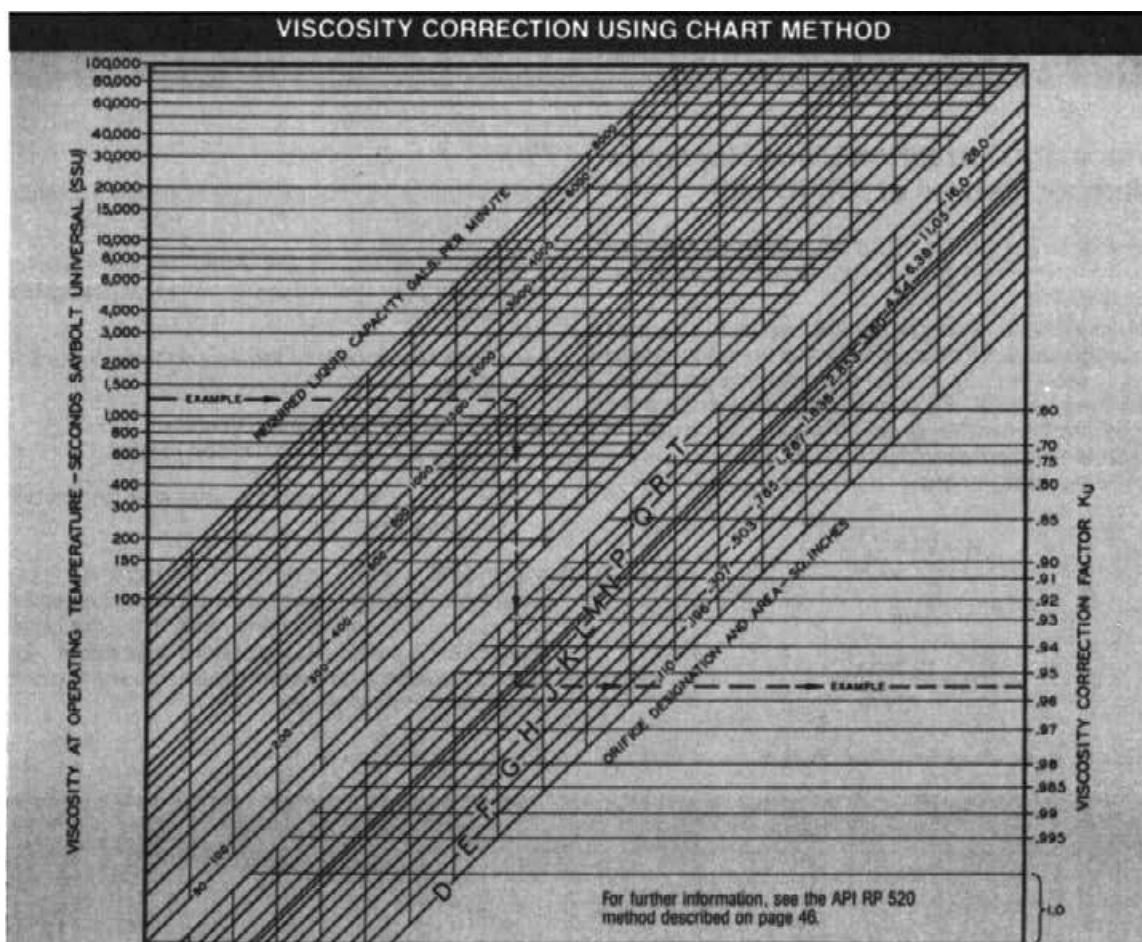
For first trial, assume  $K_u$  for viscosity = 1.0

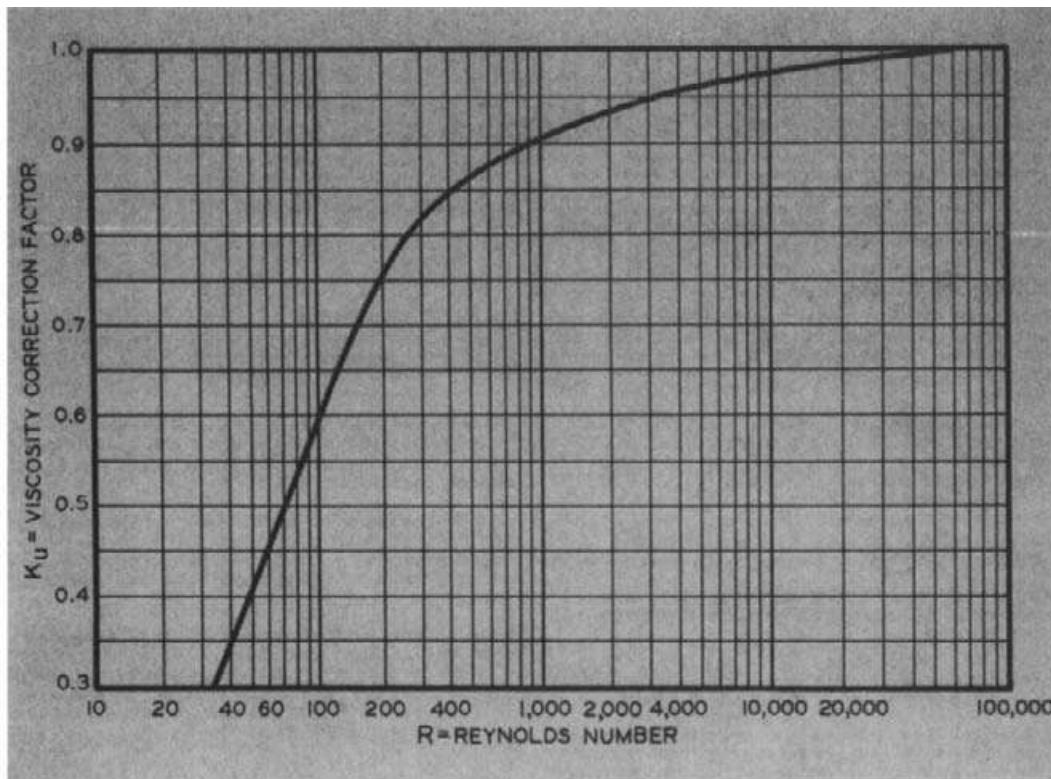
For final calculation use  $K_u$  from Figures 9-23 and 9-24 or a correlation equation defined by:

$$K_u = -2.38878 + 1.2502 (\ln Re) - 0.17510 (\ln Re)^2 + 0.01087 (\ln Re)^3 - 0.00025 (\ln Re)^4 \quad (9-15c)$$



**Figure 9-22** Liquids overpressure sizing factor,  $K_p$ , for other than 25% overpressure. Applies to non-code liquids only using conventional and balanced valves. (By permission from Teledyne Farris Engineering Co.)





### Viscosity Correction Using Reynold's Number Method of API RP520

As an alternate to Figure 7-23, you may wish to use the method given in API RP 520 for sizing viscous liquids.

When a relief valve is sized for viscous liquid service, it is suggested that it be sized first as for nonviscous type application in order to obtain a preliminary required discharge area, A. From manufacturer's standard orifice sizes, the next larger orifice size should be used in determining the Reynolds number R, from either of the following relationships:

$$R = \frac{V_L(2,800 G)}{\mu \sqrt{A}}$$

or

$$*R = \frac{12,700 V_L}{UVA}$$

\*Use of this equation is not recommended for viscosities less than 100 SSU.

Where:

V<sub>L</sub> = flow rate at the flowing temperature in U.S. gallons per minute.

G = specific gravity of the liquid at the flowing temperature referred to water = 1.00 at 70 degrees Fahrenheit.

$\mu$  = absolute viscosity at the flowing temperature, in centipoises.

A = effective discharge area, in square inches (from manufacturers' standard orifice areas).

U = viscosity at the flowing temperature, in Saybolt Universal seconds.

After the value of R is determined, the factor K<sub>v1</sub> is obtained from the graph. Factor K<sub>v</sub> is applied to correct the "preliminary required discharge area." If the corrected area exceeds the "chosen standard orifice area", the above calculations should be repeated using the next larger standard orifice size.

<sup>1</sup>K<sub>v</sub> of API = K<sub>v</sub> of Teledyne Farris

**Figure 9-24** Viscous liquid valve sizing using the method of API RP-520. (Reprinted by permission from Teledyne Farris Engineering Co. and *Sizing, Selection and Installation of Pressure Relieving Devices in Refineries*, Part I "Sizing and Selection", API RP-520, 5th ed., Jul 1990, American Petroleum Institute.)

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$K_u$  is then substituted in Eqs 9-15 and 9-15a (SI units). Determine the needed Reynolds number,  $Re$ , using the next size larger orifice. Area is determined from that made in the first trial calculation [5].

$$Re = V_L \frac{(2800 G)}{\mu \sqrt{A}} \quad (9-16)$$

or

$$Re = \frac{12,700 V_L}{(U \sqrt{A})}, \text{ (do not use when } U < 100 \text{ SSU}) \quad (9-17)$$

SI units:

$$Re = V_L \frac{(18,800 G)}{\mu \sqrt{A}} \quad (9-16a)$$

or

$$Re = \frac{(85,220 V_L)}{U \sqrt{A}} \quad (9-17a)$$

where

$Re$  = Reynolds number

$\mu$  = absolute viscosity at the flowing temperature, cP

$P_1$  = set pressure, psig

$U$  = viscosity at the flowing temperature, Saybolt universal seconds (SSU) (See Appendix C-12 and C-13)

$P_2$  = total back pressure, psig.

## 9.21 REACTION FORCES

The discharge of a pressure-relief valve with unsupported discharge piping will impose a reactive load on the inlet of the valve as a result of the reaction force of the flowing fluid. This is particularly essential where piping discharging to atmosphere includes a 90° turn and has no support for the outlet piping. All reactive loading due to the operation of the valve is then transmitted to the valve and inlet piping.

The following formula is based on a condition of critical steady state flow of a compressible fluid that discharges to the atmosphere through an elbow and a vertical discharge pipe. The reaction force ( $F$ ) includes the effects of both momentum and static pressure [5d]. The formula is applicable for any gas, vapor, or steam and is expressed by

$$F = \frac{W}{366} \sqrt{\frac{kT}{(k+1)M}} + (AP) \quad (9-17b)$$

In metric units

$$F = 129W \sqrt{\frac{kT}{(k+1)M}} + 0.1(AP) \quad (9-17c)$$

where

$A$  = area of the outlet at the point of discharge, in.<sup>2</sup> (mm<sup>2</sup>)

$C_p$  = specific heat at constant pressure

$C_v$  = specific heat at constant volume

$F$  = reaction force at the point of discharge to the atmosphere, lb<sub>f</sub> (N)

$k$  = ratio of specific heats ( $C_p/C_v$ ) at the outlet conditions

$M$  = molecular weight of the process fluid

$P$  = static pressure within the outlet at the point of discharge, psig (barg)

$T$  = temperature at the outlet, °R (K)

$W$  = flow of any gas or vapor, lb<sub>m</sub>/h (kg/s).

A computer program SAFETY has been developed for sizing relief valves for gas, vapor, liquid, steam, air, and fire relief conditions. Table 9-6 shows typical computer results of Example 9-3.

## 9.22 CALCULATIONS OF ORIFICE FLOW AREA USING PRESSURE RELIEVING BALANCED BELLOWS VALVES, WITH VARIABLE OR CONSTANT BACK PRESSURE

Must be used when back pressure variation exceeds 10% of the set pressure of the valve. Flow may be critical or non-critical for balanced valves. All orifice areas,  $A$ , in square inches [42]. The sizing procedure is the same as for conventional valves listed above (Eq. 9-10), but uses equations given below incorporating the correction factors  $K_v$  and  $K_w$ . With variable back pressure, use maximum value for  $P_2$  [5a], [42].

a) For vapors or gases, lb/h

$$A = \frac{W \sqrt{TZ}}{CK_d P_1 K_v \sqrt{M}}, \text{ in.}^2 \quad (9-18)$$

Metric units in kg/h:

$$A = \frac{1.317 W \sqrt{TZ}}{CK_d P_1 K_v \sqrt{M}}, \text{ cm}^2 \quad (9-18a)$$

b) For vapors or gases, scfm

$$A = \frac{V \sqrt{GTZ}}{1.175 CK_d P_1 K_v}, \text{ in.}^2 \quad (9-19)$$

In metric units in Normal m<sup>3</sup>/h:

For vapor or gases, Normal m<sup>3</sup>/h

$$A = \frac{V \sqrt{GTZ}}{3.344 CK_d P_1 K_v}, \text{ cm}^2 \quad (9-19a)$$

c) For steam, lb/h

$$A = \frac{W_s}{51.5 K_d K_v K_{sh} K_n P_1}, \text{ in.}^2 \quad (9-20)$$

Metric units in kg/h:

$$A = \frac{W_s}{52.49 K_d K_v K_{sh} K_n P_1}, \text{ cm}^2 \quad (9-20a)$$

d) For air, scfm

$$A = \frac{V_a \sqrt{T}}{418 K_d P_1 K_v}, \text{ in.}^2 \quad (9-21)$$

Metric units, Normal m<sup>3</sup>/h:

$$A = \frac{V_a \sqrt{T}}{1189.3 K_d P_1 K_v}, \text{ cm}^2 \quad (9-21a)$$

e) For liquids, GPM; ASME code valve

$$A = \frac{V_L \sqrt{G}}{38.0 K_d K_w K_u \sqrt{\Delta P}}, \text{ in.}^2 \quad (9-22)$$

f) For liquids, GPM, non-ASME code valve

$$A = \frac{V_L \sqrt{G}}{38.0 K_d K_p K_w K_u \sqrt{(1.25 P_1 - P_2)}}, \text{ in.}^2 \quad (9-23)$$

**EXAMPLE 9-3**

In a process plant, ammonia is used to control the pH during a production process to obtain maximum yield of a product. For this purpose, liquid ammonia is vaporized by passing steam through a coil in a vaporizer. A relief valve from the vaporizer is set at a pressure of 290 psig. If the relieving rate of ammonia vapor is 620 lb/h, determine the size of the relief valve required to relieve ammonia vapor during an emergency.

Design data:

Ratio of specific heat capacities ( $C_p/C_v$ ) = 1.33

Molecular weight of ammonia = 17.03

Critical pressure,  $P_c$  = 111.3 atm

Critical temperature,  $T_c$  = 405.6 K.

Constants in Antoine Equation:

$$A = 16.9481$$

$$B = 2132.5$$

$$C = -32.98$$

*Solution*

From Antoine's equation,

$$\ln P = A - \frac{B}{T+C}$$

Relieving pressure = mmHg

Relieving temperature = K

Relieving pressure = set pressure + overpressure

$$\begin{aligned} &+ \text{atmospheric pressure} \\ &= 290 + (290 \times 0.1) + 14.7 \\ &= 333.7 \text{ psia} \end{aligned}$$

*Conversion*

$$1 \text{ psi} = 51.715 \text{ mmHg}$$

$$1 \text{ atm} = 14.7 \text{ psi}$$

$$\text{Relieving pressure} = 17,257.3 \text{ mmHg}$$

Relieving temperature from Antoine's equation,

$$\ln P = A - \frac{B}{T+C}$$

$$\ln(17,257.296) = 16.9481 - \frac{2132.5}{(T - 32.98)}$$

$$-7.192(T - 32.98) = -2132.5$$

$$-7.192T + 237.19 = -2132.5$$

$$T = \frac{2369.69}{7.192}$$

$$= 329.5 \text{ K}(593.45^\circ \text{ R})$$

Critical pressure = 1636.1 psia

Critical temperature = 703.41° R

Since compressibility factor  $Z$  is a function of temperature  $T_r$  and  $P_r$ , then

$$T_r = T/T_c = 593.45/730.41$$

$$= 0.812$$

$$P_r = P/P_c = 333.7/1636.1$$

$$= 0.204$$

From the Nelson and Obert chart,  $Z = 0.837$ .

Coefficient "C" for gas related to specific heats.

Substituting the value for  $k$  in Eq. (9-11h),

$$\begin{aligned} C &= 520 \left[ k \left( \frac{2}{(k+1)} \right)^{\frac{(k+1)}{(k-1)}} \right]^{0.5} \\ C &= 520 \left[ (1.33) \left( \frac{2}{1.33+1} \right)^{\frac{(1.33+1)}{(1.33-1)}} \right]^{0.5} \\ &= 350 \end{aligned}$$

For vapor, the orifice area (from Eq. (9-10)) is

$$A = \frac{W\sqrt{TZ}}{CK_dPK_b\sqrt{M}} \text{ in.}^2$$

$$\begin{aligned} A &= \frac{620\sqrt{(593.45)(0.812)}}{(350)(0.953)(333.7)(1)\sqrt{17.03}} \\ &= 0.0296 \text{ in.}^2 \end{aligned}$$

The nearest standard orifice area is 0.110 in.<sup>2</sup>, having an inlet and outlet valve body sizes of 1 and 2 in. respectively. The designation is 1D2 pressure-relief valve.

The maximum vapor rate with the nearest standard orifice area is:

$$\begin{aligned} W &= ACK_dPK_b\sqrt{M/(TZ)} \\ &= (0.11)(350)(0.953)(333.7)(1)\sqrt{17.03/(593.45 \times 0.81)} \\ &= 2305 \text{ lb/h.} \end{aligned}$$

Metric unit in dm<sup>3</sup>/min:

$$A = \frac{V_L\sqrt{G}}{84.89K_dK_pK_wK_u\sqrt{(1.25P_1 - P_2)}}, \text{ cm}^2 \quad (9-23a)$$

When the back pressure  $P_2$  is variable, use the maximum value,

where (Courtesy of Teledyne Farris Engineering Co. [42])

$A$  = required orifice area in in.<sup>2</sup> This is as defined in the ASME code and ANSI/API Std 526.

$W$  = required vapor capacity, lb/h

$W_s$  = required steam capacity, lb/h

**EXAMPLE 9-4**

A bellows type pressure-relief valve is required to protect a vessel containing an organic liquid. The required relieving capacity is 310 US gpm. The inlet temperature is 170° F and the set pressure is 100 psig. Allowable overpressure is 25% with a built-up back pressure of 25 psig. The fluid's physical properties such as the specific gravity and viscosity are 1.45 and 3200 cP respectively. Determine the orifice size of the valve.

Take the correction factors

$$K_d = 0.62, K_w = 0.92, K_c = K_p = K_u = 1$$

*Solution*

Non-ASME Code Liquid Valves [5c] are non-board certified for liquids, but code acceptable for other services.  $K_p$  from Figure 9-22,  $K_d = 0.62$ , and 25% overpressure.

Substituting values in Eq. (9-15),

$$A = \frac{V_L \sqrt{G}}{38 K_d K_w K_c K_p K_u \sqrt{1.25 P_1 - P_2}} \text{ in.}^2$$

$$A = \frac{(310) \sqrt{1.45}}{38 (0.62) (0.92) (1.0) (1.0) (1.0) \sqrt{1.25 (100) - 25}}$$

$$= 1.722 \text{ in.}^2$$

The next standard orifice area is 1.838 in.<sup>2</sup>, having an inlet and outlet valve body sizes of 3 and 4 in. respectively. The designation is 3K4 pressure-relief valve.

The maximum flow rate with the standard orifice area is

$$V_L = A (38) (K_d) (K_w) (K_c) (K_p) (K_u) \sqrt{(1.25 P_1 - P_2) / G}$$

$$= 1.838 (38) (0.62) (0.92) (1.0) (1.0) (1.0)$$

$$\sqrt{(1.25 (100) - 25) / 1.45}$$

$$= 330.85 \text{ gpm}$$

Next, calculate the Reynolds number using the manufacturer's orifice area from Eq. (9-16),

$$Re = V_L \frac{(2800G)}{\mu \sqrt{A}}$$

$$Re = \frac{(310) (2800 \times 1.45)}{(3200) \sqrt{1.838}}$$

$$= 290.0$$

Determine the viscosity correction factor  $K_u$  from Eq. (9-15c),

$$K_u = -2.38878 + 1.2502 (\ln Re) - 0.17510 (\ln Re)^2$$

$$+ 0.01087 (\ln Re)^3 - 0.00025 (\ln Re)^4$$

$$K_u = -2.38878 + 1.2502 (\ln 290) - 0.17510 (\ln 290)^2$$

$$+ 0.01087 (\ln 290)^3 - 0.00025 (\ln 290)^4$$

$$= 0.794$$

The required orifice area with calculated  $K_u = 0.794$  is

$$A = \frac{(310) \sqrt{1.45}}{38 (0.62) (0.92) (1.0) (1.0) (0.794) \sqrt{1.25 (100) - 25}}$$

$$= 2.17 \text{ in.}^2$$

The nearest standard orifice area is 2.853 in.<sup>2</sup>, with *L* designation having preferred valve body sizes, 3–4 in. or 4–6 in.

The maximum liquid flow rate with the standard orifice area is

$$V_L = A (38) (K_d) (K_w) (K_c) (K_p) (K_u) \sqrt{(1.25 P_1 - P_2) / G}$$

$$= 2.853 (38) (0.62) (0.92) (1.0) (0.794) (1.0)$$

$$\sqrt{(1.25 (100) - 25) / 1.45}$$

$$= 407.76 \text{ gpm}$$

Substituting the values in Eq. (9-14),

$$A = \frac{V_L \sqrt{G}}{38 K_d K_w K_c K_u \sqrt{P_1 - P_2}}$$

$$A = \frac{(310) \sqrt{1.45}}{38 (0.62) (0.92) (1) (1) \sqrt{100 - 25}}$$

$$= 1.989 \text{ in.}^2$$

The error in the orifice area between the ASME code and Non-ASME code is  $\epsilon = \frac{(1.989 - 1.722)}{1.722} \times 100\% = 15.5\%$

Percentage deviation in the calculated orifice areas between the ASME code valves and Non-ASME code Liquid Valves [5a] non-board certified for liquids is 15.5%. This is because the Non-ASME code uses a 25% overpressure, whereas the ASME code formula is based on only 10%. It should be noted that Eq. (9-15) is applicable only to relief valves not requiring capacity certification (Section 4.6 of API 520). Table 9-7 shows computer results of Example 9-4.

$V$  = required gas capacity, scfm

$V_a$  = required air capacity, scfm

$V_L$  = required liquid capacity, gpm (dm<sup>3</sup>/min)

$G$  = specific gravity of gas (air = 1.0) or specific gravity of liquid (water = 1.0) at actual discharge temperature.

A specific gravity at any lower temperature will obtain a safe valve size.

$M$  = average molecular weight of vapor

$P_1$  = relieving pressure in psia = [set pressure, psig + overpressure, psig + 14.7] psia. Minimum overpressure = 3 psi

$P_1$  = set pressure at inlet, psig

$P_2$  = back pressure at outlet, psig

$\Delta P$  = Set pressure + overpressure, psig – back pressure, psig. At 10% overpressure  $\Delta P = 1.1 P_1 - P_2$ . Below 30 psig set pressure,  $\Delta P = P_1 + 3 - P_2$ .

$T$  = inlet temperature of absolute, °R = (°F + 460)

$Z$  = compressibility factor corresponding to  $T$  and  $P$ . If this factor is not available, compressibility correction can be safely ignored by using a value of  $Z = 1.0$ .

$C$  = gas or vapor flow constant, Figure 9-25

$k$  = ratio of specific heats,  $C_p/C_v$ . If a value is not known the use of  $k = 1.001$ ,  $C = 315$  will result in a safe valve size. Isentropic coefficient,  $n$ , may be used instead of  $k$  (Table 9-5).

**TABLE 9-6**

Vapor Relief Valve Sizing:	Valve Name:	Example 9-3
Required relief vapor rate, lb/hr.:		620.000
Valve set pressure, psig:		290.000
Percentage of valve overpressure (%):		10.000
Upstream fluid temperature, °F:		133.4000
Fluid compressibility factor:		0.8370
Fluid molecular weight:		17.030
Ratio of specific heats of fluid $K = Cp/Cv$ :		1.310
Gas constant determined by ratio of K:		347.913
Calculated valve orifice area, in <sup>2</sup> :		0.030
Nearest standard orifice area, in <sup>2</sup> :		0.110
Nearest standard orifice size:	D	
Preferred valve body size (In-Out) inches:	1-2	
Maximum vapor flowrate with std. orifice, lb/hr.		2305.658
Reaction force, lb:		28.0

**TABLE 9-7**

Liquid Relief Valve Sizing:	Valve Name:	Example 9-4
Required liquid flowrate, U.S. gpm:		310.000
Valve set pressure, psig:		100.000
Valve back pressure, psig:		25.000
Fluid specific gravity:		1.4500
Fluid viscosity, cP:		3200.000
Viscosity correction factor:		1.000
Back pressure correction factor:		0.920
Overpressure correction factor:		1.000
Calculated valve orifice area, in <sup>2</sup> :		1.722
Nearest standard orifice area, in <sup>2</sup> :		1.838
Nearest standard orifice size:	K	
Preferred valve body size (In-out) inches:		3-4
Maximum liquid flowrate with std. orifice, lb/hr.:		330.845
Viscosity correction factor:		0.794
Reynolds number:		290.
Calculated valve orifice area, in <sup>2</sup> :		2.170
Nearest standard orifice area, in <sup>2</sup> :		2.853
Nearest standard orifice size:	L	
Preferred valve body size (In-out) inches:		3-4 or 4-6
MAXIMUM LIQUID FLOWRATE, gpm:		408

$K_p$  = liquid capacity correction factor for overpressures lower than 25% for non-code liquids equations only (see Figure 9-22)

$K_b$  = vapor or gas flow correction factor for constant back pressures above critical pressure (see Figure 9-26)

$K_v$  = vapor or gas flow factor for variable back pressures for balanced seal valves only (see Figures 9-27a and b)

$K_w$  = liquid flow factor for variable back pressures for balanced seal valves only (see Figure 9-28). For atmosphere,  $K_w = 1.0$

$K_u$  = liquid viscosity correction factor (see Figure 9-23 or 9-24).

$K_{sh}$  = steam superheat correction factor (see Table 9-8).

$K_n$  = Napier steam correction factor for set pressures between 1500 and 2900 psig (see Table 9-9).

$K_d$  = coefficient of discharge [42].\*

0.953 for air, steam, vapors, and gases<sup>†</sup>

0.724 for ASME code liquids

0.64 for non-ASME code liquids

0.62 for rupture disks and non-reclosing spring loaded devices ASME [1], Par. UG-127.

To convert flow capacity from scfm to lb/h use

$$W = \frac{(M)(V)}{6.32}$$

where

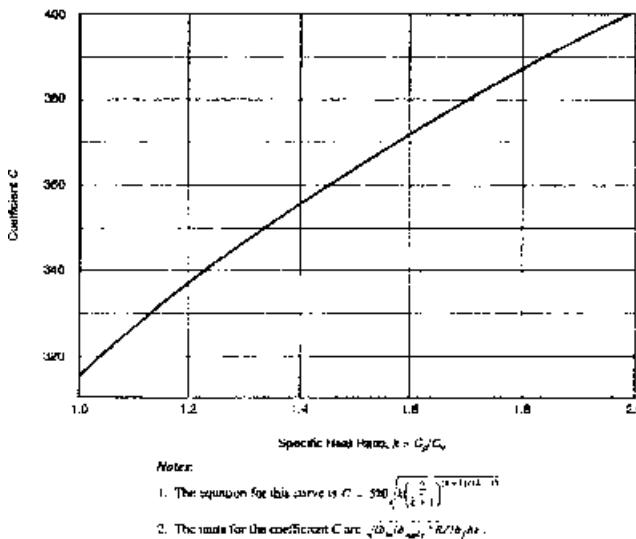
$M$  = molecular weight of flowing media

$V$  = flow capacity, scfm

$W$  = flow capacity, lb/h.

\*0.975 per API RP-520, balanced valve.

<sup>†</sup>Some manufacturers' National Board Certified Tests will have different values for some of their valves. Be sure to obtain the manufacturer's certified coefficient for the valve you select.



**Figure 9-25** Constant “C” for gas or vapor related to specific heats. (By permission from *Sizing, Selection and Installation of Pressure-Relieving Devices in Refineries*, Part I “Sizing and Selection”, API RP-520, 5th ed., Jul 1990.)

Where the pressure-relief valve is used in series with a rupture disk, a combination capacity of 0.8 must be applied to the denominator of the referenced equations. Refer to a later section of this text or to specific manufacturers.

### 9.23 SIZING VALVES FOR LIQUID EXPANSION (HYDRAULIC EXPANSION OF LIQUID FILLED SYSTEMS/EQUIPMENT/PIPING)

The API code RP-520 [5a] suggests the following to determine the liquid expansion rate to protect liquid-filled (full) systems or locations where liquid could be trapped in parts of a system or an area could be subjected to blockage by process or operational accident. When thermal input from any source can/could cause thermal expansion of the enclosed liquid:

$$\text{GPM} = BH / (500 GC_h) \quad (9-24)$$

This relation can be converted to solve for the required orifice area at 25% overpressure for non-viscous liquids discharging to atmosphere [31]

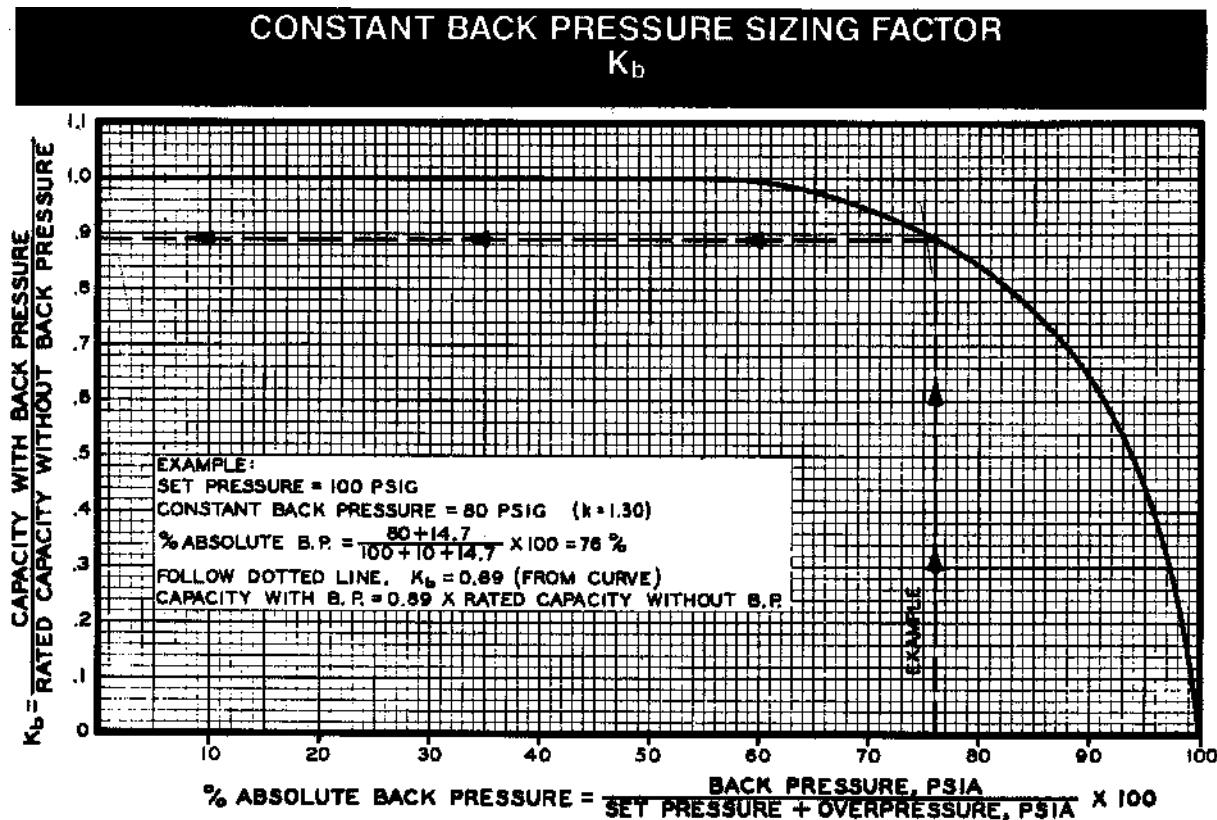
$$A = BH / (13600) (P_1 G)^{0.5}, \text{ in.}^2 \quad (9-24a)$$

where

GPM = flow rate at the flowing temperature, gpm

B = cubical expansion coefficient per °F for the liquid at the expected temperature. For values, see specific liquid data or see table below for typical values.

H = total heat transfer rate, Btu/h. For heat exchangers, reference [5a] recommends that this value be taken as the maximum exchanger duty during operation.



**Figure 9-26** Constant back pressure sizing factor,  $K_b$ , conventional valves – vapors and gases. (By permission from Teledyne Farris Engineering Co.)

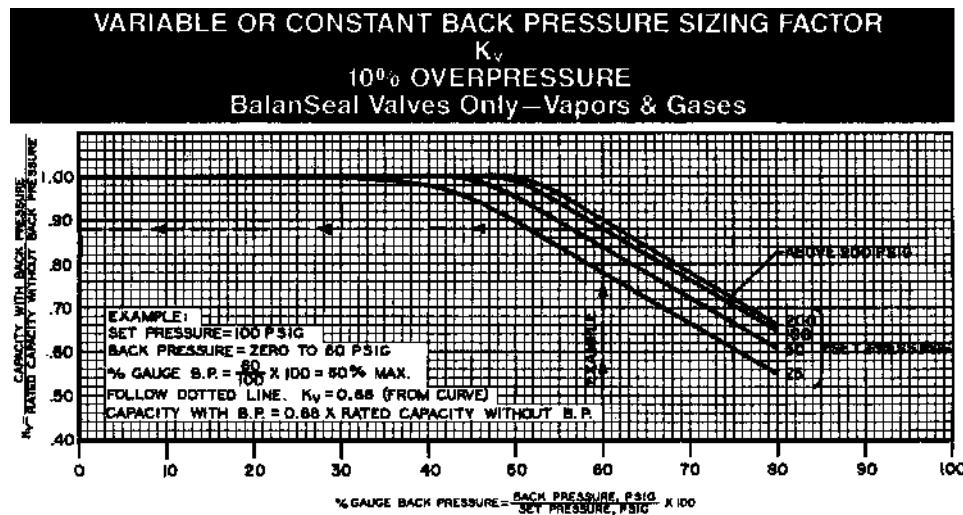


Figure 9-27a Variable or constant back pressure sizing factor,  $K_v$ , at 10% overpressure. BalanSeal® valves only – vapors and gases. (By permission from Teledyne Farris Engineering Co.)

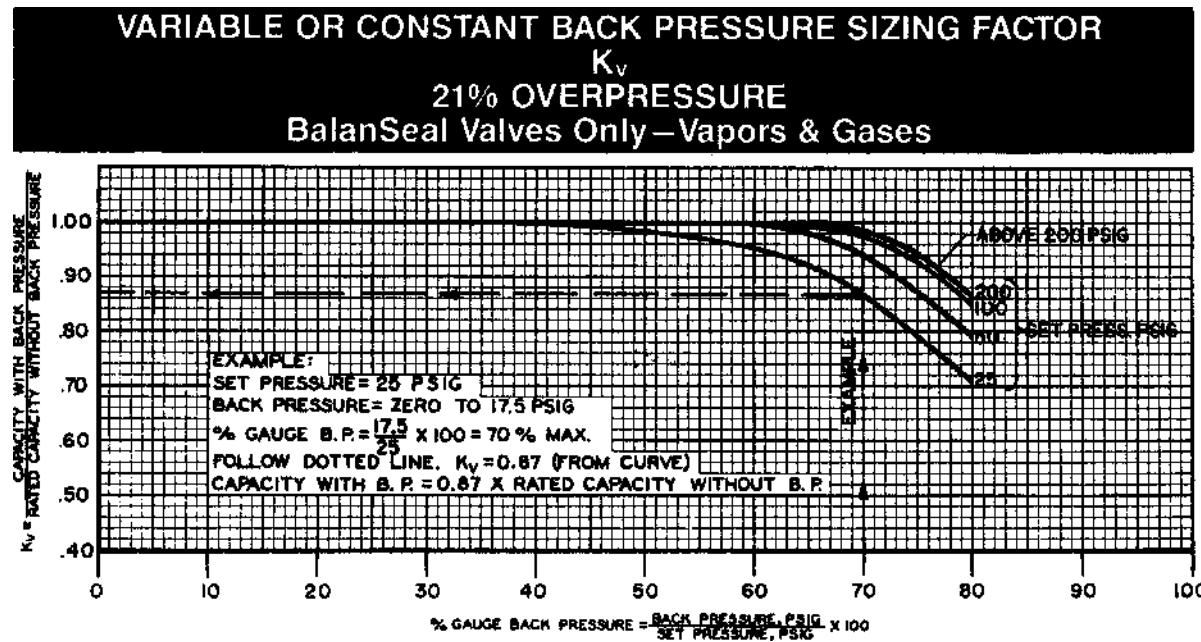


Figure 9-27b Variable or constant back pressure sizing factor,  $K_v$ , at 21% overpressure, BalanSeal® valves only – vapors and gases. (By permission from Teledyne Farris Engineering Co.)

$G$  = specific gravity referred to water. Ignore fluid compressibility.

$C_h$  = specific heat of the trapped fluid, Btu/lb/°F

$A$  = required orifice area, in.<sup>2</sup>

$P_1$  = set pressure of valve, psig.

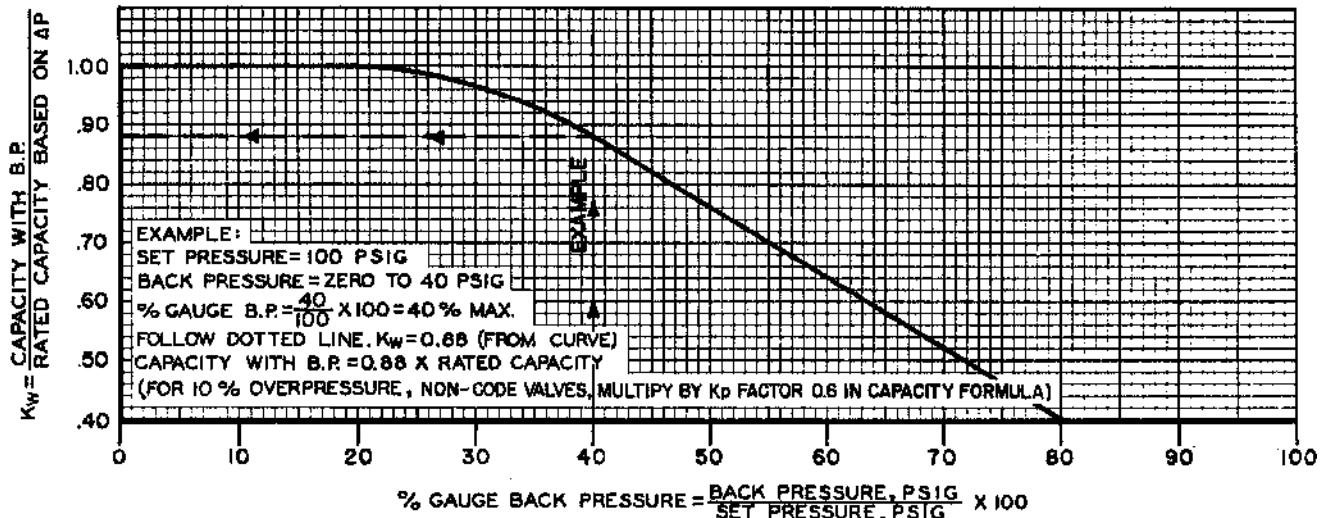
Reference [5a] in Appendix D cautions that if the vapor pressure of the fluid at the temperature is greater than the relief set/ design pressure, then the valve must be capable of handling the rate of vapor generation. Other situations should be examined as the thermal relief by itself may be insufficient for total relief.

#### Typical Values for Cubic Expansion Coefficient

Liquid	Value
3–34.9 degree API gravity	0.0004
35–50.9 degree API gravity	0.0005
51–63.9 degree API gravity	0.0006
64–78.9 degree API gravity	0.0007
79–88.9 degree API gravity	0.0008
89–93.9 degree API gravity	0.00085
94–100 degree API gravity and lighter	0.0009
Water	0.0001

(Source: By permission from API [5a])

### VARIABLE OR CONSTANT BACK PRESSURE SIZING FACTOR $K_w$



**Figure 9-28** Variable or constant back pressure sizing factor,  $K_w$ , for liquids only, BalanSeal® valves. Use this factor as a divisor to results of constant back pressure equations or tables. (By permission from Teledyne Farris Engineering Co.)

Note: Reference of the API gravity values to refinery and petrochemical plant fluids will show that they correspond to many common hydrocarbons.

#### 9.24 SIZING VALVES FOR SUBCRITICAL FLOW: GAS OR VAPOR BUT NOT STEAM [5D]

If the ratio of back pressure to inlet pressure to valve exceeds the critical pressure ratio,  $P_c/P_1$

$$\left(\frac{P_c}{P_1}\right) = \left[\frac{2}{(k+1)}\right]^{k/(k-1)}$$

the flow through the valve is subcritical. The required area (net, free unobstructed) is calculated for a conventional relief valve, including sizing a pilot-operated relief valve [5d]:

$$A = \frac{W}{735F_2K_dK_c} \sqrt{\frac{ZT}{MP_1(P_1 - P_2)}}, \text{ in.}^2 \quad (9-25a)$$

or

$$A = \frac{V}{4645F_2K_dK_c} \sqrt{\frac{ZTM}{P_1(P_1 - P_2)}}, \text{ in.}^2 \quad (9-25b)$$

or

$$A = \frac{V}{864F_2K_dK_c} \sqrt{\frac{ZTG}{P_1(P_1 - P_2)}}, \text{ in.}^2 \quad (9-25c)$$

In SI units

$$A = \frac{17.9W}{F_2K_dK_c} \sqrt{\frac{ZT}{MP_1(P_1 - P_2)}}, \text{ mm}^2 \quad (9-26a)$$

or

$$A = \frac{47.95V}{F_2K_dK_c} \sqrt{\frac{ZTM}{P_1(P_1 - P_2)}}, \text{ mm}^2 \quad (9-26b)$$

or

$$A = \frac{258V}{F_2K_dK_c} \sqrt{\frac{ZTG}{P_1(P_1 - P_2)}}, \text{ mm}^2 \quad (9-26c)$$

When using a balanced/bellows relief valve in the sub-critical, use Eqs (9-18–9-22); however, the back pressure correction factor for this condition should be supplied by the valve manufacturer [5d]. For sub-critical, conventional valve

$$F_2 = \sqrt{\left[\frac{k}{k-1}\right](r)^{2/k} \left[\frac{1-(r)^{(k-1)/k}}{1-r}\right]} \quad (9-27)$$

where

$A$  = required effective discharge area of the device, in.<sup>2</sup> (mm<sup>2</sup>)

$W$  = required flow through the device, lb/h (kg/h)

$F_2$  = coefficient of subcritical flow

$$= \sqrt{\left(\frac{k}{k-1}\right)(r)^{2/k} \left(\frac{1-r^{(k-1)/k}}{1-r}\right)} \quad (\text{Figure 9-29})$$

$r$  = ratio of back pressure to upstream relieving pressure,  $P_2/P_1$

$K_d$  = effective coefficient of discharge. For preliminary sizing, use the following values:

- (i) 0.975, when a pressure-relief valve is installed with or without a rupture disk in combination;
- (ii) 0.62, when a pressure-relief valve is not installed and sizing is for a rupture disk.

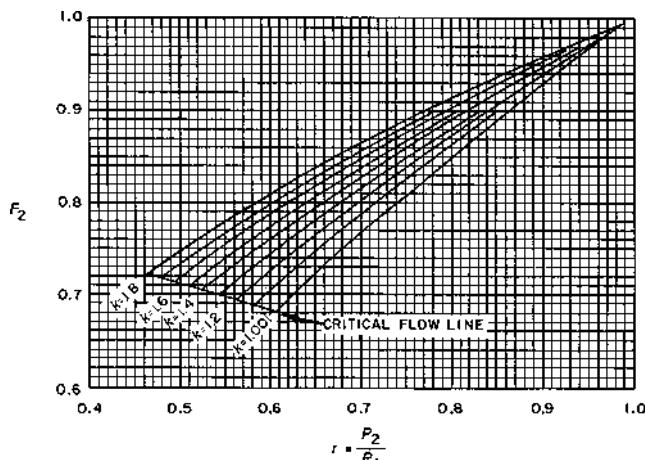
**TABLE 9-8 Steam Superheat Correction factors,  $K_{sh}$**

Set Pressure psig	Saturated steam Temperature °F.	Total Temperature in Degrees Fahrenheit																																					
		280	300	320	340	360	380	400	420	440	460	480	500	520	540	560	580	600	620	640	660	680	700	720	740	760	780	800	820	840	860	880	900	920	940	960	980	1000	
15	250	100	100	100	99	99	98	98	97	96	95	94	93	92	91	90	89	88	87	86	86	85	84	83	83	82	81	81	80	79	79	78	78	77	76	76	75	75	
20	259	100	100	100	99	99	98	98	97	96	95	94	93	92	91	90	89	88	87	86	86	85	84	83	83	82	81	81	80	79	79	78	78	77	77	76	75	75	
40	278	-	100	100	100	99	99	98	97	96	95	94	93	92	91	90	89	88	87	87	86	85	84	84	83	82	82	81	80	79	79	78	78	77	77	76	75	75	
60	308	-	-	100	99	99	98	97	96	95	94	93	92	91	90	89	88	87	87	86	85	84	84	83	82	82	81	80	80	79	78	78	77	77	76	75	75		
80	324	-	-	-	100	100	99	99	98	97	96	94	93	92	91	90	89	89	88	87	86	85	84	84	83	82	82	81	80	80	79	78	78	77	77	76	75	75	
100	338	-	-	-	-	100	100	99	98	97	96	95	94	93	92	91	90	89	88	87	86	85	85	84	83	82	81	80	80	79	78	78	77	77	76	75	75		
120	350	-	-	-	-	-	100	100	99	98	97	96	95	94	93	92	91	90	89	88	87	86	85	85	84	83	82	82	81	80	80	79	78	78	77	77	76	75	
140	361	-	-	-	-	-	100	100	99	98	96	95	94	93	92	91	90	89	88	87	86	85	85	84	83	82	82	81	80	80	79	78	78	77	77	76	75		
160	371	-	-	-	-	-	100	100	99	98	97	95	94	93	92	91	90	89	88	87	86	86	85	84	83	82	82	81	80	80	79	79	78	77	77	76	75		
180	380	-	-	-	-	-	-	100	99	98	97	96	95	93	92	91	90	89	88	87	86	86	85	84	83	82	82	81	80	80	79	79	78	77	77	76	75		
200	388	-	-	-	-	-	-	100	99	99	97	96	95	93	92	91	90	89	88	87	86	86	85	84	83	82	81	81	80	79	79	78	77	77	76	75			
220	395	-	-	-	-	-	-	-	100	100	99	98	96	95	94	93	92	91	90	89	88	87	86	85	84	84	83	82	81	81	80	79	79	78	78	77	76	75	
240	403	-	-	-	-	-	-	-	100	99	98	97	95	94	93	92	91	90	89	88	88	87	86	85	84	84	83	82	82	81	81	80	79	78	78	77	76	75	
260	409	-	-	-	-	-	-	-	100	99	98	97	96	94	93	92	91	90	89	88	87	86	86	85	85	84	83	82	81	81	80	79	79	78	78	77	76	75	
280	416	-	-	-	-	-	-	-	100	100	99	97	96	95	93	92	91	90	89	88	87	86	86	85	85	84	83	82	82	81	80	80	79	78	78	77	76	75	
300	422	-	-	-	-	-	-	-	-	100	99	99	98	96	95	93	92	91	90	89	88	87	86	86	85	84	83	82	82	81	80	80	79	79	78	77	76	75	
350	436	-	-	-	-	-	-	-	-	100	99	97	96	94	93	92	91	90	89	88	87	86	85	85	84	83	83	82	81	81	80	79	78	78	77	76	76		
400	448	-	-	-	-	-	-	-	-	100	99	98	96	95	93	92	91	90	89	88	87	86	85	84	84	83	82	81	81	80	79	79	78	77	77	76	76		
450	460	-	-	-	-	-	-	-	-	100	99	97	96	94	93	92	91	90	89	88	87	86	86	85	84	83	82	82	81	81	80	79	79	78	78	77	76	76	
500	470	-	-	-	-	-	-	-	-	100	99	98	97	96	94	93	92	91	90	89	88	87	86	86	85	85	84	83	82	82	81	81	80	79	79	78	78	77	76
550	480	-	-	-	-	-	-	-	-	100	99	97	95	94	92	91	90	89	88	87	86	86	85	85	84	83	82	82	81	81	80	80	79	78	78	77	76		
600	489	-	-	-	-	-	-	-	-	100	99	98	96	94	93	92	90	89	88	87	86	86	85	84	83	82	82	81	81	80	80	79	78	78	77	76			
650	497	-	-	-	-	-	-	-	-	100	99	97	96	94	93	92	91	90	89	88	87	86	86	85	84	83	82	82	81	81	80	79	79	78	77	77			
700	506	-	-	-	-	-	-	-	-	100	99	97	96	94	93	92	91	90	89	88	87	86	86	85	84	83	82	81	81	80	79	79	78	77	77				
750	513	-	-	-	-	-	-	-	-	100	100	98	96	95	93	92	90	89	88	87	86	86	85	84	83	83	82	81	81	80	79	79	78	77	76				
800	520	-	-	-	-	-	-	-	-	-	100	99	97	95	94	92	91	90	89	88	87	86	86	85	84	84	83	82	81	80	80	79	78	78	77				
850	527	-	-	-	-	-	-	-	-	-	100	99	97	95	94	92	91	90	89	88	87	86	86	85	84	83	82	82	81	81	80	79	78	78	77				
900	533	-	-	-	-	-	-	-	-	-	100	100	99	97	95	93	92	90	89	88	87	86	86	85	84	83	82	81	81	80	79	79	78	77					
950	540	-	-	-	-	-	-	-	-	-	100	99	97	96	94	93	91	90	89	88	87	86	85	84	83	82	82	81	81	80	79	79	78	77					
1000	546	-	-	-	-	-	-	-	-	-	100	99	98	96	94	93	91	90	89	88	87	86	86	85	84	83	83	82	81	81	80	79	79	78	77				
1050	552	-	-	-	-	-	-	-	-	-	100	100	99	97	95	93	92	90	89	88	87	86	86	85	84	84	83	82	81	81	80	79	79	78	77				
1100	558	-	-	-	-	-	-	-	-	-	-	100	99	98	95	94	92	91	89	88	87	86	86	85	84	83	82	81	81	80	79	78	78	77					
1150	563	-	-	-	-	-	-	-	-	-	-	100	99	98	96	94	92	91	90	88	87	86	86	85	84	83	82	81	81	80	79	78	78	77					
1200	569	-	-	-	-	-	-	-	-	-	-	100	99	98	97	95	93	91	90	89	88	87	86	85	84	83	82	81	81	80	79	78	78	77					
1250	574	-	-	-	-	-	-	-	-	-	-	100	99	97	95	93	92	90	89	88	87	86	85	84	83	82	81	80	79	79	78	77	77						
1300	579	-	-	-	-	-	-	-	-	-	-	100	99	98	96	94	92	91	90	89	88	87	86	85	84	83	82	81	81	80	79	79	78	77					
1350	584	-	-	-	-	-	-	-	-	-	-	100	99	98	96	94	93	91	90	88	87	86	85	84	83	82	81	81	80	79	78	78	77						
1400	588	-	-	-	-	-	-	-	-	-	-	100	100	99	97	95	93	92	90	89	88	86	85	84	83	82	81	81	80	79	78	78	77						
1450	593	-	-	-	-	-	-	-	-	-	-	-	100	99	98	96	94	92	90	89	88	86	85	84	83	82	81	81	80	79	78	78	77						
1500	597	-	-	-	-	-	-	-	-	-	-	-	100	99	98	97	95	94	92	91	89	88	87	86	85	84	83	82	81	81	80	79	78	78	77				
1600	606	-	-	-	-																																		

**TABLE 9-9 Napier Steam Correction Factors,  $K_n$ , for Set Pressures Between 1500 and 2900 psig**

Set pressure (psig)	$K_n$																			
1500	1.005	1640	1.014	1780	1.025	1920	1.037	2060	1.050	2200	1.066	2340	1.083	2480	1.104	2620	1.128	2760	1.157	
1510	1.005	1650	1.015	1790	1.026	1930	1.038	2070	1.051	2210	1.067	2350	1.085	2490	1.105	2630	1.130	2770	1.159	
1520	1.006	1660	1.016	1800	1.026	1940	1.039	2080	1.052	2220	1.068	2360	1.086	2500	1.107	2640	1.132	2780	1.161	
1530	1.007	1670	1.016	1810	1.027	1950	1.040	2090	1.053	2230	1.069	2370	1.087	2510	1.109	2650	1.134	2790	1.164	
1540	1.007	1680	1.017	1820	1.028	1960	1.040	2100	1.054	2240	1.070	2380	1.089	2520	1.110	2660	1.136	2800	1.166	
1550	1.008	1690	1.018	1830	1.029	1970	1.041	2110	1.055	2250	1.072	2390	1.090	2530	1.112	2670	1.138	2810	1.169	
1560	1.009	1700	1.019	1840	1.030	1980	1.042	2120	1.057	2260	1.073	2400	1.092	2540	1.114	2680	1.140	2820	1.171	
1570	1.009	1710	1.019	1850	1.031	1990	1.043	2130	1.058	2270	1.074	2410	1.093	2550	1.115	2690	1.142	2830	1.174	
1580	1.010	1720	1.020	1860	1.031	2000	1.044	2140	1.059	2280	1.075	2420	1.095	2560	1.117	2700	1.144	2840	1.176	
1590	1.011	1730	1.021	1870	1.032	2010	1.045	2150	1.060	2290	1.077	2430	1.096	2570	1.119	2710	1.146	2850	1.179	
1600	1.011	1740	1.021	1880	1.033	2020	1.046	2160	1.061	2300	1.078	2440	1.098	2580	1.121	2720	1.148	2860	1.181	
1610	1.012	1750	1.023	1890	1.034	2030	1.047	2170	1.062	2310	1.079	2450	1.099	2590	1.122	2730	1.150	2870	1.184	
1620	1.013	1760	1.023	1900	1.035	2040	1.048	2180	1.063	2320	1.081	2460	1.101	2600	1.124	2740	1.152	2880	1.187	
1630	1.014	1770	1.024	1910	1.036	2050	1.049	2190	1.049	2330	1.082	2470	1.102	2610	1.126	2750	1.155	2890	1.189	
																			2900	1.192

Courtesy of Teledyne-Farris Engineering Co., Cat. 187C.



**Figure 9-29** Values of  $F_2$  for subcritical flow of gases and vapors.  
(Reprinted by permission from *Sizing, Selection and Installation of Pressure Relieving Devices in Refineries*, Part I "Sizing and Selection", API RP-520, 5th ed. Jul 1990, American Petroleum Institute)

- $K_c$  = combination correction factor for installation with a rupture disk upstream of the pressure-relief valve  
= 1.0, when a rupture disk is not installed  
= 0.9, when a rupture disk is installed in combination with a pressure-relief valve and the combination does not have a published value
- $Z$  = compressibility factor for the deviation of the actual gas from a perfect gas, evaluated at relieving inlet conditions
- $T$  = relieving temperature of the inlet gas or vapor,  
 $^{\circ}\text{R} = ^{\circ}\text{F} + 460 (K = ^{\circ}\text{C} + 273)$
- $M$  = molecular weight of the gas or vapor
- $P_1$  = upstream relieving pressure, psia (kPa). This is the set pressure plus the allowable overpressure plus atmospheric pressure.
- $P_2$  = back pressure, psia (kPa)
- $V$  = required flow through the device, SCFM at 14.7 psia and 60° F ( $\text{Nm}^3/\text{min}$  at 101.325 kPa and 0° C)
- $G$  = specific gravity of gas at standard conditions referred to air at standard conditions (normal conditions).  $G = 1$  for air at 14.7 psia and 60° F (101.325 kPa and 0° C).

## 9.25 EMERGENCY PRESSURE RELIEF: FIRES AND EXPLOSIONS RUPTURE DISKS

Process systems can develop pressure conditions that cannot timely or adequately be relieved by pressure-relieving valves as described earlier. These conditions are primarily considered to be (1) internal process explosions due to runaway reactions (see Design Institute for Emergency Relief Systems (DIERS) [13]) in pressure vessels or similar containers such as an atmospheric grain storage silo (dust explosion typically) or storage bin; (2) external fires developed around, under, or encompassing a single process vessel or a system of process equipment, or an entire plant; and (3) other conditions in which rapid/instantaneous release of developed pressure and large volumes of vapor/liquid mixture is vital to preserve the integrity of the equipment. For these conditions, a rupture disk may perform a vital safety-relief function. Sometimes the combination of a rupture disk and pressure-relieving valve will satisfy a prescribed situation, but the valve cannot be relied on for instantaneous release (response time lag of usually a few seconds).

The ASME Pressure Vessel Code [1] and the API codes or recommended procedures [5, 8, and 9] recognize and set regulations

and procedures for capacity design, manufacture, and installation of rupture disks, once the user has established the basis of capacity requirements.

## 9.26 EXTERNAL FIRES

There have been at least six different formulas proposed and used to determine the proper and adequate size of rupture disk openings for a specific relieving condition. The earlier studies of Sylvander and Katz [33] led to the development of the ASME and API recommendations. This approach assumes that a fire exists under or around the various vessels in a process. This fire may have started from static discharge around flammable vapors, flammable liquids released into an area drainage ditch, combustible gas/vapors released through flange leaks or ruptures, overpressure, or many other potential hazards. The codes [9] suggest typical lists of potential hazards and some approach to determine the types of process conditions that can cause a fire. A suggested extended list (Figure 9-14a) is presented earlier in this chapter and in the following paragraphs. There are no formulas to establish a code capacity for the volume of vapors to be released under anyone of the possible plant "failure" conditions. Therefore, the codes assume, based on evaluation of test data, that fire is under or around the various vessels, and that absorbed heat vaporizes the contained fluid. The information presented is taken from API-RP-520/521 latest editions [5, 8, and 9] and the ASME code [1]. The designer should be familiar with the details in these codes.

## 9.27 SET PRESSURES FOR EXTERNAL FIRES

The MAWP (discussed earlier) for the vessel or each vessel should be the maximum set pressure for the rupture disk. Furthermore, estimated flame temperatures of usually 2500–3500° F should be used to establish the reduced vessel metal wall temperatures (recognizing the benefits of code recommended fireproof insulation if properly applied to prevent dislodging by fire water hose pressures impacting on the insulation). The MAWP should then be re-established by calculation using the metal wall code allowable stresses at the new estimated reduced metal temperature. This should be the maximum set pressure for the rupture disk provided the new lower value does not cause it to be below or too close to the usual expected process operating temperature. In such a case, the set pressure should be 25% above the operating condition, exclusive of fire, not exceeding the MAWP values [34].

When a rupture disk relieves/blows/ruptures, it creates a rapid depressuring of the process system and a likely discharge of some or all vapor/liquid in the vessel(s), discharging to the properly designed disposal system. Therefore, great care should be given to setting the rupture disk pressure because it does not have an accumulation factor, but bursts at the prescribed pressure of the disk, taking into account the code allowed manufacturing tolerances. Two-phase flow will most likely occur when the disk (or safety relief valve) blows (see later references to explosions and DIERS work [13, 22]).

For unexpected runaway or process overpressure not subject to external fire, the rupture disk set pressure, which is the bursting pressure, should be sufficiently higher than the expected "under acceptable control" conditions for the operation to avoid the frequent burst and shutdown of the process. Usually this is found to be about 20–30% above the maximum expected peak operation pressure. Again, recognize the requirements for the relieving device set pressure not to exceed the actual vessel MAWP at the expected relieving temperature (by calculation or pilot plant test data).

## 9.28 HEAT ABSORBED

The amount of heat absorbed by a vessel exposed to an open fire is markedly affected by the size and character of the installation and by the environment. These conditions are evaluated by the following equivalent formulas, in which the effect of size on the heat input is shown by the exponent of  $A_w$ , the vessel wetted area, and the effect of other conditions, including vessel external insulation, is included in a factor  $F$ [5]:

$$q = 21,000 FA_w^{-0.18} \quad (9-28)$$

$$Q = 21,000 FA_w^{+0.82} \quad (9-29)$$

### THE SEVERE CASE

Where the facility does not have prompt fire fighting equipment and inadequate drainage of flammable materials away from the vessel,

$$Q = 34,500 FA_w^{+0.82} \quad (9-30)$$

where

$q$  = average unit heat absorption, in BTU/h/ft<sup>2</sup> of wetted surface.

$Q$  = total heat absorption (input) to the wetted surface, Btu/h.

$A_w$  = total wetted surface, ft<sup>2</sup>.

The expression  $A_w^{-0.18}$  is the area exposure factor or ratio.

This recognizes that large vessels are less likely to be completely exposed to the flame of an open fire than small vessels. It is recommended that the total wetted surface ( $A$  in the foregoing formulas) be limited to that wetted surface included within a height of 25 ft above

"grade" or, in the case of spheres and spheroids, to the elevation of the maximum horizontal diameter or a height of 25 ft, whichever is greater. (A more conservative approach is recommended.) The term "grade" usually refers to ground grade, but may be any level at which a sizable area of exposed flammable liquid could be present [5, 8].

$F$  = environment factor, values of which are shown in Table 9-10 for various types of installation.

Surface areas of vessel elliptical heads can be estimated by

1.15 × cross-sectional area of vessel.

These are the basic formulas for the usual installation, with good drainage and available fire-fighting equipment. These formulas are plotted on Figure 9-30a showing curves for  $Q$  for various values of factor  $F$ . The approximate amount of insulation corresponding to the factors is indicated.

Referring to the wetted surface,  $A_w$ , the surface areas of ASME flanged and dished head, ASME elliptical heads, hemispherical heads, and so on are often the end assemblies on a cylindrical vessel. If a formula is not available to accurately estimate the wetted surface, or the blank diameters used for fabrication (see Appendix), which would give a close approximation of the inside surface of the head, use an estimated area for the dished or elliptical heads as 1.2 × cross-section area of the vessel based on its diameter.

## 9.29 SURFACE AREA EXPOSED TO FIRE

The surface area of a vessel exposed to fire which is effective in generating vapor is that area wetted by its internal liquid contents. The liquid contents under variable level conditions should ordinarily be taken at the average inventory (e.g., see note below).

TABLE 9-10 Environmental Factor,  $F$

Type of Installation	Factor*
Bare vessel	1.0
Insulated vessel <sup>†</sup> (These arbitrary insulation conductance values are shown as examples and are in British Thermal Units/h/ft <sup>2</sup> /°F)	
(a) 4.0 Btu/h/ft <sup>2</sup> /°F (1 in. thick)	0.3
(b) 2.0 Btu/h/ft <sup>2</sup> /°F (2 in. thick)	0.15
(c) 1.0 Btu/h/ft <sup>2</sup> /°F (4 in. thick)	0.075
(d) 0.67 Btu/h/ft <sup>2</sup> /°F	0.05
(e) 0.5 Btu/h/ft <sup>2</sup> /°F	0.0376
(f) 0.4 Btu/h/ft <sup>2</sup> /°F	0.03
(g) 0.32 Btu/h/ft <sup>2</sup> /°F	0.026
Water-application facilities, on bare vessel <sup>‡</sup>	1.0
Depressurizing and emptying facilities <sup>§</sup>	1.0
Underground storage	0.0
Earth-covered storage above grade	0.03

\* These are suggested values for the conditions assumed in code [5] Par. D 5.21. When these conditions do not exist, engineering judgment should be exercised either in selecting a higher factor in providing means of protecting vessels from fire exposure as suggested in [5], Par. D 8.

<sup>†</sup> Insulation shall resist dislodgement by fire hose streams. For example, a temperature difference of 1600°F was used. These conductance values are based on insulation having conductivity of 4Btu-in/hr-ft<sup>2</sup>-°F/in. at 1000°F and correspond to various thicknesses of insulation between 1 and 12 in.

<sup>‡</sup> See code for recommendations regarding water application and insulation.

<sup>§</sup> Depressurizing will provide a lower factor if done promptly, but no credit is to be taken when safety valves are being sized for fire exposure. See [35], Part I, Par. D 8.2.

By permission from API-RP-520, American Petroleum Institute, Div. of Refining (1967) and adapted for this current edition by this author from later editions of the code (197) and (1990). Items d, e, and f above from API-RP-520, 5th ed. (1990). For complete reference, see the latest code cited in its entirety.

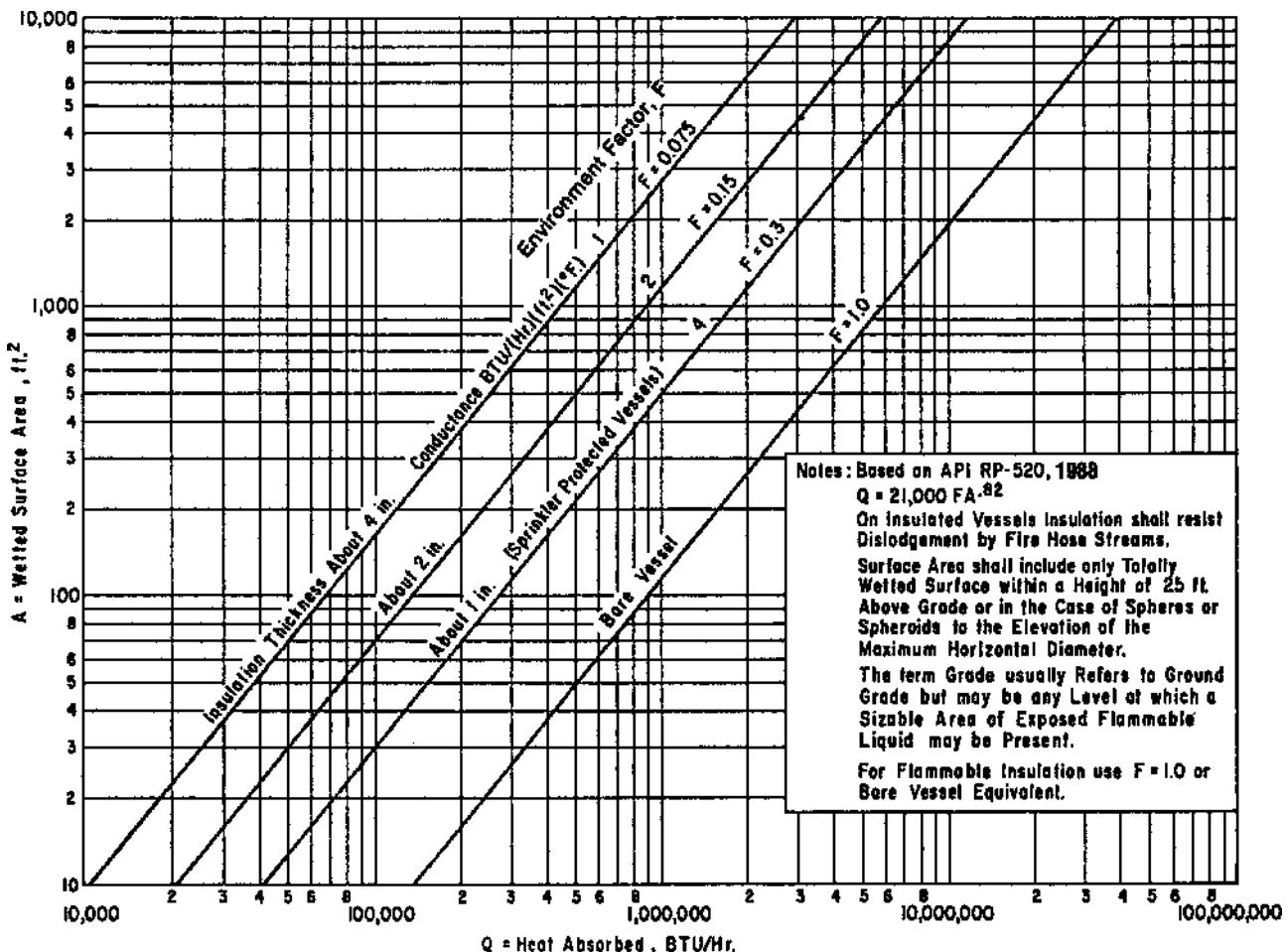


Figure 9-30a API formula for heat absorbed from fire on wetted surface of pressure vessel. (Prepared by permission from *Sizing, Selection and Installation of Pressure Relieving Devices in Refineries*, Part I "Sizing and Selection", API RP-520, 5th ed. Jul 1990, American Petroleum Institute.)

1. **Liquid-full vessels** (such as treaters) operate liquid full. Therefore, the wetted surface would be the total vessel surface within the height limitation.
2. **Surge drums** (vessels) usually operate about half full. Therefore, the wetted surface would be calculated at 50% of the total vessel surface, but higher if design is based on greater figure.
3. **Knockout drums** (vessels) usually operate with only a small amount of liquid. Therefore, the wetted surface would be in proportion, but to maximum design liquid level.
4. **Fractionating columns** usually operate with a normal liquid level in the bottom of the column and a level of liquid on each tray. It is reasonable to assume that the wetted surface be based on the total liquid within the height limitation – both on the trays and in the bottom.
5. **Working storage tanks'** wetted surface is usually calculated on the average inventory, but at least 25 ft height, unless liquid level can reasonably be established as higher; if established as higher, then use higher value. This should be satisfactory not only because it conforms to a probability, but also because it provides a factor of safety in the time needed to raise the usually large volume of the liquid's sensible heat to its boiling point.

It is recommended that the wetted area be at least to the height as defined in the definition of area,  $A_w$ .

Note: E.E. Ludwig [34] suggested that determining  $A_w$  values may be more conservative and not conform exactly to Code [33a, c and d] recommendations. The Code [5a], Part 1, Sect D, Par. D.4] reads, "to determine vapor generation, only that portion of the vessel that is wetted by its internal liquid and is equal to or less than 25 feet above the source of flame needs to be recognized."

6. In E.E. Ludwig's experience [34] in investigating many industrial fires and explosions, it is suggested that the height limit of 25 ft above "grade" or fire source level is too low for many process plants, and therefore the effect of a large external fire around equipment can reach to 100 ft with 75 ft perhaps being acceptably conservative. Ludwig expressed concern in using the 25-ft limit, for example, for a horizontal butane storage "bullet" tank, 15 ft diameter and raised 15 ft off grade to its bottom. Furthermore, the fact that any fire will engulf the entire vessel should be considered and the wetted surface should be the entire vessel. The same concern applies to a vertical distillation column over 25 ft high. He suggested that the wetted surface should be at least 80% of the vessel height and further recognized that the tray liquid will wet the walls and be evaporated only as long as there is liquid to drain off the trays. But for a conservative approach, he assumed that there are always wetted walls in the column.

For packed columns, the wetted walls are at least to the top of the packing, with some entrainment height above that. Therefore, a vertical packed column of 60 ft of packing above the liquid level in the sump, which has a sump of 10 ft and a skirt of 10 ft, would be considered 70 ft of vertical height of wetted perimeter, not counting the skirt. If the wetted area reached a total height of 80 ft from grade, minus the unwetted area of the skirt height, because no liquid is in the skirt space, the 80 ft would not be used to establish the vertical height for fire exposure, but use 60 ft + 10 ft, or 70 ft wetted height. This approach is more conservative than the code [[5]a, c, and d], which is based on Ludwig's experience in investigating the damage levels from fires [34].

Each situation must be evaluated on its own merits or conditions and operating situation, and even its environment with respect to the plant flammable processing equipment.

### 9.30 RELIEF CAPACITY FOR FIRE EXPOSURE

In calculating the relief capacity to take care of external fire the following equation is used:

$$W = Q/L \quad (9-31)$$

where

$W$  = weight rate of flow of vapors, lb/h

$L$  = latent heat at allowable pressure, Btu/lb

$Q$  = total heat absorption from external fire, Btu/h.

### 9.31 CODE REQUIREMENTS FOR EXTERNAL FIRE CONDITIONS

Paragraph UG-125 (3) of the ASME code [1] requires that supplemental relieving capacity be available for an unfired pressure vessel subject to external accidental fire or other unexpected source of heat. For this condition, relieving devices must be installed to prevent the pressure from rising more than 21% [9] above the MAWP of the vessel. The set pressure should not exceed the vessel MAWP. A single relief device may be used for this capacity as long as it also meets the normal overpressure design for other possible causes of 10%. If desirable, multiple separate devices can be installed to satisfy both potential overpressure situations.

For this condition, the API-RP-521 code [9] (Figure 9-7a) shows an allowable 16% maximum accumulation relieving pressure above the set pressure. For external fire conditions on a vessel, the maximum allowable accumulation pressure is 21% above the set pressure [9] for both single or multiple relieving devices (Figure 9-7a).

### 9.32 DESIGN PROCEDURE

The usual procedure for determining relief area requirements is as follows.

- Determine the external surface area exposed to fire, as set forth by Eq. (9-29),

$$Q = 21,000 F A_w^{+0.82}$$

and Table 9-10 and Section 9.25.

- Determine the heat absorbed,  $Q$ , from Figure 9-30a.

- Calculate the rate of vaporization of liquid from Eq. (9-31)

$$W = \frac{Q}{L}$$

- Verify critical pressure from Eq. (9-7) and establish actual back pressure for relieving device.
- Calculate relieving area by applicable equation for critical or non-critical flow, using the flow rate determined in (3) above. (See Eq. (9-10) and following). The area actually selected for orifice of safety type valve must have orifice equal to or greater than calculated requirements. For a rupture disk application, the full free open cross-sectional area of pipe connections in inlet and exit sides must be equal to or be greater than the calculated area.
- Select a valve or rupture disk to accommodate the service application.
- To provide some external protection against the damage that an external fire can do to a pressure-relief valve or rupture disk, Ludwig [34] recommends that these devices be insulated after installation in such a manner as not to restrict their action but to provide some measure of reliable performance, even if the vessel is not insulated.

### 9.33 PRESSURE-RELIEF VALVE ORIFICE AREAS ON VESSELS CONTAINING ONLY GAS, UNWETTED SURFACE

Due to gas expansion from external fire, the API code [8] provides for calculation of the pressure-relief valve orifice area for a gas-containing vessel exposed to external fire on the unwetted surface:

$$A = F' A_3 / \sqrt{P_1}, \text{ in.}^2 \quad (9-32)$$

Based on air and perfect gas laws, vessel is uninsulated, and it will not reach rupture conditions. Review [5a] for specific design situations

where

$A$  = effective discharge area of valve, in.<sup>2</sup>

$A_3$  = exposed surface area of vessel, ft<sup>2</sup>

$F'$  = operating environment factor, minimum value recommended = 0.01, when the minimum value is unknown, use  $F' = 0.045$ , which can be calculated by [5c]

$$F' = \frac{0.1406}{C K_d} \left[ \frac{(T_\omega - T_1)^{1.25}}{T_1^{0.6506}} \right] \quad (9-33)$$

$P_1$  = upstream relieving pressure, in psia. This is the set pressure plus the allowable overpressure plus the atmospheric pressure, psia.

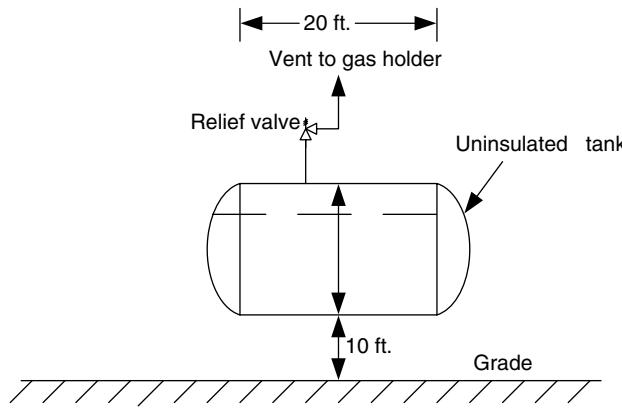
where

$C$  = coefficient determined by the ratio of the specific heats of the gas at standard conditions. This can be obtained from Equation 2 [5a] in 4.3.2.1 of API Recommended Practice 520, Part I, or Figure 9-25.

$K_d$  = coefficient of discharge (obtainable from the valve manufacturer).  $K_d$  is equal to 0.975 for sizing relief valves.

**EXAMPLE 9-5**

Calculate the required area for the relief valve for a horizontal non-insulated storage tank, exposed to fire and containing liquid vinyl chloride monomer ( $\text{CH}_2=\text{CHCl}$ ). The tank dimensions are shown in Figure 30b. The design pressure is 100 psig, and the discharge from the relief valve will be vented to a gas holder operating at 0.5 psig. A 20% accumulation (overpressure) is assumed over the design pressure. The average inventory of the tank contents will equal 75% of the vessel's inside diameter.



**Figure 9-30b** A horizontal, non-insulated storage tank of Example 9-5.

**Solution**

The wetted surface for the vessel equals the wetted area of the two heads plus that of the cylindrical section.

Thus:

$$A_w = \frac{2\pi(yD_1)^2}{4} + D_2L\pi$$

where

$y$  = the fraction of the vessel's internal diameter (ID) that is equivalent to average liquid inventory

$D_1$  = diameter of circular blank from which head is shaped  
( $D_1$  will depend on the type of head)

$D_2$  =  $y$  times the ID of the tank

$L$  = tangent-to-tangent length of the cylindrical section.

$$D_1 = 11 \text{ ft}$$

$$A_w = \frac{2\pi[(0.75)(11)]^2}{4} + (0.75)(10)(20)\pi \\ = 578.15 \text{ ft}^2$$

The heat input to the wetted surface of the vessel is given by Eq. (9-29),

$$Q = 21,000FA_w^{0.82}$$

where

$F$  = environmental factor,  $F = 1.0$  (Bare vessel)

$$Q = 21,000(1)(578.15)^{0.82} \\ = 3.864 \times 10^6 \text{ Btu/h}$$

The mass flow rate of vinyl chloride  $m$ , lb/h is

$$m = Q/\lambda = 3.864 \times 10^6/116 \\ = 33,314.6 \text{ lb/h}$$

The relieving temperature at the set pressure plus 20% overpressure plus 14.7 is 135°F. At this temperature, the latent heat of vaporization is 116 Btu/lb. Molecular weight  $M$  of vinyl chloride is 62.5. Ratio of specific heat capacities,  $k$ , is 1.17

The relieving pressure,

$$P = \text{set pressure} + \text{overpressure} + \text{atmospheric pressure} \\ = 100 + (100 \times 0.2) + 14.7 \\ = 134.7 \text{ psia}$$

The critical properties of vinyl chloride are

$$P_c = 809 \text{ psia} \\ T_c = 313.7^\circ \text{ F} \\ P_r = P/P_c = 134.7/809 = 0.167 \\ T_r = T/T_c = (135 + 460)/(313.7 + 460) = 0.77.$$

From the compressibility factor chart (see Figure 3-27),  $Z = 0.86$ . The value of  $C$  with  $k = 1.17$  is obtained by substituting the value of  $k$  in Eq. (9-11H),

$$C = 520 \sqrt{k \left( \frac{2}{k+1} \right)^{(k+1)/(k-1)}} \\ C = 520 \sqrt{1.17 \left( \frac{2}{1.17+1} \right)^{(1.17+1)/(1.17-1)}} \\ C = 334.17$$

For vapors and gases, in lb/h  $K_b = 1.0$ ; "C" from Figure 9-25,  $P$  is the relieving pressure absolute, psia. Substituting the values in Eq. (9-10),

$$A = \frac{W\sqrt{TZ}}{CK_dPK_b\sqrt{M}}, \text{ in.}^2 \\ A = \frac{(33314.5)\sqrt{(595)(0.86)}}{(334.17)(0.953)(134.7)(1)\sqrt{62.5}} \\ = 2.22 \text{ in.}^2$$

The nearest standard orifice area is 2.853 in.<sup>2</sup> The pressure-relief valve designation is  $L$  and the preferred valve body size is 3–4 or 4–6.

The maximum vapor rate with the nearest standard orifice area is

$$W = ACK_dPK_b\sqrt{M/(TZ)} \\ = (2.853)(334.17)(0.953)(134.7)(1)\sqrt{62.5/(595 \times 0.86)} \\ = 42,772.2 \text{ lb/h}$$

Table 9-11 shows computer results for fire relief condition using SAFETY of Example 9-5.

**TABLE 9-11**

Fire Relief Valve Sizing:	Valve Name:	Example 9-5
Environmental factor (non-insulated vessel):		1.000
Wetted surface area of vessel, ft <sup>2</sup> :		578.150
Latent heat of vaporization, Btu/lb:		116.000
Valve set pressure, psig:		100.000
Percentage of valve overpressure, %:		20.000
Upstream fluid temperature, °R:		135.000
Fluid compressibility factor:		0.860
Fluid molecular weight:		62.500
Ratio of specific heats of fluid, k=Cp/Cv:		1.170
Gas constant determined by ratio of k:		334.173
Vapor relief flow rate, lb/hr.:		33314.590
Calculated valve orifice area, in <sup>2</sup> :		2.172
Nearest standard orifice area, in <sup>2</sup> :		2.853
Nearest standard orifice size:	L	
Preferred valve body size (In-out) inches:		3-4 or 4-6
Maximum vapor flowrate with std. orifice, lb/hr.:		43760.010
Reaction force, lb:		270.9

$T_w$  = vessel wall temperature, in degrees Rankine  
 $(^{\circ}\text{R} = ^{\circ}\text{F} + 460)$ .

$T_1$  = gas temperature, absolute, in degrees Rankine  
 $(^{\circ}\text{R} = ^{\circ}\text{F} + 460)$ , at the upstream pressure, determined from the following relationship:

$$T_1 = \frac{P_1}{P_{\eta}} T_{\eta}$$

where

$P_{\eta}$  = normal operating gas pressure, in pounds per square inch absolute, psia

$T_{\eta}$  = normal operating gas temperature, in degrees Rankine  
 $(^{\circ}\text{R} = ^{\circ}\text{F} + 460)$ .

The recommended maximum vessel wall temperature for the usual carbon steel plate materials is 1100°F. Where vessels are fabricated from alloy materials, the value for  $T_w$  should be changed to a more appropriate recommended maximum [5a].

The relief load can be calculated directly, in lb/h [5a]:

$$W = 0.1406 \sqrt{MP_1} \left[ A_3 \frac{(T_w - T_1)^{1.25}}{T_1^{1.1506}} \right], \text{lb/h} \quad (9-34)$$

### 9.34 RUPTURE DISK SIZING DESIGN AND SPECIFICATION

Rupture or burst pressure of the metal disks must be specified at least 25–40% greater than the normal non-pulsing operating pressure of the vessel or system being protected. For low operating pressures less than 5 to 10 psig, the differences between operating pressure and set pressure of a valve or disk may need to be greater than that just cited.

For mild pulsations, the disk bursting pressure should be 1.75 times the operating pressure; and for strong pulsations, use two times the operating pressure [43]. Non-metallic impregnated graphite disks may be used to burst at 1.34 times operating pressure as these are less subject to fatigue. The bursting pressure must never be greater than the MAWP of the vessel, and proper allowance must be made for the possible pressure variations, plus and minus, due to the manufacturer's rupture pressure range. See Table 9-2 and specific manufacturers' literature. The ASME code [1] Par. UG-127 requires disks to burst within  $\pm 5\%$  of the stamped bursting pressure at a specified disk temperature at the time of burst.

### 9.35 SPECIFICATIONS TO MANUFACTURER

When ordering rupture disks, the following information and specifications should be given to the manufacturer.

- Net inside diameter of opening leading to the flange or holding arrangement for the disk in inches; or the cubic feet of vapor at stated conditions of burst pressure, or both.
- Preferred material of construction, if known; otherwise, state service to obtain recommendation.
- Type of hold-down arrangement: flanged (slip on, weld neck, screwed, stud) union, screwed, or special.
- Material of construction for hold-down (flange, screwed) arrangement. Usually forged carbon steel is satisfactory, although aluminum or other material may be required to match vessel and/or atmosphere surrounding the disk assembly.
- Temperature (a) for continuous operation and (b) at burst pressure.
- Required relief or burst pressure in vessel and the back pressure on the disk, if any.
- Disks to be ASME code certified.

When the flow capacity for relief can be given to the disk manufacturer, together with the conditions at bursting pressure (including temperature), the manufacturer can check against a selected size and verify the ability of the disk to relieve the required flow.

### 9.36 SIZE SELECTION

Rupture disks are used for the same purpose as safety valves and, in addition, serve to relieve internal explosions in many applications. If the pressure rise can be anticipated, then the volume change corresponding to this change can be calculated by simple gas laws, and the capacity of the disk at the relieving pressure is known. The system must be examined and the possible causes of overpressure and their respective relief capacities identified before a reliable size can be determined. See Figure 9-14a.

### 9.37 CALCULATION OF RELIEVING AREAS: RUPTURE DISKS FOR NON-EXPLOSIVE SERVICE

The vessel nozzle diameter (inside) or net free area for relief of vapors through a rupture disk for the usual process applications is calculated in the same manner as for a safety relief valve, except

that the nozzle coefficient is 0.62 for vapors and liquids. Most applications in this category are derived from predictable situations where the flow rates, pressures, and temperatures can be established with a reasonable degree of certainty.

For rupture disk sizing the downstream pressure is assumed to reach the critical flow pressure although the downstream pressure initially may be much lower. Under these conditions the flow through the "orifice" that the disk produces on rupture is considered to be at critical flow. The assumptions of critical pressure do not apply where a fixed downstream side pressure into which the disk must relieve is greater than the critical pressure.

The coefficient of discharge,  $K_d$ , is the actual flow divided by the theoretical flow and must be determined by tests for each type or style and size of rupture disk as well as pressure-relieving valve. For rupture disks, the minimum net flow area is the calculated net area after a complete burst of the disk, making allowance for any structural members that could reduce the net flow area of the disk. For sizing, the net flow area must not exceed the nominal pipe size area of the rupture disk assembly [1].

The bursting pressure,  $P_b$ , of the conventional tension-loaded disk is a function of the material of which the disk is fabricated, as well as its thickness and diameter; the temperature at which the disk is expected to burst, and not just the temperature corresponding to the disk set pressure [17]. This type of disk is best suited to be set on  $P_b$  at least 30% above the system operating pressure. The reverse-buckling disk with knives to aid the bursting is compression loaded because the dome of the disk faces the internal vessel pressure. The bursting pressure,  $P_b$ , of this disk is dependent on the dome's geometrical shape and the characteristics of the knife blades, but it is essentially independent of thickness and generally does not need a vacuum support [17]. It is often used when the operating pressure is as high as  $0.9P_b$ . There are some potential problems or even hazards with this design if the knives fail, come loose, or corrode, and the use must be examined carefully. This disk like any other should never be installed upside down from its original design position. It should not be used in partial or total liquid service.

The reverse-buckling disk, without knives but with a pre-scored disk surface, offers some features that do not depend on the knives being in place because the thickness of the metal disk dome along the score line determines the bursting pressure of the disk.

### 9.38 THE MANUFACTURING RANGE (MR)

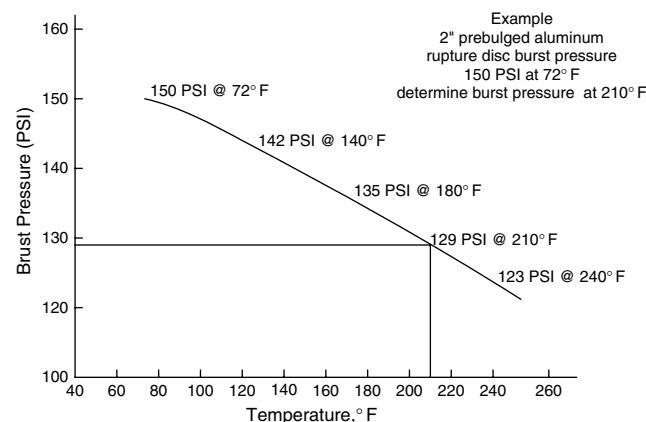
The ASME code [1] requires that a ruptured disk must be stamped with a bursting pressure that falls within the manufacturing range (MR). This range identifies the allowable range of variation from a specified burst pressure to the actual burst pressure provided by the manufacturer and as agreed upon with the disk user. The stamped burst pressure of a lot of rupture disks is the average burst pressure of all the destructive tests performed per code requirements. The average of the tests must fall within the manufacturing range (see Table 9-2).

The thickness of one material for manufacture of a disk, along with the specific disk type, is the key factor in establishing at what pressure range a disk of a specified bursting diameter will actually burst on test and then in actual service.

To specify a rupture disk:

- identify the desired bursting pressure,  $P_b$
- list the required MR.

For example, a system requiring a bursting pressure,  $P_b$ , of 150 psig at 400°F, would have an MR range of -4 to +7% at the operating temperature with a burst pressure tolerance of  $\pm 5\%$ . The disk supplied by a specific manufacturer (MR varies with



**Figure 9-31** Establishing stamped rupture disk bursting pressure at coincident temperature, Method 2. (By permission from Fike Metal Products Div., Fike Corporation.)

manufacturer and pressure ranges) could have a bursting pressure as low as  $-(0.04)(150) = -6$  psi or 144 psig; or as high as  $(+0.07)(150) = +10.5$  psi or 160.5 psig. If the disk is stamped at the operating temperature at 144 psig, it could burst at  $\pm 5\%$  or 7.2 psi or 136.8 psig or 151.2 psig. On the other hand, if the disk were stamped at operating temperature to burst at 160.5 psig (its highest) then it could actually burst  $\pm 5\%$  of this, or  $\pm 8.03$  psi, giving an actual burst pressure of 168.5 to 152.5 psig. Adapted from [17] by permission.

The code requires that the disks be burst on test by one of three methods using four sample disks, but not less than 5% from each lot. Figure 9-31 illustrates test results for burst pressure versus temperature of a disk design, all fabricated from the same material, and of the same diameter.

It is critical that such an examination be made to be certain that the bursting or set pressure at this temperature does not exceed the MAWP of the vessel at the operating temperature as per the ASME code [1] (Figures 9-32a and 9-32b).

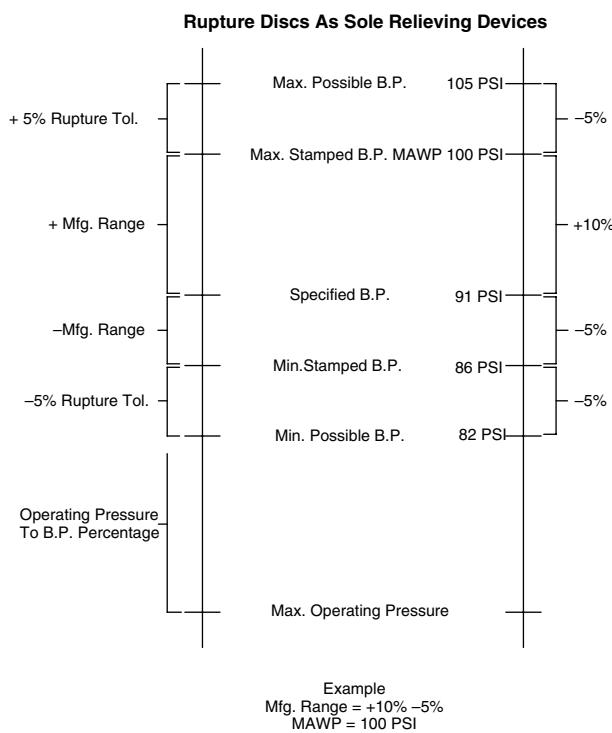
As allowed by the code [1], the average of the manufacturer's disks burst tests could be stamped, for example  $(144 + 160.5)/2 = 152.3$  psig, with an actual  $\pm 5\%$  of 152.3 psig allowed for actual burst pressure of any disk *at the operating temperature*.

### 9.39 SELECTION OF BURST PRESSURE FOR DISK, $P_b$ (TABLE 9-3)

It is essential to select the type or style of rupture disk before making the final determination of the final burst pressure, and even this selection must recognize the pressure relationships between the disk's manufacturing range and the vessel's MAWP (Table 9-3) (also see Figures 9-32a and 9-32b).

Table 9-12 summarizes the usual recommended relationship between the operating pressure of a process (should be the maximum expected upper range level) and the set pressure of the rupture disk. Recognize that the set pressure of the disk must not exceed the MAWP of the vessel (see Figures 9-32a and b). The burst pressure,  $P_b$ , can now be defined. The use of the MR discussed earlier applied to the burst pressure will establish the most probable maximum  $P_b$ .

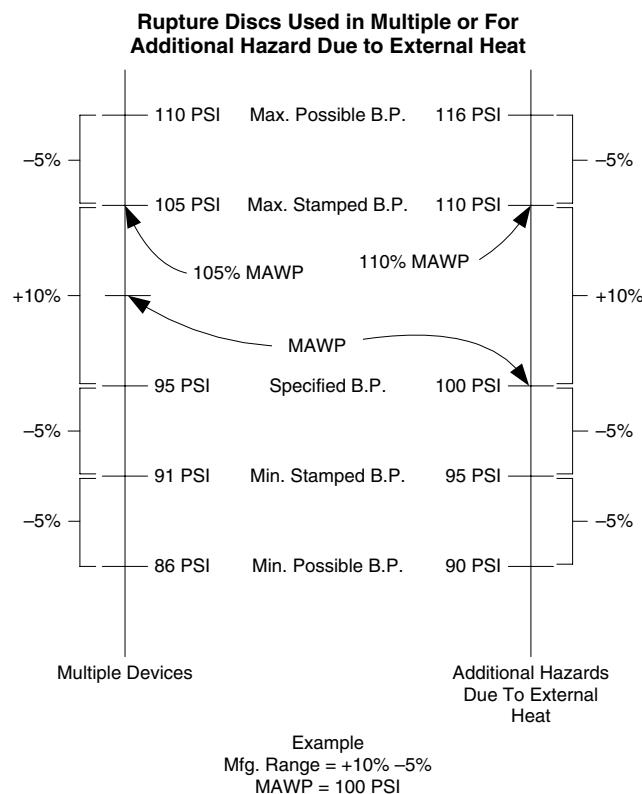
The burst pressure maximum cannot exceed the MAWP of the vessel. Depending on the situation, it may be necessary to work backward to the operating pressure maximum to see if this is



**Figure 9-32a** Rupture disks as sole relieving devices. (By permission from Fike Metal Products Div., Fike Corporation.)

usable. Table 9-12 summarizes typical rupture disk characteristics noting that the maximum normal operating pressure of the system is shown as a function of the rupture disk bursting pressure,  $P_b$ .

From this guide to determine the bursting pressure of the rupture disk, it is apparent that some thought must be given to the process and the equipment design and the ultimate MAWP. It is not an arbitrary selection. When changing services on a vessel, the MAWP and the operating pressures of the process must be established and then the bursting specifications of the disk determined. When reordering rupture disks for a specific service to repeat a performance of a ruptured disk, it is important to set the bursting specifications and the MR exactly the same as the original order, otherwise the maximum stamped  $P_b$  could be too high for the vessel and the system not properly protected against over-pressure.



**Figure 9-32b** Rupture disks used in multiple or for additional hazard due to external heat. (By permission from Fike Metal Products Div., Fike Corporation.)

#### 9.40 EFFECTS OF TEMPERATURE ON DISK

The temperature at the burst pressure must be specified to the manufacturer, as this is essential in specifying the metal or composite temperature stresses for the disk finally supplied. Higher temperatures reduce the allowable working stress of the disk materials. Reference [17] shows that temperature has an effect on metals in decreasing order, with the least effect on the lowest listed metal:

- aluminum
- stainless steel (changes after 400° F]
- nickel
- Inconel.

**TABLE 9-12 Summary of Rupture Disk Characteristics**

Type of Disk	Vacuum Support Required?	Fragment upon Rupture?	Gas Service?	Liquid or Partially Liquid Service?	Maximum Normal Operating Pressure
Conventional Pre-scored	Sometimes	*Yes/No*	Yes	Yes	$0.7P_b$
Tension-loaded	No	No	Yes	Yes	$0.85P_b$
Composite	Sometimes	Yes	Yes	Yes	$0.8P_b$
Reverse-buckling with knife blades	No	No	Yes	No	$0.9P_b$
Pre-scored Reverse-buckling	No	No	Yes	No	$0.9P_b$

By permission from Nazario, F.N., *Chem. Eng.*, Jun 20, 1988, p. 86 [17].

\* Depends on manufacturer's specifications (E.E. Ludwig) [34]

---

**EXAMPLE 9-6**  
**Rupture Disk Selection**

Examination of the temperature control ranges of a process reactor reveals that the normal controls are to maintain a pressure of the reacting mixture of 80 psig, while the upper extreme could be 105 psig, which would be defined as the normal maximum operating pressure.

Select a conventional rupture disk, then from Table 9-12  $P_{\max \text{ op}} = 0.7 P_b$ . Thus,

$$105 = 0.7 P_b$$

$$P_{b-\min} = 105/0.7 = 150 \text{ psig min. rupture disk burst pressure}$$

From manufacturing range table for this type of disk, MR = +10/-5% at 150 psig rupture pressure minimum. The maximum rupture pressure =  $150 + 10\% = 165$  psig, plus/minus the disk tolerance of 5%, allowing a final maximum burst pressure of 173.2 psig.

The minimum stamped bursting pressure of the disk would be 150 psig + 5% tolerance = 157.5 psig.

With a -5% MR, the specified burst pressure would be  $157.5 + 5\% = 165.4$  psig.

With a +10% upper MR, the maximum stamped burst pressure of the disk could be  $165.4 + 10\% = 181.9$  psig.

Using the code allowed tolerance for burst, the disk could burst at  $181.9 + 5\% = 190.9$  psig. The MAWP for the vessel cannot be less than 181.9 psig.

---

When specifying the material at the disk temperature, the heat loss at the disk/disk holder as well as in a flowing pipe must be recognized and the assembly may need to be insulated. This is important as it relates to the actual temperature at the bursting pressure. Establishing this burst temperature is an essential part of the system safety and must not be guessed at or taken lightly.

Table 9-13 presents a temperature conversion table for various metals from one manufacturer for conventional pre-bulged, tension-loaded disks with pressure on concave side (not prescored) as an illustration of the effect of lower or elevated temperatures referenced to 72°F on the burst pressure of a stamped disk. For other types of disk designs and from other manufacturers, the specific data for the style disk must be used to make the appropriate temperature correction.

#### 9.41 RUPTURE DISK ASSEMBLY PRESSURE DROP

The ruptured or burst disk on a vessel or pipe system presents a pressure drop to flow at that point, and it can be estimated by assuming the disk is a flat plate-orifice [17] with a discharge coefficient,  $K_d$ , of 0.62. As an alternative, the disk assembly can be assumed to be the equivalent of a section of pipe equal to 75 nominal disk diameters in length.

#### 9.42 GASES AND VAPORS: RUPTURE DISKS [5a, PAR, 4.8]

The sizing is based on the use of the ASME code [44] flow coefficient:

$K_d = 0.62$  [1] (Par. UG-127) for standard metal disks, but use

$K_d = 0.888$  for graphite disks [45]

$K_d$  = actual flow/theoretical flow = coefficient of discharge.

To select the proper sizing equation, determine whether the flowing conditions are sonic or subsonic from the equations. When the absolute pressure downstream or exit of the throat is less than or equal to the critical flow pressure,  $P_c$ , then the flow is critical and the designated equations apply [5a]. When the downstream pressure exceeds the critical flow pressure,  $P_c$ , then sub-critical flow will occur, and the appropriate equations should be used [5a].

When  $P_1$  is increased, the flow through an open disk increases and the pressure ratio,  $P_2/P_1$ , decreases when  $P_2$  does not change, until a value of  $P_1$  is reached, and there is no further increase in mass flow through the disk. The value of  $P_1$  becomes equal to  $P_c$ , and the ratio is the critical pressure ratio, and the flow velocity is sonic (equals the speed of sound).

The maximum velocity (sonic) of a compressible fluid in pipes is [40]

$$v_s = \sqrt{kgRT} \quad \text{or} \quad (9-35)$$

$$v_s = \sqrt{kgP'\bar{V}(144)} \quad \text{or} \quad (9-36)$$

$$v_s = 68.1\sqrt{kP'\bar{V}} \quad (9-37)$$

where

$v_s$  = sonic or critical velocity of a gas, ft/s

$k$  = ratio of specific heats,  $C_p/C_v$

$g$  = acceleration of gravity = 32.174 ft/s/s

$R$  = individual gas constant =  $MR/M = 1545/M$

$M$  = molecular weight

$MR$  = universal gas constant = 1545

$T$  = absolute temperature, °R =  $(460 + t^{\circ}\text{F})$

$t$  = temperature, °F

$P'$  = pressure, psia, at outlet end or a restricted location in pipe when pressure drop is sufficiently high

$\bar{V}$  = specific volume of fluid,  $\text{ft}^3/\text{lb}$ .

For sonic flow refer [5a].

When actual pressure ratio,  $P_2/P_1$ , is less than critical pressure ratio, flow is sonic or critical.

From Eq. (9-7), critical pressure ratio

$$\frac{P_c}{P_1} = \left[ \frac{2}{(k+1)} \right]^{k/(k-1)}$$

where

$P_c$  = critical flow throat pressure, psia

$P_b$  = stamped bursting pressure, psia = burst pressure + overpressure allowance (ASME code of 10%) plus atmospheric pressure, psia.

*Important note:* When actual system ratio,  $P_2/P_1$ , is less than critical pressure ratio, calculated above by Eq. (9-7), flow is sonic. When actual  $P_2/P_1$  ratio is greater than critical pressure ratio, flow is subsonic.

$P_2$  = back pressure or exit pressure, psia

$P_1$  = upstream relieving pressure, psia.

For sonic flow conditions refer [46].

From Eq. (9-10),

$$A = \frac{W}{CK_d P_b K_b} \left( \frac{ZT}{M} \right)^{0.5}$$

**TABLE 9-13 Temperature Conversion Table for Conventional Rupture Disks Only**

Temperature Correction Factor in % from Rupture Pressure at 72° F										
Disk Temp. (° F)	Rupture Disk Metals					Disk Temp. (° F)	Rupture Disk Metals			
	Alum	Silver	Nickel	Monel	Inconel		Nickel	Monel	Inconel	316 S.S.
-423	170	164	165	155	132	200	300	93	87	94
-320	152	152	144	140	126	181	310	92	87	94
-225	140	141	126	129	120	165	320	92	86	94
-200	136	138	122	126	118	160	330	92	86	94
-150	129	130	116	123	115	150	340	92	86	94
-130	127	126	116	121	114	145	350	91	85	93
-110	122	123	115	120	113	141	360	91	85	93
-100	120	122	115	119	112	139	370	91	85	93
-90	120	121	114	118	112	136	380	91	85	93
-80	120	120	114	118	112	136	390	90	84	93
-70	119	120	113	116	110	132	400	90	84	93
-60	119	119	112	115	110	130	410	90	84	93
-50	119	118	112	114	109	128	420	90	84	93
-40	118	117	111	113	108	125	430	89	84	93
-30	117	115	110	112	108	123	440	89	83	93
-20	116	112	109	111	107	121	450	89	83	93
-10	115	110	108	110	106	118	460	88	83	93
0	114	108	107	109	105	116	470	88	83	93
10	113	107	106	108	105	114	480	87	83	93
20	111	105	105	106	104	112	490	87	82	94
30	110	104	104	105	103	110	500	86	82	94
40	108	103	103	104	102	107	520	85	82	94
50	106	102	102	103	102	105	540	84	82	94
60	103	101	101	101	101	103	560	83	81	94
72	100	100	100	100	100	100	580	82	81	94
80	100	100	100	99	100	99	600	81	81	94
90	99	99	99	98	99	98	620	79	80	94
100	98	99	99	97	99	98	640	78	80	94
110	97	98	98	96	99	95	660	77	79	93
120	97	98	98	95	98	94	680	76	79	93
130	96	97	97	95	98	93	700	75	78	93
140	95	96	97	94	98	92	720	73	77	93
150	94	95	96	93	97	91	740	72	77	93
160	93	94	96	93	97	90	760	—	76	93
170	92	93	96	92	97	90	780	—	76	93
180	90	92	95	92	96	89	800	—	75	92
190	89	91	95	91	96	89	820	—	—	92
200	88	90	95	91	95	88	840	—	—	92
210	87	89	94	90	95	88	860	—	—	92
220	85	87	94	90	95	87	880	—	—	91
230	84	86	94	89	95	87	900	—	—	91
240	84	84	94	89	95	86				74
250	81	84	93	89	95	86				
260			93	88	94	86				
270			93	88	94	85				
280			93	88	94	85				
290			93	87	94	84				
300			93	87	94	84				

By permission from B.S. and B. Safety Systems, Inc.

**Example**

What is rupture pressure at 500° F of a nickel disk rated 300 psi at 72° F?

1. Consult temperature conversion table. Correction factor for nickel disk at 500° F is 86%.
2. Multiply disk rating at 72° F by correction factor:  $300 \times 0.86 = 258$

Rupture pressure of a nickel disk rated 300 psi at 72° F is therefore 258 psi at 500° F.

If you require a disk for a specific pressure at elevated or cold temperature and want to determine if it is a standard disk, convert the required pressure at elevated or cold temperature to pressure at 72° F.

*Note:* This conversion table does not apply to Type D or reverse buckling disks.

where

$A$  = minimum net required flow discharge area after complete burst of disk, in.<sup>2</sup>.

$C$  = sonic flow constant for gas or vapor based on ratio of specific heats,  $k$ , Figure 9-25, when  $k$  is not known use  $k = 1.001$ , or  $C = 315$

$W$  = required flow, lb/h

$M$  = molecular weight

$K_d$  = coefficient of discharge,  $K = 0.62$  for rupture disks, except some coefficients are different. For example, the Zook graphite standard ASME disks when tested mono-style, Figures 9-9b and 9-13a have a  $K_d$  of 0.888, and when inverted, Figure 9-13b have a  $K_d$  of 0.779. Consult manufacturer for special disks.

$T$  = flowing relieving temperature, °F + 460 = °R absolute

$Z$  = compressibility factor for deviation from perfect gas if known, otherwise use  $Z = 1.0$  for pressures below 250 psia, at inlet conditions.

$P_b$  = stamped bursting pressure plus overpressure allowance (ASME 10% or 3 psi whichever is greater) plus atmospheric pressure (14.7), psia.

#### Volumetric flow: scfm standard conditions (1.4.7 psia and 60° F)

$$A = \frac{Q_s(MTZ)^{1/2}}{6.326CK_dP_b}, \text{ in.}^2 \quad (9-38)$$

where  $Q_s$  = required flow, ft<sup>3</sup>/min at standard conditions of 14.7 psia and 60° F, scfm.

Actual flowing conditions, acfm:

$$A = \frac{5.596Q_A}{CK_d} \sqrt{\frac{M}{TZ}} \quad (9-39)$$

where  $Q_A$  = required flow, ft<sup>3</sup>/min at actual conditions, acfm.

**Steam: Rupture disk sonic flow; critical pressure = 0.55 and  $P_2/P_1$  is less than critical pressure ratio of 0.55.**

Dry and saturated steam pressure up to 1500 psig [5a], [46]:

$$A = \frac{W}{51.5K_dP_bK_nK_{sh}}, \text{ in.}^2 \quad (9-40)$$

where

$W$  = flow, lb/h

$K_d$  = coefficient of discharge = 0.62

$K_n$  = correction for Napier equation = 1.0, when

$$P_1 \leq 1515 \text{ psig} = (0.1906P_1 - 1000)/(0.2292P_1 - 1061)$$

where  $P_1 > 1515$  psia and  $\leq 3215$  psia, Table 9-9.

$K_{sh}$  = superheat correction factor, see Table 9-8. For saturated steam at any pressure,  $K_{sh} = 1.0$

$P_b$  = stamped bursting pressure, psia.

#### 9.43 API FOR SUBSONIC FLOW: GAS OR VAPOR (NOT STEAM)

For rupture disks, pressure ratio is greater than critical pressure; mass flow: lb/h

$$P_c/P > \text{critical pressure ratio } [2/(k+1)]^{k/(k-1)}$$

$$A = \frac{W}{735C_2K_d} \sqrt{\frac{ZT}{MP_1(P_1 - P_2)}} \quad (9-41)$$

where  $C_2$  is subsonic flow conditions based on ratio of specific heats (see Table 9-14 and equation for  $C_2$ )

Volumetric flow, scfm conditions:

$$A = \frac{Q_s}{4645.2C_2K_d} \sqrt{\frac{ZTM}{P_1(P_1 - P_2)}} \quad (9-42)$$

Actual flowing conditions, acfm:

$$A = \frac{Q_A}{131.43C_2K_d} \sqrt{\frac{P_1M}{ZT(P_1 - P_2)}} \quad (9-43)$$

Converting actual flow conditions to standard conditions of 60° F and 14.7 psia [47]:

$$Q_s = [(520/14.7)(P_{act}/T_{act})](Q_{act}) = \text{SCFM}, \\ 60^\circ \text{ F and } 14.7 \text{ psia} \quad (9-44)$$

where

$P_{act}$  = pressure actual, psia

$T_{act}$  = temperature actual, °R = °F + 460° F

$Q_{act}$  = flow at actual conditions: acfm, at actual flowing conditions

$Q_s$  = required flow, ft<sup>3</sup>/min at standard conditions (14.7 psia at 60° F), (60° F + 460 = 520° R).

#### 9.44 LIQUIDS: RUPTURE DISK

The test for critical or non-critical does not apply. These equations apply to single-phase (at inlet) liquids, non-flashing to vapor on venting, fluid viscosity is less than or equal to water [46].

ASME mass flow:

$$A = \frac{W}{2407K_d\sqrt{(P_b - P_2)\rho_1}}, \text{ in.}^2 \quad (9-45)$$

where

$W$  = flow, lb/h

$\rho$  = fluid density, lb/ft<sup>3</sup>

$K_d$  = coefficient of discharge = 0.62

$P_b$  = stamped bursting disk pressure plus accumulation of 10% plus atmospheric pressure, psia

$P_2$  = pressure on outlet side of rupture disk, psia.

$$\text{Volumetric flow : } A = \frac{W_L \sqrt{sp gr}}{38K_d\sqrt{P_b - P_2}}, \text{ in.}^2 \quad (9-46)$$

where

$sp gr$  = fluid specific gravity relative to water = 1.0 at 60° F

$W_L$  = liquid flow, gpm.

#### 9.45 SIZING FOR COMBINATION OF RUPTURE DISK AND PRESSURE-RELIEF VALVE IN SERIES COMBINATION

When the rupture disk is installed on the inlet side of the pressure-relief valve (see Figures 9-10–9-12), the ASME code requires that for untested disk–valve combinations the relieving capacity of the combination be reduced to 80% of the rated relieving capacity of the pressure-relief valve [1].

**TABLE 9-14 Constant,  $C_2$ , for Gas or Vapor for Subsonic Flow Conditions**

$P_2/P_1$	$P_2/P_1$											
	<b>k</b>	<b>0.95</b>	<b>0.90</b>	<b>0.85</b>	<b>0.80</b>	<b>0.75</b>	<b>0.70</b>	<b>0.65</b>	<b>0.60</b>	<b>0.55</b>	<b>0.50</b>	<b>0.45</b>
1.001	158.1	214.7	251.9	277.8	295.7	307.4	313.7					
1.05	158.4	215.5	253.3	280.0	298.7	311.2	318.4	321.0				
1.10	158.7	216.3	254.7	282.0	301.5	314.8	322.9	326.4				
1.15	158.9	216.9	255.9	283.9	304.1	318.2	327.1	331.4				
1.20	159.2	217.6	257.0	285.6	306.5	321.3	331.0	336.0				
1.25	159.4	218.1	258.1	287.2	308.7	324.2	334.5	340.4				
1.30	159.6	218.7	259.1	288.7	310.8	326.9	337.9	344.4	346.8			
1.40	159.9	219.2	260.0	290.1	312.7	329.4	341.1	348.2	351.3			
1.45	160.0	220.1	261.6	292.6	316.2	334.0	346.8	355.2	359.5			
1.50	160.2	220.5	262.3	293.7	317.8	336.0	349.4	358.3	363.3			
1.55	160.3	220.8	263.0	294.8	319.3	338.0	351.8	361.3	366.8			
1.60	160.4	221.2	263.7	295.8	320.7	339.8	354.1	364.2	370.2	372.3		
1.65	160.6	221.5	264.3	296.7	322.0	341.6	356.3	366.87	373.4	376.1		
1.70	160.7	221.8	264.8	297.6	323.2	343.2	358.4	369.4	376.4	379.6		
1.75	160.8	222.1	265.4	298.5	324.4	344.8	360.4	371.8	379.3	383.0		
1.80	160.9	222.4	265.9	299.3	325.5	346.2	362.3	374.1	382.1	386.3		
1.90	161.0	222.9	266.9	300.7	327.6	349.0	365.7	378.4	387.2	392.3		
2.00	161.2	233.4	267.7	302.1	329.5	351.5	368.9	382.3	391.8	397.8	400.1	
2.10	161.4	223.8	268.5	303.3	331.2	353.7	371.8	385.8	396.1	402.8	405.9	
2.20	161.5	224.2	269.2	304.4	332.8	355.8	374.4	389.1	400.1	407.5	411.4	
2.30	161.6	224.5	269.9	305.4	334.2	357.7	376.9	392.1	403.7	411.8	416.4	

$$C_2 = 735 \sqrt{\frac{k}{k-1} \left[ \left( \frac{P_2}{P_1} \right)^{\frac{2}{k}} - \left( \frac{P_2}{P_1} \right)^{\frac{k+1}{k}} \right]}$$

By permission from Continental Disk Corp., Cat. 1-1110, p. 4.

**TABLE 9-15 Rupture Disk/Relief Valve Combination Capacity Factors**

Disk Type	Disk Size	Disk Material	Set Pressure (psig)	Teledyne Ferris 2600 & 4500	Dresser 1900 1900-30 1900-35	Crosby JOS/JBS	Crosby JB	Crosby JO	Kunkle 5000 Thru 5999	Lonergan D & DB
ZAP	1.5	Monel	60–74	0.982	–	0.985	0.983	0.980	–	–
	1.5	Monel	75 Plus	0.982	–	0.985	0.990	0.986	–	–
	1.5	Nickel	30–49	–	–	0.965	–	0.975	–	–
	1.5	Nickel	50–59	0.989	0.984	0.965	–	0.975	0.988	0.966
	1.5	Nickel	60 Plus	0.985	0.984	0.992	–	0.994	0.988	0.996
ZAP (contd)	3.0	Stainless Steel	15–29	0.963	0.966	–	–	–	–	–
	3.0	Stainless Steel	30–34	0.993	0.966	0.955	–	–	–	–
	3.0	Stainless Steel	35–49	0.993	0.966	0.970	–	–	–	–
	3.0	Stainless Steel	50–59	0.993	0.966	0.970	0.976	0.981	–	–

Extracted by permission from Continental Disc Corp., Bul. #1-1111, p. 4. Only portion of original tables presented for illustration.  
Note: ZAP is a reverse-acting rupture disk using replaceable knife blades. Patented. Other disk and valve manufacturer.

For flow tested combinations, see a few typical data in Table 9-15. Note, for example, that by using a Continental disk reverse acting knife blade rupture disk with a Crosby JOS/JBS pressure-relief valve, the combined effect is to multiply the rated capacity of the Crosby valve by a multiplier of 0.985 for a set pressure in the 60–74 psig range using a 1.5-in. disk with Monel metal. Other disk and valve manufacturers have their own combination data, which when available avoids the requirement of derating the capacity to 80% of the rated capacity of the pressure-relief valve. Disks (metal or graphite) that fragment should never be used because these may become potential problems for the safety valve performance. Therefore, a non-fragmenting disk should be selected,

such as a reverse acting/reverse buckling preferably pre-scored design, but knife blades are a viable alternate.

#### EXAMPLE 9-7

##### Safety Relief Valve for Process Overpressure

The conditions set forth on the Operational Checklist, Figure 9-14a, are used in the example specified on the specification form (Figure 9-33).

Job No.				Spec. Dwg. No.
B/M No.				A.
				Page _____ of _____ Pages
				Unit Price
				No. Units
SAFETY VALVE SPECIFICATIONS				Item No.
DESCRIPTION				
Make	Model	Standard	Type-Back Pressure	Standard
Size (Inlet x Orifice No. x Outlet)	<u>4</u> x <u>L</u> x <u>6</u>		Phase	Vapor
Set Press <sup>1</sup>	<u>75</u> (psig)	@ <u>347</u> °F	Full Flow Back Press.	<u>0</u> psig
Req'd Orifice Area	<u>2.02</u>	sq. in.:	Selected Orifice Area	<u>2.853</u> sq. in.
Accessories <sup>2</sup>	Screwed Cap, No Test Gag; No Lifting Gear			
Inlet Nozzle: Press. Class	<u>250 # -4"</u>	Facing	Raised	
Outlet Nozzle: Press. Class	<u>125 # -6"</u>	Facing	Raised	
Mfgr's Rating	<u>250 #</u> psig (Max) @	<u>450</u> °F:	<u>125 # (outlet)</u>	PSIG (Min) @ <u>300</u> °F
<sup>1</sup> Refers to Initial relief pressure.				
<sup>2</sup> Refers to cap type, lifting, gag, etc.				
MATERIALS				
Body and Bonnet	Cast Iron	Trim	Bronze	
Nozzle and Disc	Bronze	Bel lows	None	
Spring	Carbon Steel, cadmium plated, or equal			
REMARKS				
Use three-way valve (Yes) (No)	<u>No</u>			
PROCESS RATING DATA				
Location	On Evaporator Shell			
Fluid	PDC	Flow Based On	Operations	Set pressure Based On Vessel Max.
System Oper. Press.	<u>60</u> psig	@	<u>347</u> °F:	<u>MW*</u> <u>113.5</u>
Req'd Cap.	<u>22,500</u> lbs/hr. @	<u>347</u> °F	Phase:	Liquid Dens.* <u>lbs/cu. ft.</u>
Liq. Viscosity*	SSU:XA overpressure <u>10 %</u>			%: Back Press. Corr. Factor
<sup>*</sup> Refers to properties at Set Pressure				
CALCULATIONS				
Use: $A = W(Z T/M)^{1/2}/C K_d P K_b$				
Since data for $c_p/c_v$ not readily available, use $k = 1.001$ and $C = 315$ , $Z = 1.0$ , $K_b = 1.0$				
$P$ (at relieving condition) = $(75)(1.1 + 14.7) = 97.2$ psia				
$T = 347 + 460 = 807R$				
$A = 22,500 [(1)(807)/113.5]^{1/2}/(315)(0.953)(1.0)$				
$A = 2.056$ sq. in.				
Note that critical flow conditions apply, since relieving pressure of 97.2 psia is over twice backpressure of 14.7 psia.				
By	Chk'd	App.	Rev.	Rev.

Figure 9-33 Safety relief valve specification for process overpressure example.

**EXAMPLE 9-8****Rupture Disk External Fire Condition**

An uninsulated 12 × 36-ft horizontal storage tank containing  $\text{CCl}_4$  is to be protected from overpressure due to external fire by means of a rupture disk. The tank does not have a sprinkler system. Storage pressure is 5 psig and should not exceed 10 psig. Tank is assumed to be full,  $k = 1.13$ . Disk burst pressure is assumed to be 10 psig. The MAWP of vessel is 50 psig. Discharge line back pressure is 1 psig.

**Solution****Heat input**

Using ASME flanged and dished heads (F&D) from Appendix Tables of Blanks, the circle size is 152 in. for a 12-ft diameter tank. Then add 3-in. straight flange which becomes 158 in. which is  $158/12 = 13.166$  ft diameter. Area of this diameter for surface area of head is  $136.14 \text{ ft}^2$  equivalent surface area of one head. For a horizontal vessel there are two heads possibly exposed to fire.

(continued)

**EXAMPLE 9-8—(continued)**

External Surface Area = cylindrical area

$$\begin{aligned} &+ \text{surface area heads (2)} \\ &= \pi(12)(36) + \left(\frac{158}{12}\right)^2 \left(\frac{\pi}{4}\right) 2 \\ &= 1357 + 272 = 1629 \text{ ft}^2 (\text{approximate}) \end{aligned}$$

**Total heat input** (from Figure 9-30a)

$$Q = 900 \times 10^6 \text{ Btu/h at area of } 1629 \text{ ft}^2$$

Solving equation  $Q = 21,000 F A^{0.82}$

$$Q = 21,000(1.0)(1629)^{0.82} = 9,036,382 \text{ Btu/h}$$

**Quantity of vapor released**

Latent heat of  $\text{CCl}_4$  = 85 Btu/lb

$$W = \frac{9,000,000}{85} = 106,000 \text{ lb/h (rounded)}$$

**Critical flow pressure**

$$P = 10 + 14.7 = 24.7 \text{ psia}$$

$$P_c = P \left[ \frac{2}{(k+1)} \right]^{k/(k-1)} = 24.7 \left[ \frac{2}{(1.13+1)} \right]^{1.13/(1.13-1)}$$

$$P_c = 24.7(0.578) = 14.3 \text{ psia, or}$$

$$\frac{P_2}{P_1} \text{ actual} = \left[ \frac{14.7 + 1.0}{(10)(1.1) + 14.7} \right] = 0.610$$

$$\frac{P_c}{P_1} = \left[ \frac{2}{(1.13+1)} \right]^{1.13/(1.13-1)} = 0.578$$

Flow is subcritical, since  $\left(\frac{P_2}{P_1}\right) > \frac{P_c}{P_1}$

**Disk area**

$$A = \frac{W}{735F_2K_d} \sqrt{\frac{ZT}{MP_b(P_b - P_2)}}$$

where

$$W = 106,000 \text{ lb/h}$$

$$K_d = 0.62$$

$$P_b = (10)(1.10) + 14.7 = 25.7 \text{ psia}$$

$$P_2 = 1.0 + 14.7 = 15.7 \text{ psia}$$

$$M = 154$$

$$T = 460 + 202 = 662^\circ \text{R (B.P. of CCl}_4 \text{ at 10 psig)}$$

$$k = 1.13$$

$$F_2 = 0.716, \text{ interpolated from Figure 9-29}$$

$$Z = 1.0, \text{ compressibility factor.}$$

Area calculated substituting in above relation:

$$\begin{aligned} A &= \frac{W}{735F_2K_d} \sqrt{\frac{ZT}{MP_b(P_b - P_2)}} \\ &= \frac{106,000}{(735)(0.716)(0.62)} \sqrt{\frac{(1.0)(662)}{(154)(25.7)(25.7 - 15.7)}} \end{aligned}$$

$$A = 42.016 \text{ in.}^2$$

Diameter is

$$\begin{aligned} d &= \sqrt{\frac{4 \text{ Area}}{\pi}}, \text{ in.} \\ &= \sqrt{\frac{(4)(42.016)}{\pi}} \\ &= 7.3 \text{ in.} \end{aligned}$$

Next standard size: Choose an 8 in. Sch. 40 pipe having a cross-sectional area of 50.0 in.<sup>2</sup> So an 8-in. frangible rupture disk will be satisfactory. Disk material to be lead or lead-covered aluminum.

#### 9.46 PRESSURE-VACUUM RELIEF FOR LOW PRESSURE STORAGE TANKS

In order to accommodate “breathing” of tanks and other equipment due to temperature changes, pumping in and out, internal vapor condensation, and other situations, adequate safety vacuum relief must be provided. In many cases, both pressure and vacuum relief are needed (Figure 9-35). For the average product storage tank the API Guide For Tank Venting RP 2000 [48] serves to set the minimum venting quantities for various tank capacities. In addition to these tabulated values, calculations are made to satisfy each condition of operation to insure that there is no situation requiring more than this vent capacity. Emergency vent capacity is also required to supplement the normal requirement in case of external fire or other unusual condition.

The normal venting to be provided must not allow pressure or vacuum conditions to develop which could cause physical damage to the tanks [48].

#### 9.47 BASIC VENTING FOR LOW PRESSURE STORAGE VESSELS

Usually, reference to low pressure venting is associated with large storage tanks of several thousand gallons capacity (ranging from a few thousand to a million); however, small low pressure tanks can be handled in the same manner. The usual operating pressure range for the typical tank is 0.5 oz/in.<sup>2</sup> to about 1.5 psig. Since low pressure vessels have pressure ratings expressed in various units, Table 9-16 can be useful for conversion.

Typical large storage vessels are illustrated in Figure 9-36. Usual operating/design pressure (max) is shown below. Operations associated with storage tanks should be carefully analyzed, since there are several factors which can significantly influence the safety relieving requirements. Usually, these are [48]

##### 1. Normal operation

- a) Outbreathing or pressure relief
- b) Inbreathing or vacuum relief

**EXAMPLE 9-9****Rupture Disk for Vapors or Gases: Non-fire Condition**

Determine the rupture disk size required to relieve the pressure in a process vessel with the following conditions:

$$k = 1.4$$

Vessel MAWP = 85 psig; also equal to disk set pressure

Vapor flow to relieve = 12,000 std.ft<sup>3</sup>/min at 60° F and 14.7 psia  
Flowing temperature = 385° F

Vapor mol. wt = 28

Back pressure on discharge of disk = 30 psig.

**Solution**

Determine if conditions on rupture are critical or non-critical, based on 10% overpressure for primary relief.

$$\frac{P_2}{P_1} = \left[ \frac{30 + 14.7}{(85)(1.10) + 14.7} \right] = 0.413$$

$$\text{Critical pressure ratio: } \frac{P_c}{P_1} = \left[ \frac{2}{(1.4 + 1)} \right]^{1.4/(1.4-1)} = 0.528$$

Since actual  $P_2/P_1 <$  critical ratio 0.528, the flow is sonic.  
Critical pressure =  $P_{cr} = 108.2(0.528) = 57.12$  psia

Flow area required to relieve [5a]:

$$A = \frac{Q_s(MTZ)^{0.5}}{6.32CK_dP_b} \quad (9-47)$$

where

$$Q_s = 12,000 \text{ scfm}$$

$$M = 28$$

$$C = 356, \text{ Figure 9-25}$$

$$K_d = 0.62 \text{ (rupture disk)}$$

$$P_b = 108.2 \text{ psia}$$

$$Z = 1.0$$

$$T = 385 + 460 = 845^\circ \text{ R.}$$

$$A = \frac{1200[(28)(845)(1.0)]^{0.5}}{6.32(356)(0.62)(108.2)}$$

$$A = 12.23 \text{ in.}^2$$

Diameter is

$$d = \sqrt{\frac{4 \text{ Area}}{\pi}}, \text{ in.}$$

$$= \sqrt{\frac{(4)(12.23)}{\pi}}$$

$$= 3.9 \text{ in.}$$

Choose next standard size: a 4 in. Sch. 40 pipe with an area of 12.6 in.<sup>2</sup> This should be adequate.

**EXAMPLE 9-10**  
**Liquids Rupture Disk**

Determine the rupture disk size required to relieve the following operating condition.

Back pressure: 0 psig

Toluene flow: 1800 gpm, sp gr = 0.90

Pressure vessel: MAWP = 25 psig

Relieving pressure: set pressure; use 25 psig

Actual relief pressure = 25 + 10% = 27.5 psig

$$A = \frac{W_L}{38K_d} \sqrt{\frac{\text{sp gr}}{(P_1 - P_2)}}$$

$$\text{sp gr} = 0.90$$

Fluid: Toluene, flow = 1800 gpm

$$A = \frac{gpm}{38K_d} \sqrt{\frac{\text{sp gr}}{(P_1 - P_2)}}$$

where

$K_d = 0.62$  per API [5a], or from manufacturer

$$P_1 = 27.5 + 14.7 = 42.2 \text{ psia}$$

$$P_2 = 0 + 14.7 = 14.7 \text{ psia}$$

The required area,

$$A = \frac{1800}{38(0.62)} \sqrt{\frac{0.90}{(42.2 - 14.7)}}$$

$$A = (76.40)(0.1809) = 13.82 \text{ in.}^2$$

Diameter is

$$d = \sqrt{\frac{4 \text{ Area}}{\pi}}, \text{ in.}$$

$$= \sqrt{\frac{(4)(13.82)}{\pi}}$$

$$= 4.19 \text{ in.}$$

Use Standard pipe size: 5 in. (cross-sectional area = 19.6 in.<sup>2</sup>) disk, or check manufacturer. Use inlet and discharge pipe size = 6 in. std.

2. Emergency conditions
  - a) Pressure venting
3. For tank design per API-Std-650 with weak roof to shell designs (roof lifts up), the venting requirements of API-Std-2000 do not apply for emergency venting to atmosphere or elsewhere.

**EXAMPLE 9-11**  
**Liquid Overpressure**

A check of possible overpressure on a heat exchanger handling 95% aqua ammonia in the tubes indicates that tube failure is probably the condition requiring maximum relieving capacity. The aqua is being pumped by a positive displacement pump. Thirty psia steam on shell side heats (not vaporizes) the aqua. In the case of tube failure, the aqua would flow into the shell and soon keep the steam from entering. The relief valve must prevent the shell from failing. The shell is designed for a working pressure of 210 psig.

The calculations are shown on the specification sheet, Figure 9-34.

**9.48 NON-REFRIGERATED ABOVE GROUND TANKS;  
API-STD-2000**

Normal operations are to be established within the design parameters for the tank, thereby avoiding conditions that would damage it. Normal venting capacity should be at least the sum of venting required for oil/fluids movement and thermal effect. Required capacity can be reduced when liquid volatility is such that vapor generation or condensation in the allowable operating range of vessel pressure will provide all or part of the venting requirements.

*Outbreathing conditions* are usually established when (a) the tank is being filled and the vapor space is being displaced with liquid, (b) thermal expansion and evaporation of the liquid, and (c) external fire on the vessel creating additional heat input to the contents.

The standard [48] specifies venting capacity of:

1. Twelve hundred cubic foot of free air per hour for every 100 bbl (4200 gal/h) of maximum filling rate, for liquids with flash points below 100° F.
2. Six hundred cubic feet of free air per hour for each 100 bbl (4200 gal/h) of maximum filling rate, for liquids with flash points of 100° F and above.
3. Thermal outbreathing or venting requirements, including thermal evaporation for a fluid (the code refers to oil) with a flash point of 100° F or below, use at least the figures of column 4 in Table 9-17.
4. Thermal outbreathing or venting requirements, including thermal evaporation for a fluid (the code uses oil) with a flash point of 100° F or above, use at least the figures of column 3 in Table 9-18.
5. To attain the total venting (outbreathing) requirements for a tank, refer to the appropriate flash point column and add the outbreathing plus the thermal venting flows.

**9.49 CORRECTIONS TO EXPRESS MISCELLANEOUS LIQUIDS VENTING IN TERMS OF FREE AIR (14.7 PSIA AND 60° F)**

Tank vent equipment ratings are expressed as free air capacity at 14.7 psia and 60° F, and in order to handle vapors from liquids

of the chemical and petrochemical industry, corrections must be made. Likewise corrections are required to recognize temperatures other than 60° F, refer to Table 9-18. The convenient constants for venting selected chemicals can be obtained from Table 9-19.

For any specific liquid vapors emergency venting,

$$V' = V_c \frac{(1,337)}{(L_v \sqrt{M})} \left( \sqrt{\frac{T}{520}} \right) \quad (9-48)$$

$V_c$  is from Table 9-20

$$\begin{aligned} V' &= \text{venting requirements, ft}^3/\text{h free air at } 14.7 \text{ psia and } 60^\circ \text{ F} \\ V' &= 1107 A_w^{0.82} \end{aligned} \quad (9-49)$$

where  $A_w$  = Wetted area,  $\text{ft}^2$ . See Table 9-20.

Correcting for effect of temperature at flowing conditions of vapor is included.

*Note:* most manufacturers' tables or charts give scfh capacities at 14.7 psia and 60° F; and these must be corrected by the gas laws to the actual volume at flowing conditions in order to represent the actual performance of the system. The tables or charts of the manufacturers read in scfh for selected relief device setting and for tank pressure, expressed as air at scfh (see Figures 9-37a and b).

The following are convenient forms to express the capacities, using Table 9-17. Note that "free" does not mean actual.

Free air capacity of valve

$$= \frac{(\text{free gas capacity})(\text{sp gr factor})}{\text{temperature correction factor}} \quad (9-50)$$

Free gas (or vapor) capacity of valve

$$= \frac{(\text{free air capacity})(\text{temperature correction factor})}{\text{sp gr factor}} \quad (9-51)$$

At the same pressure and temperature, the free gas capacity of a valve varies inversely as the square root of the specific gravity of the vapor, with the sp gr of air = 1.0.

The free air capacity of a valve varies directly as the square root of the absolute standard temperature, expressed as  $460^\circ \text{ F} + 60^\circ \text{ F}$ , divided by the square root of the valve absolute inlet temperature in ° R.

The correction factors are noted for convenience in Table 9-18. The factors are determined as follows: If the molecular weight of vapor in tank is 26.1, then the sp gr of gas =  $26.1/29 = 0.90$ , referenced to air; so the sp gr correction factor =  $(0.90)^{1/2} = 0.9486$ .

If the temperature at the valve inlet is expected to be 50° F, then the temperature correction factor

$$= \left( \frac{460+60}{460+50} \right)^{1/2} = \left( \frac{520}{510} \right)^{1/2} = (1.0196)^{1/2} = 1.00975$$

For relief valve sizing, refer to manufacturer's tables at 2 oz/in.<sup>2</sup> pressure setting and 3.5 oz/in.<sup>2</sup> maximum allowable tank pressure.

*Inbreathing* requires vacuum relief for the tank. The usual conditions arise from liquid flow out of the tank or from condensation or contraction of the vapors by reduction in temperature of the tank contents caused by atmospheric changes (not by system mechanical refrigeration).

Job No.				Spec. Dwg. No.
B/M No.				A. _____ Page _____ of _____ Pages Unit Price _____ No. Units _____
<b>SAFETY VALVE SPECIFICATIONS</b>				
<b>DESCRIPTION</b>				
Make	Model	Standard Type	Back Pressure	Standard
Size (Inlet × Orifice No. × Outlet)	×	×	Phase	Liquid
Set Press <sup>1</sup>	157 (psig)	@ 100 °F	Full Flow Back Press.	0 psig
Req'd Orifice Area	0.0951	sq. in.:	Selected Orifice Area	0.110 sq. in.
Accessories <sup>2</sup>	No lift gear, No finned bonnet, No test gag			
Inlet Nozzle: Press. Class	1 1/2", 150 #	Facing	Raised	
Outlet Nozzle: Press. Class	2", 150 #	Facing	Raised	
Mfrg's Rating	230 psig (Max) @	100 °F: 160	PSIG (Min) @	450 °F
<sup>1</sup> Refers to initial relief pressure. <sup>2</sup> Refers to cap type, lifting, gag, etc.				
<b>MATERIALS</b>				
Body and Bonnet	Cast carbon steel	Trim	Stainless steel	
Nozzle and Disc	Orged stainless steel	Belows	None	
Spring	Carbon steel			
<b>REMARKS</b>				
Use three-way valve (Yes) (No)	No			
Plant standards do not accept 1" flanged connections, therefore use 1½"				
<b>PROCESS RATING DATA</b>				
Location	Shell side of heater			
Fluid	95% Aqua Ammonia	Flow Based On	Lube failure and pump capacity	Set pressure Based On Press.
System Oper. Press.	30(shell) psig	@ 275 °F: MW*	17	
Req'd Cap.	40 gpm	lbs/hr. @ 100 °F	Phase: Liquid	Liquid Dens.* 36 lbs/cu. ft.
Liq. Viscosity*	0.05 cp	SSU:XA overpressure 25	%: Back Press. Corr. Factor	none
<sup>*</sup> Refers to properties at Set Pressure				
<b>CALCULATIONS</b>				
Max Working Pressure (shell side) = 210 psig				
* For 25% accumulation, set pressure = (0.75)(210) = 157 psig				
* Relieving pressure @ 10% overpressure = [157 + 10%] + (14.7) = 187.4 psia				
For Liquid Relief: per ASME Code				
$A = \text{gpm} (\sqrt{\rho}) / 38.0 K_d \Delta P(K_u)$				
$K_d = 0.64; K_u = 1.0$ @ normal viscosity; gpm = 40				
$\Delta P = 157 + 10\%(157) - 0 = 172.7$ psig				
$A = (40) (\sqrt{0.578}) / 38.0 (0.64) \sqrt{172.7} = 0.095 \text{ in}^2$ .				
Select 1" std. pipe size with cross-sectional area = 0.864 in <sup>2</sup> .				
By	Chk'd	App.	Rev.	Rev.

**Figure 9-34** Safety relief valve specification for liquid overpressure example.

In accordance with code [48] the inbreathing capacity is to be determined by the following.

1. Use maximum liquid flow out of tank (considered as oil by code) as equivalent to 560 ft<sup>3</sup> of free air per hour for each 100 bbl (4200 gal/h) of maximum emptying rate. This applies to oils of any flash point. Also includes gravity flow conditions.
2. For thermal inbreathing use at least the cubic feet of free air per hour given in Table 9-17, column heading number 2. This also applies to oils or fluids of any flash point.

Convert the free air rates to the proper product in the tanks using the corrections outlined in a previous paragraph. Keep in mind that the manufacturer's rating tables are in free air; however, the

actual process calculations provide flows in terms of the actual liquids at actual temperatures and pressures. It is important that the manufacturer be given the actual fluid conditions to ensure proper capacity rating.

*Emergency venting* of large tanks is usually associated with external fire conditions around the tanks. Under such conditions the venting requirements may be increased above the normal design levels previously reviewed. Emergency venting may be by [48]:

1. Weak roof-to-shell attachment for fixed roof tanks, as per API-Std-650 for *Welded Steel Tanks for Oil storage*. The joint fails and excess pressure can be relieved (above the "normal" design provided). Such tanks do not require additional emergency vent equipment; however, it can be provided in order to prevent the

roof seam failure with its attendant replacement/ maintenance requirements. This type can only be used outside of a building, not confined.

2. For fixed roof tanks without the weak roof design, the required total venting at the time of emergency is determined as below, since normal and "thermal" venting and inbreathing can be ignored. The capacity of normal outbreathing equipment can be counted toward emergency requirements.

- a. Tanks designed for 1 psig or below are to have the emergency venting rate determined from Table 9-20. For more than 2800 ft<sup>2</sup> of exposed wetted internal surface, no increase in venting is required. All the code recommendations assume that the stored liquid has the physical and thermal characteristics of hexane.
- b. Tanks designed for pressures over 1 psig (and up to 15 psig maximum as covered by API-Std-2000) are to have the emergency venting rate determined from Table 9-20; however, when the exposed surface area is above 2800 ft<sup>2</sup>, the total rate of venting is to be calculated by (Figure 9-38):

$$V_c = 1107 A_w^{0.82}$$

#### EXAMPLE 9-12 Converting Valve Capacities

The capacity of a valve as read from a manufacturer's table or chart is 45,000 ft<sup>3</sup>/h of free air (14.7 psia and 60° F). What is the capacity of the valve in terms of the vapors expected to pass through the valve under the rated conditions at the same setting if methanol is in the tank at 55° F?

$$sp\ gr = \frac{32.04(\text{MW of methanol})}{29} = 1.104$$

The *sp gr* correction factor = 1.0518 (interpolated) and the temperature correction at 55° F = 1.0048; both are from Table 9-18.

Free gas capacity of valve (at 14.7 psia and 60° F)

$$= \frac{(\text{free air capacity})(\text{temperature correction factor})}{(\text{sp gr correction factor})} \quad (9-52)$$

Free air capacity of valve

$$= \frac{(\text{free gas capacity})(\text{sp gr correction factor})}{(\text{temperature correction factor})} \quad (9-53)$$

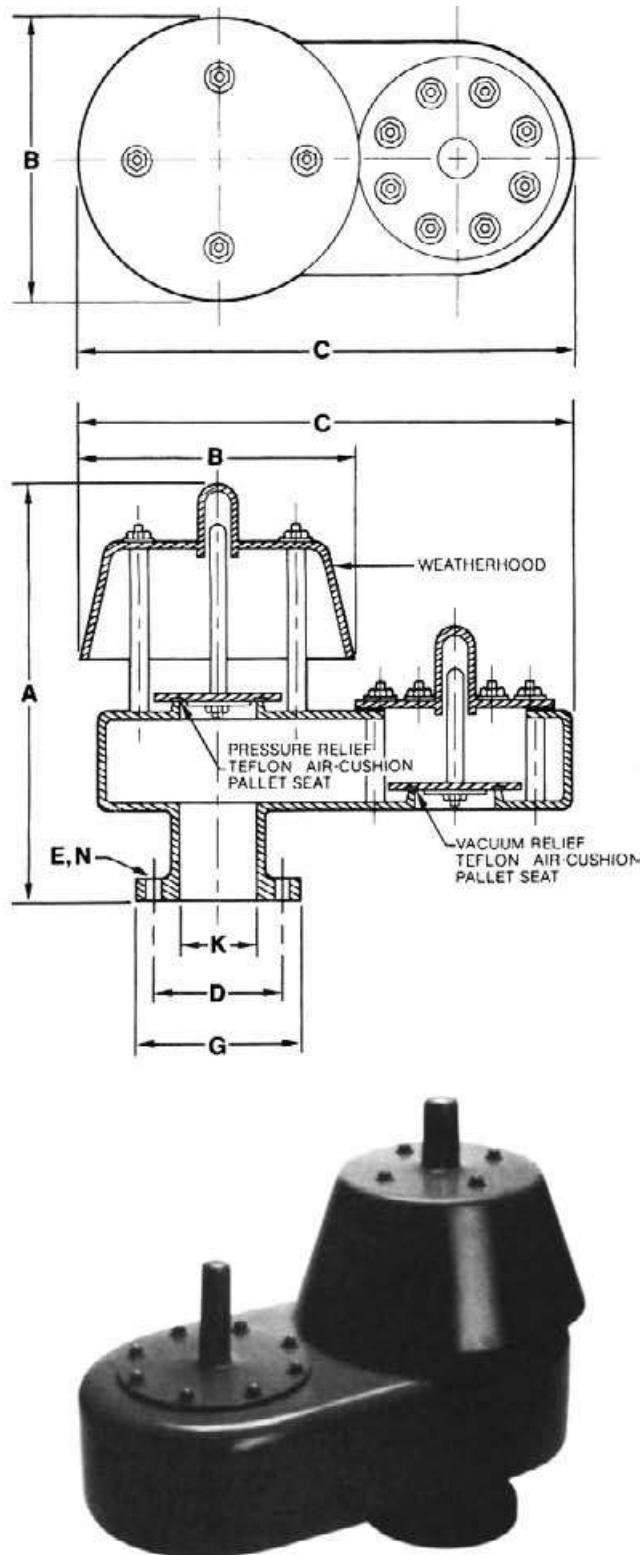
Free gas capacity of valve

$$\begin{aligned} &= \frac{(45,000)(1.0)}{1.0518} \\ &= 42,783 \text{ scfh at } 60^\circ \text{ F and } 14.7 \text{ psia.} \end{aligned}$$

If this volume were needed to be expressed at 55° F and 1.5 psig, then using the gas laws:

Actual flowing volume at 55° F and 1.5 psig

$$\begin{aligned} &= (42,783 \text{ scfh}) \frac{(460+55)}{(460+60)} \frac{(14.7)}{(14.7+1.5)} \\ &= 38,449 \text{ ft}^3/\text{h} \end{aligned}$$



**Figure 9-35** Dead weight type pressure and vacuum-relief valve for low pressure storage vessels. (By permission from The Protectoseal Co.)

**TABLE 9-16 Convenient Pressure Conversions**

Oz./in. <sup>2</sup>	Lb./in. <sup>2</sup>	In. Hg(0° C)	In. H <sub>2</sub> O (4° C)
1	0.06250	0.1272513	1.730042
16	1	2.036021	27.68068
7.85846	0.4911541	1	13.59548
0.57802	0.0361262	0.07355387	1

where

$V_c$  = venting requirement, cubic feet of free air per hour, at 14.7 psia and 60° F.

$A_w$  = wetted surface area, ft<sup>2</sup>, see Table 9-20.

Note that the above formula is based on the one in API-RP-520 [8]:

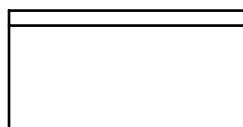
$$Q = 21,000FA^{0.82}$$

where  $Q$  = total heat input, Btu/h

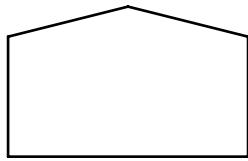
This is again based on using hexane as the reference liquid. For any stored liquid, the cubic feet of free air for equipment rating is (with greater accuracy) [48] (also see Table 9-17)

$$V'_c = V_c \frac{(1337)}{(L\sqrt{M})} \left( \sqrt{\frac{T}{520}} \right)$$

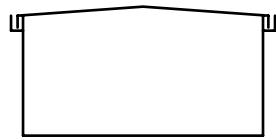
### LOW PRESSURE TYPES



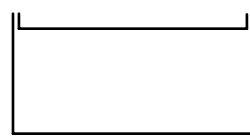
LOADED OR WEIGHTED ROOF  
8–9 oz./sq. in.



FIXED CONE ROOF  
0.5 oz./sq. in.  
Up to 3.0 oz./sq. in.

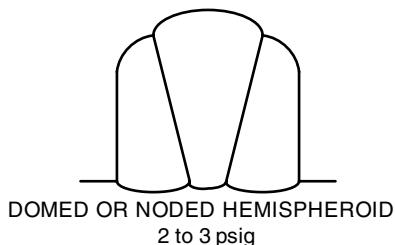


LIFTING ROOF  
3.5 to 4.5 oz./sq. in.

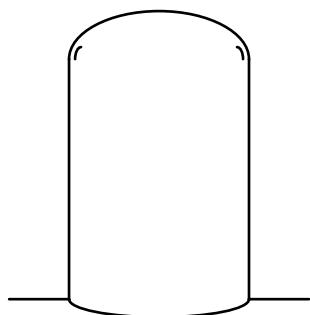


FLOATING ROOF  
1.5 to 2.25 in. water

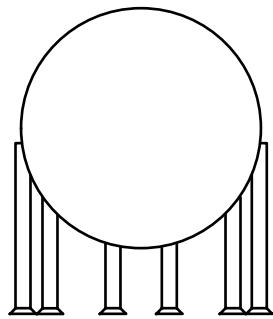
### MEDIUM PRESSURE TYPES



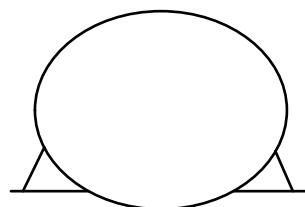
DOMED OR NODED HEMISPEROID  
2 to 3 psig



HEMISPEROID  
15 psig



SPHERE  
30–220 psig



SPHEROID  
50 psig

**Figure 9-36** Representative configurations of large storage tanks.

**TABLE 9-17 Total Rate of Emergency Venting Required for Fire Exposure Vs. Wetted Surface Area (wetted area vs. ft<sup>3</sup> of free air/h, 14.7 psia, 60°F)**

Wetted Area <sup>a</sup> (square feet)	Venting Requirement (cubic feet of free air <sup>b</sup> per hour)	Wetted Area <sup>a</sup> (square feet)	Venting Requirement (cubic feet of free air <sup>b</sup> per hour)
20	21,100	350	288,000
30	31,600	400	312,000
40	42,100	500	354,000
50	52,700	600	392,000
60	63,200	700	428,000
70	73,700	800	462,000
80	84,200	900	493,000
90	94,800	1000	524,000
100	105,000	1200	557,000
120	126,000	1400	587,000
140	147,000	1600	614,000
160	168,000	1800	639,000
180	190,000	2000	662,000
200	211,000	2400	704,000
250	239,000	2800	742,000
300	265,000	>2800 <sup>c</sup>	—

By permission from API-Std-2000, 3rd Ed., 1982, reaffirmed Dec 1987, American Petroleum Institute [48].

*Note:* Interpolation for intermediate values. The total surface area does not include the area of ground plates but does include roof areas less than 30 feet above grade.

<sup>a</sup> The wetted area of the tank or storage vessel shall be calculated as follows: For spheres and spheroids, the wetted area is equal to 55% of the total surface area or the surface area to a height of 30 feet (9.14 meters) whichever is greater. For horizontal tanks, the wetted area is equal to 75% of the total surface area. For vertical tanks, the wetted area is equal to the total surface area of the shell within a maximum height of 30 feet (9.14 meters) above grade.

<sup>b</sup> At 14.7 pounds per square inch absolute (1.014 bar) and 60°F (15.56°C).

<sup>c</sup> For wetted surfaces larger than 2800 square feet (260.1 square meters), see 1.3.2.1, 1.3.2.2., and 1.3.2.4.

where

$V_c$  = cubic feet of free air per hour from Table 9-20 or Eq. (9-49)  
for any fluid

$L$  = Latent heat of vaporization of liquid, Btu/lb

$M$  = Molecular weight of liquid

$T$  = temperature of the relief vapor, °R.

The vent size may be determined based on the pressure that the tank can safely withstand [48].

The code [48] allows a credit or reduction of required emergency venting for specific conditions. The final total emergency venting requirements may be determined as:

Condition	To Obtain Final Value*, Multiply Previous $V_c$ by
Drainage is away from tank	0.5
Tank has 1" external insulation	0.3
Tank has 2" external insulation	0.15
Tank has 4" external insulation	0.075
Tank has water spray <sup>†</sup>	1.0

\* Insulation values must be based on conductance of 4 Btu/h ft<sup>2</sup> °F/in. and insulation must be capable of withstanding fire hose streams without dislodging, and must be non-combustible.

<sup>†</sup> Water spray is not considered reliable, so no reduction is allowed.

Due to the importance of complying with all standards and regulations, and the necessary reduction in detail in such a summary as presented here, it is recommended that the ASME, API, and

National Fire Protection Agency (NFPA) standards and codes be consulted for final design detail and installation conformity.

Refer to API-2000 [48] for recommendation regarding installation requirements. Tank venting equipment capacities are expressed as free air per hour. For handling other fluids besides hexane or gasoline, use equation for  $V_c''$  (Eq. 9-48) to convert to equivalent free air flow.

## 9.50 EMERGENCY VENT EQUIPMENT

There are several types available as illustrated in Figures 9-39a, b, and c. See Figure 9-40 for specifications worksheet.

## 9.51 REFRIGERATED ABOVE GROUND AND BELOW GROUND TANKS [48]

The presence of low temperature conditions creates additional problems in design and selection of relieving devices, primarily because of the possibility of ice formation.

The capacity of pressure-vacuum devices for a tank is to be determined at 110% of their "start to discharge" pressure. Except for emergency pressure relieving because of fire, the capacity may be determined at 120% [48].

## 9.52 NORMAL CONDITIONS

At least the following conditions or some combination must be considered in establishing the normal pressure-vacuum relief capacity requirements [48].

1. loss of refrigeration
2. liquid overfilling

**TABLE 9-18 Gravity and Temperature Correction Factors for Low Pressure Venting Calculations for Vapors**

sp gr of Gas Air = 1.00	sp gr Factors	Temp. (° F)	Factor
0.20	0.447	5	1.0575
0.30	0.548	10	1.0518
0.40	0.632	15	1.0463
0.50	0.707	20	1.0408
0.60	0.775	25	1.0355
0.65	0.806	30	1.0302
0.70	0.837	35	1.0249
0.75	0.866	40	1.0198
0.80	0.894	45	1.0147
0.85	0.922	50	1.0098
0.90	0.949	55	1.0048
0.95	0.975	60	1.0000
1.00	1.000	70	0.9905
1.05	1.025	80	0.9813
1.10	1.050	90	0.9732
1.20	1.095	100	0.9638
1.30	1.141	110	0.9551
1.40	1.185	120	0.9469
1.50	1.223	130	0.9388
1.60	1.265	140	0.9309
1.70	1.305	150	0.9233
1.80	1.340	160	0.9158
1.90	1.380	170	0.9084
2.00	1.412	180	0.9014
2.50	1.581	200	0.8932
3.00	1.731	220	0.8745
3.50	1.870	240	0.8619
4.00	2.000	260	0.8498
		280	0.8383
		300	0.8272
		320	0.8165
		340	0.8063
		360	0.7963
		380	0.7868
		400	0.7776
		420	0.7687

By permission from Groth Equipment Corp., Tank Equipment Division.

### EXAMPLE 9-13 Converting Required Free Air Capacity

The capacity of a relief valve for a tank has been calculated to be 50,000 ft<sup>3</sup>/h at 110°F and 1.0 psig, with a benzene vapor. Determine the required free air capacity.

For benzene:

$$MW = 78.11$$

$$sp\ gr = \frac{78.11}{29} = 2.69$$

Temperature correction = 0.9551 from Table 9-18.

$$sp\ gr\ correction = (2.69)^{1/2} = 1.64$$

The volume at standard conditions of 14.7 psia and 60°F

$$= (50,000 \text{ SCFH}) \frac{(460+60)}{(460+110)} \frac{(15.7)}{(14.7)} = 48,717 \text{ ft}^3/\text{h}$$

Free air capacity of valve at same setting

$$= \frac{(48,717)(1.64)}{0.9551} = 83,651 \text{ schf.}$$

3. control valve failure
4. vapor displacement during filling
5. withdrawal of stored liquid at maximum rate (refer to section on non-refrigerated tanks)
6. withdrawal of vapor at maximum compressor suction rate
7. drop of barometric pressure.

Under some circumstances it may not be appropriate to allow air to enter (inbreathe) into the tank, then the use of some other inert gas, such as nitrogen or natural gas, is acceptable on a pressure control basis; however, this cannot take the place of a vacuum-relieving device to allow air to enter under a final emergency condition.

**TABLE 9-19 Convenient Constants for Venting Selected Chemicals**

Chemical	$L\sqrt{M}$	Molecular Weight	Heat of Vaporization (Btu/lb) at Boiling Point
Acetaldehyde	1673	44.05	252
Acetic acid	1350	60.05	174
Acetic anhydride	1792	102.09	177
Acetone	1708	58.08	224
Acetonitrile	2000	41.05	312
Acrylonitrile	1930	52.05	265
<i>n</i> -Amyl alcohol	2025	88.15	216
iso-Amyl alcohol	1990	88.15	212
Aniline	1795	93.12	186
Benzene	1493	78.11	169
<i>n</i> -Butyl acetate	1432	116.16	133
<i>n</i> -Butyl alcohol	2185	74.12	254
iso-Butyl alcohol	2135	74.12	248
Carbon disulfide	1310	76.13	150
Chlorobenzene	1422	112.56	134
Cyclohexane	1414	84.16	154
Cyclohexanol	1953	100.16	195
Cyclohexanone	1625	98.14	164
<i>o</i> -Dichlorobenzene	1455	147.01	120
<i>cis</i> -Dichloroethylene	1350	96.95	137
Diethyl amine	1403	73.14	164
Dimethyl acetamide	1997	87.12	214
Dimethyl amine	1676	45.08	250
Dimethyl formamide	2120	73.09	248
Dioxane (diethylene ether)	1665	88.10	177
Ethyl acetate	1477	88.10	157
Ethyl alcohol	2500	46.07	368
Ethyl chloride	1340	64.52	167
Ethylene dichloride	1363	98.97	137
Ethyl ether	1310	74.12	152
Furan	1362	68.07	165
Furfural	1962	96.08	200
Gasoline	1370–1470	96.0	140–150
<i>n</i> -Heptane	1383	100.20	138
<i>n</i> -Hexane	1337	86.17	144
Hydrogen cyanide	2290	27.03	430
Methyl alcohol	2680	32.04	474
Methyl ethyl ketone	1623	72.10	191
Methyl methacrylate	1432	100.14	143
<i>n</i> -Octane	1412	114.22	132
<i>n</i> -Pentane	1300	72.15	153
<i>n</i> -Propyl acetate	1468	102.13	145
<i>n</i> -Propyl alcohol	2295	60.09	296
iso-Propyl alcohol	2225	60.09	287
Tetrahydro furan	1428	72.10	168
Toluene	1500	92.13	156
Vinyl acetate	1532	86.09	165
<i>o</i> -Xylene	1538	106.16	149

By permission from Technical Manual, Protestoseal Co., Bensenville, Ill.

Note: For data on other chemicals, see chemistry handbook.

### 9.53 EMERGENCY VENTING FOR FIRE EXPOSURE

For refrigerated tanks the total venting requirement is the value determined from Table 9-20, multiplied by the environment factor  $F$  from Table 9-21. For tanks greater than  $2800 \text{ ft}^2$  exposed wetted surface, use the venting formula for  $V_c$  (Eq. 9-49). Do not apply the factors from Table 9-21.

### 9.54 FLAME ARRESTORS

Flame arrestors, Figure 9-41, are valuable and necessary when the danger of flammable fumes exists and a tank is venting directly to atmosphere, with no conservation vent. Here a flame arrestor should be mounted on the open vent of the tank to guard against flashback into the tank of flames, lightning ignition, and so on of the exiting fumes.

**TABLE 9-20 Total Rate of Emergency Venting Required for Fire Exposure Vs. Wetted Surface Area (wetted area vs. ft<sup>3</sup> of free air/h, 14.7 psia, 60° F)**

Wetted Area <sup>a</sup> (Square Feet)	Venting Requirement (Cubic Feet of Free Air <sup>b</sup> per Hour)
20	21,100
30	31,600
40	42,100
50	52,700
60	63,200
70	73,700
80	84,200
90	94,800
100	105,000
120	126,000
140	147,000
160	168,000
180	190,000
200	211,000
250	239,000
300	265,000
350	288,000
400	312,000
500	354,000
600	392,000
700	428,000
800	462,000
900	493,000
1000	524,000
1200	557,000
1400	587,000
1600	614,000
1800	639,000
2000	662,000
2400	704,000
2800	742,000
> 2800 <sup>c</sup>	—

By permission from API-Std-2000, 3rd Ed., 1982, reaffirmed Dec 1987, American Petroleum Institute [48].

Note: Interpolation for intermediate values. The total surface area does not include the area of ground plates but does include roof areas less than 30 ft above grade.

<sup>a</sup> The wetted area of the tank or storage vessel shall be calculated as follows: For spheres and spheroids, the wetted area is equal to 55% of the total surface area or the surface area to a height of 30 ft (9.14 m) whichever is greater. For horizontal tanks, the wetted area is equal to 75% of the total surface area. For vertical tanks, the wetted area is equal to the total surface area of the shell within a maximum height of 30 ft (9.14 m) above grade.

<sup>b</sup> At 14.7 psia (1.014 bar) and 60° F (15.56° C).

<sup>c</sup> For wetted surfaces larger than 2800 ft<sup>2</sup> (260.1 m<sup>2</sup>), see 1.3.2.1, 1.3.2.2., and 1.3.2.4.

Some conservation vents (pressure–vacuum) have built-in flame arrestors on a single compact unit mounted on the tank vent. Refer to API Safety Data 2210, Flame Arrestors for Tank Vents.

A special study [49] was commissioned by the API entitled “Mitigation of Explosion Hazards of Marine Vapor Control Systems.” The report examines the effects of deflagrations and detonations in pipes in the region of detonation flame arrestors. The primary objective was “to resolve potential operational hazards that relate to the use of detonation flame arrestors in marine vapor control systems.” The work examined deflagrations, deflagrations transitioning to detonations, effects of overpressure (explosion

pressure versus the initial pressure at ignition), stable detonations (with relatively stable speed, peak and side-on overpressure), and overdriven detonations (unstable, occur during transition from deflagration to detonation). In propane, typical stable detonation, speed is 1810 m/s (5950 ft/s), and a typical peak side-on overpressure is about 18.3 times the initial pressure. The side-on overpressure is measured in the side wall of straight pipe flush with the pipe wall. Reflected side-on pressure is measured perpendicular, that is, facing the oncoming flow.

## 9.55 PILOT-OPERATED VENT VALUES

The code [48] allows pilot-operated vent control valves provided the vent can operate automatically in case the pilot valves/system failed.

## 9.56 EXPLOSIONS

The three basic types of explosions to be concerned about in the chemical and petrochemical environment are combustion explosions (deflagrations), detonation explosions, and boiling liquid expanding vapor explosions (BLEVEs) [50].

Other than reactive metals explosions, which do not truly fall in the types noted above, the three main categories of explosions are flammable gases, liquids/vapors, and dusts. Because their sources are different, they cannot be treated in the same manner for discussion.

A **confined explosion** occurs in a contained vessel, building piping network, or other confined situation. A confined explosion has different characteristics than an unconfined explosion [51]. These explosions may be deflagrations or detonations with the detonation being much more destructive due to the higher and more rapidly moving pressure wave generated. Schwab expressly states that a vessel containing flammable vapor as a mixture when ignited with the resultant pressure buildup will explode. If the vessel does not rupture, but contains the deflagration or detonation, there is no explosion because the requirement for mechanical work has not been met.

**Combustion explosions** are explosions resulting from the uncontrolled rapid mixing and reaction of a flammable vapor from a flammable liquid with air (or oxygen) ignited from an ignition source such as flame, heat, electric spark, or static discharge. The combustion is extremely rapid with a flame propagation rate of about 7 ft/s, with the evolution of heat, light, and an increase in pressure [50]. The violence of the explosion depends on the rate at which the energy is released.

A **deflagration** is a slow burning exothermic reaction similar to the combustion explosion, but which propagates from the burning gases into the unreacted material at a velocity that is less than the speed of sound in the unreacted material. Most (not all) explosions are deflagrations.

**Detonation explosions** are similar to combustion explosions and are exothermic reactions that proceed into the unreacted material at a velocity much greater than the speed of sound in an unreacted material. These are accompanied by a flame front shock wave in the material followed closely by a combustion wave that releases the energy and sustains the shock wave at extremely high pressure [51, 52]. In hydrocarbons, the velocity can reach 6000–9000 ft/s.

A detonation generates much greater pressure and is therefore more destructive than a deflagration. For example, in a closed vessel, a hydrocarbon mixture detonation with air may generate 2.5 times (294 psi) a similar mixture in a deflagration. A deflagration may turn into a detonation, especially if traveling down a long pipe.

**EXAMPLE 9-14****Storing Benzene in Cone Roof Tank**

A 26,000-gal outdoor vertical storage tank contains benzene. The tank is not insulated, but has a dike around it with a volume equal to one-and-a-half times the volume of the tank. Size-15 ft diameter  $\times$  20 ft. Tank does not have a weak roof. Temperature = 60°F, pressure = 14.7 psia.

Data:

Flash point of benzene: < 100°F

Latent heat of vaporization: 169 Btu/lb

Tank size (nominal): 15 ft diameter and 20 ft high

Operating pressure: 1.5 oz/in.<sup>2</sup>

Maximum allowable tank pressure: 3.5 oz/in.<sup>2</sup>

Maximum allowable vacuum: 1.0 oz/in.<sup>2</sup>

Relief valve settings:

(1) For pressure: 2 oz/in.<sup>2</sup>

(2) For vacuum: 0.5 oz/in.<sup>2</sup>

Filling rate: 350 gpm (max)

Draw-out rate (emptying): 250 gpm

**A. Normal venting requirements****1. Pressure: filling**

$$\text{Rate} = (350 \text{ gpm})(60) = 21,000 \text{ gal/h}$$

Code requires 1200 ft<sup>3</sup> of free air per hour for each 4200 gal/h maximum filling rate

$$\text{scfh free air required} = \frac{(1200)}{(4200)} (21,000) = 6000$$

**2. Thermal outbreathing**

Using Table 9-17, for 26,000 gal capacity, required vent = 371 scfh free air at 14.7 psia and 60°F.

**3. Total pressure relief requirement**

$$= 6000 + 371 = 6371 \text{ scfh free air}$$

**4. Total corrected pressure relief requirement**

$$V'_c = \frac{V_c(1337)}{LM^{1/2}} = \frac{6371(1337)}{(169)(78.11)^{1/2}}$$

= 5703 scfh free air at 14.7 psia and 60°F when tank contains benzene.

Read manufacturer's capacity tables or charts to select model and size of pressure relief, reading table to closest capacity, but toward larger size if in between. Consult manufacturer for final selection.

**B. Vacuum (Inbreathing)**

1. The draw-out on emptying is given as 250 gpm. Using the code requirements, free air:

$$[(560 \text{ ft}^3 \text{ air}/4200 \text{ gal/h})][(250)(60)] = 2000 \text{ ft}^3 \text{ free air/h}$$

at 14.7 psia and 60°F

**2. Thermal inbreathing:**

From code table (using vacuum), Table 9-17.

For tanks of 26,000-gal capacity:

At 21,000 gal, scfh air = 500

At 42,000 gal, scfh air = 1000

$$\text{Volume} = 500 + \frac{(26,000 - 21,000)}{(42,000 - 21,000)} (1000 - 500)$$

= 619 scfh free air at 14.7 psia and 60°F.

Total vacuum relief required

$$= 2000 + 619 = 2619 \text{ scfh free air.}$$

Select vacuum-relief valve by referring to manufacturer's tables at vacuum relief setting of 0.5 and 1.0 oz/in.<sup>2</sup> maximum allowable vacuum for tank. Note, no correction is required as is shown for section A above, since air is flowing in, and not benzene.

**C. Emergency venting requirements (bare tank)**

Total external surface exposed to fire =

$$= \pi D(\text{height}) = \pi(15)(20) = 942.5 \text{ ft}^2$$

Use entire shell area, since height < 30 ft.

Wetted area: Entire 20 ft above grade. From Table 9-20 at 942.5 ft<sup>2</sup>, scfh free air relief:

$$900 \text{ ft}^2 = 493,000 \text{ scfh}$$

$$1000 \text{ ft}^2 = 524,000 \text{ scfh}$$

For 942.5 ft<sup>2</sup>:

$$\text{Volume} = 493,000 + \frac{(942.5 - 900)}{(1000 - 900)} (542,000 - 493,000)$$

Venting Rate = 513,825-scfh free air at 14.7 psia and 60°F

Corrected venting rate for benzene in tank

$$V'_c = \frac{V_c(1337)}{LM^{1/2}} = \frac{513,825(1337)}{(169)(78.11)^{1/2}}$$

$$= 459,945 \text{ scfh.}$$

A shock wave is generated by the expansion of gases created by the reaction of one flammable gas/liquid with an oxidant, such as air or pure oxygen, or an oxidizing material, or by pure thermal effects [29, 41, 50 and 51]. Table 9-22 and Figure 9-42a illustrate the velocities of travel of a detonating shock wave. Similar data can be or has been developed for many other industrial compounds (see, for example, [4, 6, 15, 16, 29, and 53]. Figure 9-42b compares selected flammability and detonability limits in air. The lean end/rich end of the flammability data does not support a detonation. Note

the extreme differences in shock wave propagation velocities in Table 9-22.

**9.57 FLAMMABILITY**

There are several characteristics of physical materials that describe or define the extent or even the possibility of a material being flammable or whether it will support combustion.

**EXAMPLE 9-15****Venting and Breathing in Oil Storage Tank**

A 675 bbl tank (15.5 ft diameter  $\times$  20 ft tall) is used for oil storage. The rate for pumping oil in or out is 978 gpm (maximum) or 1400 bbl/h. Flash point of oil is above 100° F. Using API Guide for Tank Venting [48]

For pressure or normal outbreathing:

Required movement =  $600 \text{ ft}^3/\text{h}/100 \text{ bbl}$  of filling rate for oils with flash point above 100° F.

Required capacity =  $(600)(1400/100) = 8400 \text{ ft}^3/\text{h}$   
Plus thermal outbreathing =  $0.60(675 \text{ bbl}) = 405$   
Total =  $8805 \text{ ft}^3/\text{h}$

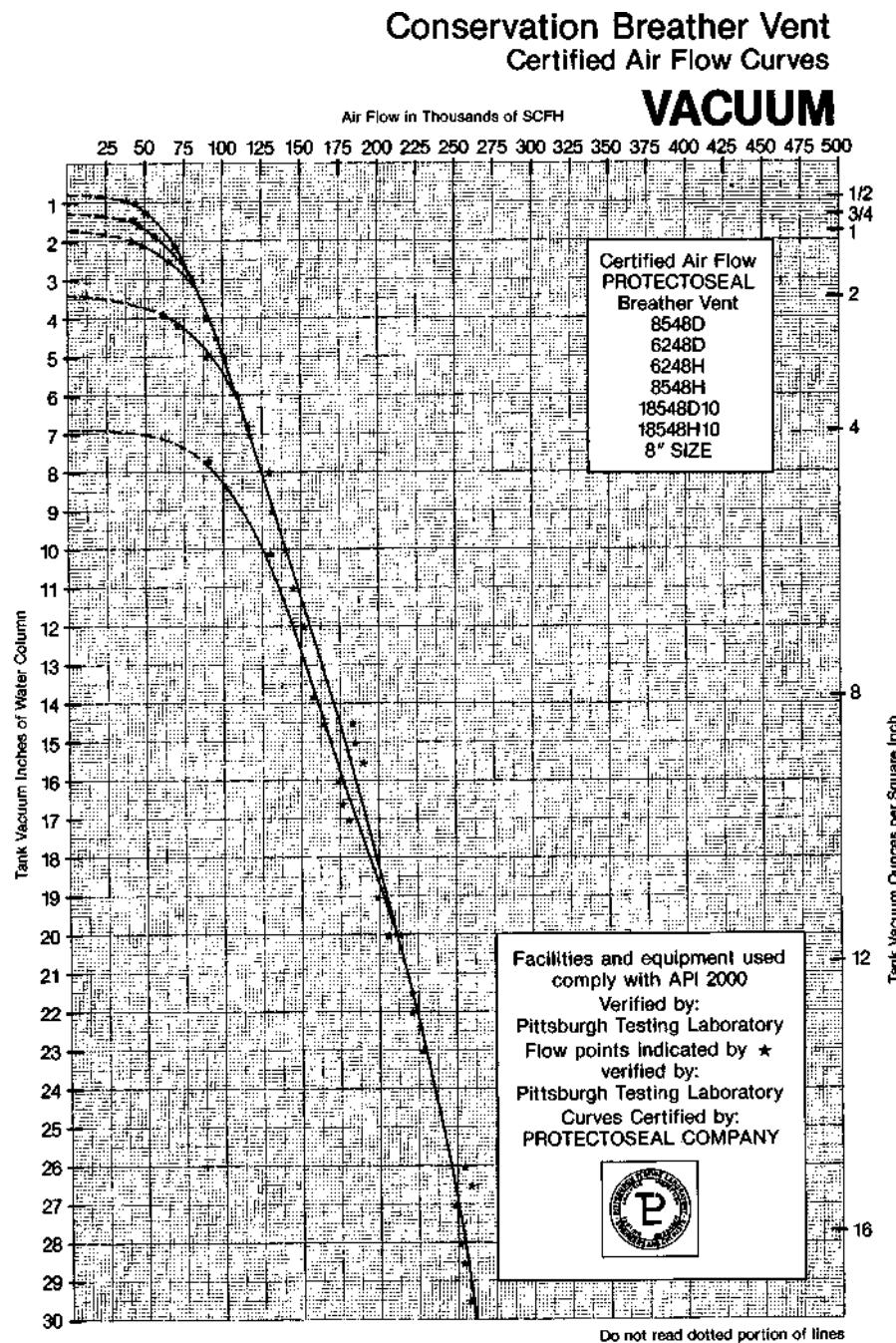


Figure 9-37a Performance curve for conservation breather vent, dead weight type for vacuum, 8 in. size. (By permission from The Protectoseal Co.)

## Conservation Breather Vent Certified Air Flow Curves **PRESSURE**

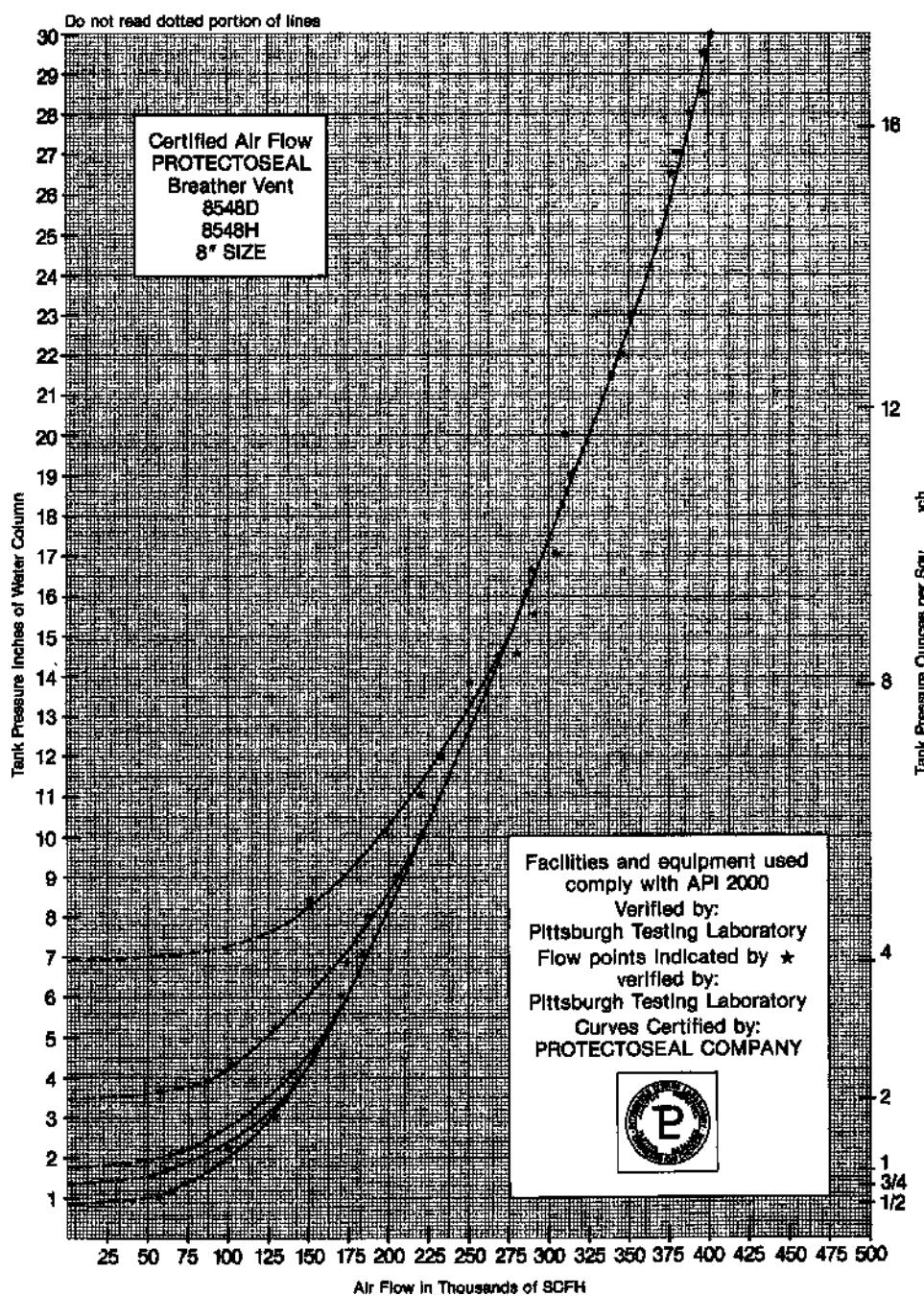
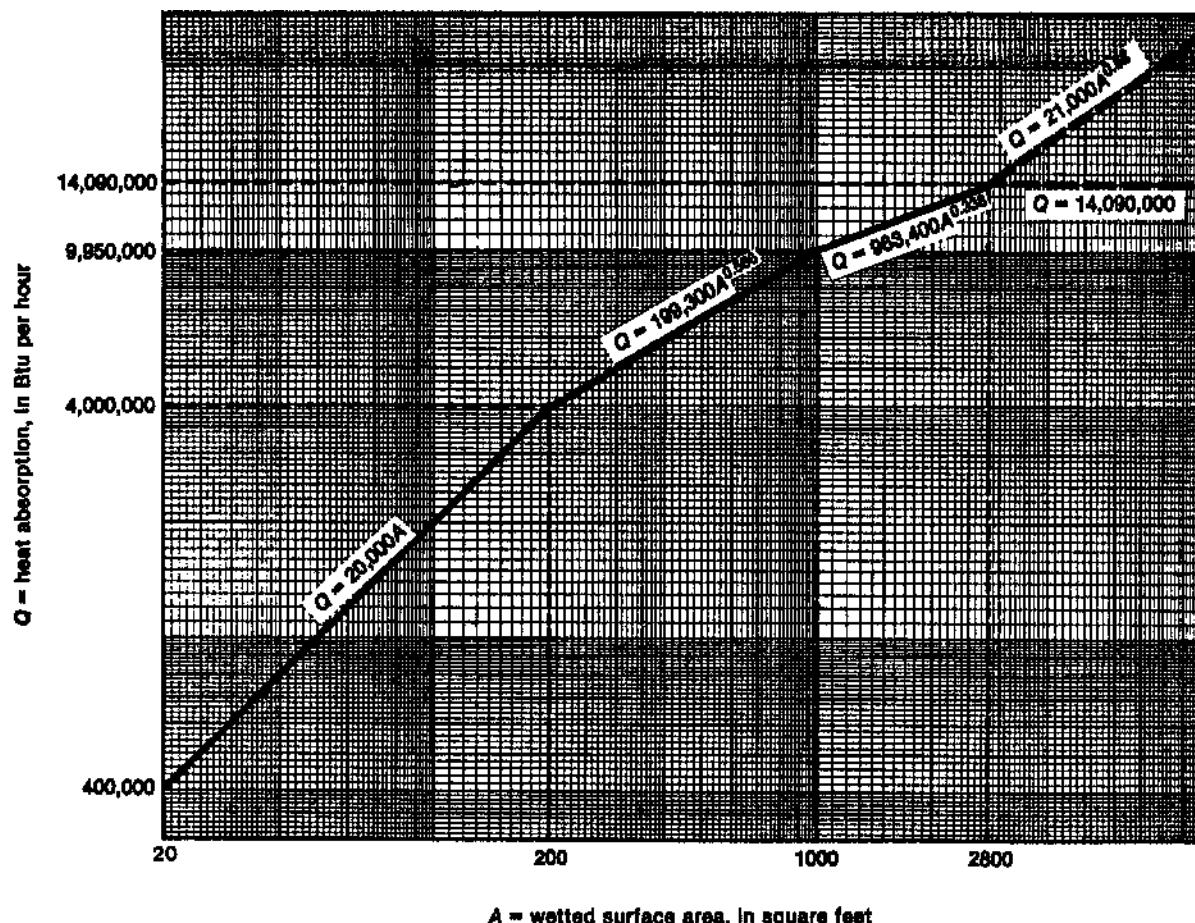


Figure 9-37b Performance curve for conservative vent, dead weight type for pressure, 8 in. size. (By permission from the Protectoseal Co.)

There are probably two known organized examination and evaluation guides and/or procedures for potential fires and explosions. These are (1) Dow's Fire and Explosion Index, Hazard Classification Guide [54] (this is too involved and extensive to present

here, but every serious safety design and researcher is urged to study this guide), and (2) DIERS, AIChE [13, 22]). This industry-supported program is targeted to understanding the complexities of venting relief devices, particularly from runaway reactions.



**NOTE:** Above 2800 square feet of wetted surface area, the total heat absorption is considered to remain constant for non-refrigerated tanks below 1 pound per square inch gage. For nonrefrigerated tanks above 1 pound per square inch gage and for all refrigerated tanks, the total heat absorption continues to increase with wetted surface area. This is the reason why the curve splits above 2800 square feet.

Figure 9-38 Curve for determining requirements for emergency venting during fire exposure. (Reprinted by permission from The American Petroleum Institute, API Std. 2000, 3rd ed., 1987, Venting Atmospheric and Low Pressure Storage Tanks.)

## 9.58 TERMINOLOGY

**Flash point of a flammable liquid:** The lowest temperature at which the liquid gives off enough vapors to form a flammable mixture with air (or pure oxygen, a special case) at or near the surface of the liquid or within its confined container. Some hazardous liquids have flash points at or below ordinary room temperatures and normally are covered by a layer of flammable vapors that will ignite immediately if a source of ignition is brought in contact [4]. Flash points are measured by "open cup" and "closed cup" methods. The open-cup data is applicable to liquid in open containers and in open pools and usually to somewhat higher temperatures than the closed-cup data. Refer to American Society for Testing Materials (ASTM) E-502, D-56, D-92, D-93, D-1310, D-3278.

The flammable liquid itself does not burn; only the vapors emitted from the liquid burn. The vaporization of a liquid depends on its temperature and corresponding vapor pressure and increases as the temperature of the liquid increases. Thus, the warmer the liquid, the more potentially hazardous it becomes.

**Fire point:** The lowest temperature at which a liquid in an open container will give off enough vapors to continue to burn

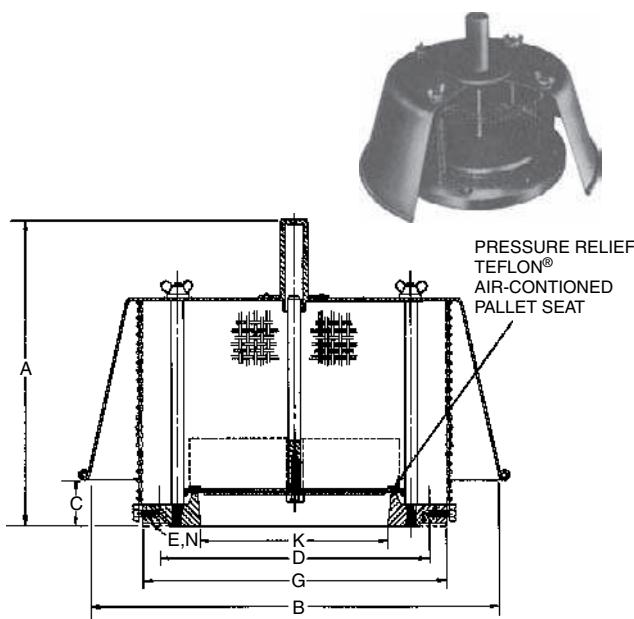
when once ignited [4]. This temperature is generally somewhat above the open-cup flash point.

**Ignition temperature:** The minimum temperature to which a material must be heated for it to ignite. Once an ignition has occurred it will continue to burn until all the available fuel or oxidant has been consumed or until the flame is extinguished by cooling or by some other means [6].

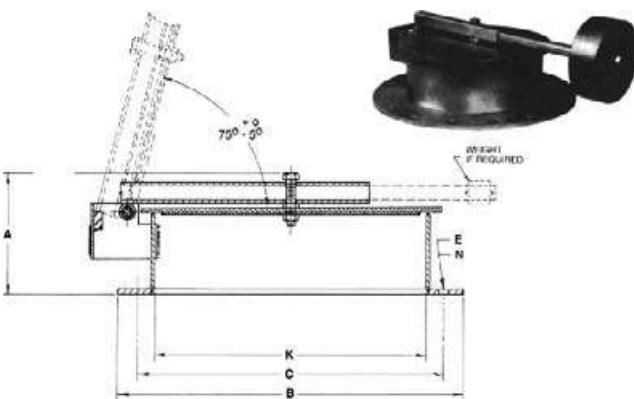
**Autoignition temperature (AIT):** The lowest temperature of a material required to initiate or cause self-sustained combustion in the absence of a spark or flame. This temperature can vary, depending on the substance, its size, and the shape of the igniting surface or container and other factors [4].

**Spontaneous heating:** Some flammable liquids combine readily with oxygen in the air at ordinary temperatures and give off heat. When the heat is generated faster than it can be dissipated, the temperature rises and ignition of the mixture may occur, as with liquids on waste or rubbish or other materials [4].

**Flammability limits or explosive range:** The entire range of concentrations of a mixture of flammable vapor or gas in air (expressed as volume percent) over which a flash will occur or a flame will travel if the mixture is ignited. Gases and vapors in air have both deflagration and detonation limits, which are often

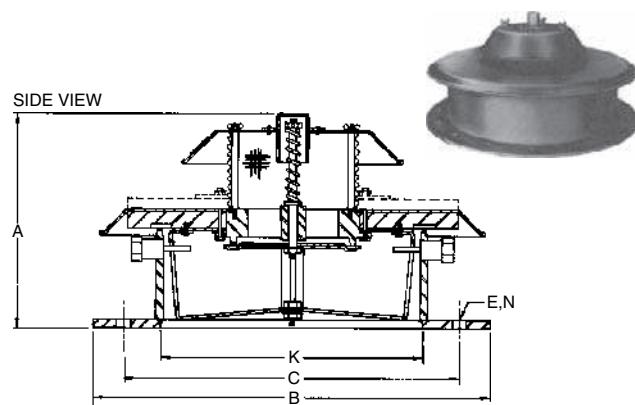


**Figure 9-39a** Emergency pressure-relief valve vent using guided weight. (By permission from The Protectoseal Co.)



**Figure 9-39b** Hinged emergency pressure manhole cover vent. (By permission from The Protectoseal Co.)

the same as flammability limits. The limits of detonability can be different and are dependent on the system conditions. Under some circumstances or some mixtures, the deflagration pressure developed by the shock waves can be 8 times the system pressure from stoichiometric fuel-air mixtures. For fuel-oxygen mixtures, the pressure increases may be as much as 20 times. The “side-on” pressures (peak pressures) from the blast (shock) wave rise almost instantaneously on surfaces orientated parallel to the direction of the wave (Figure 9-43, Baker [55], and Figure 9-44). For gas detonations, the pressures are about 2 times those for deflagrations. The reflected pressure can be another factor of 2 or greater. Therefore, for a detonating fuel-oxygen mixture, the pressure rise may be a 40-fold increase [6]. There will be no flame (except possibly a “cold flame,” not examined here) or explosion when the concentrations are outside these flammability limits [4]. (See Sample listings of flammability data Tables 9-23 and 9-24. Also see in Figures 9-45–9-47). The flammability limits are designated.



**Figure 9-39c** Combination pressure/vacuum manhole cover vent. (By permission from The Protectoseal Co.)

- 1. Lower flammability (explosive) limits (LEL or LFL):** The lowest percentage concentration at which a flash or flame can develop and propagate from the source of ignition when in contact with a source of ignition in a combustible material.
- 2. Upper flammability (explosive) limits (UEL or UFL):** The highest percentage concentration at which a flash or flame can develop and propagate flame away from the source of ignition when in contact with a source of ignition in a combustible material. See Tables 9-23 and 9-24 [6] for common flammable compounds.

Figure 9-46 illustrates a typical relationship of limits of flammability and ignitability for a methane-air mixture. Note that energy required to ignite a flammable mixture (within its LEL and UEL) varies with the composition, and that a 0.2 millijoule ( $m_j$ ) spark is inadequate to ignite even a stoichiometric mixture at atmospheric pressure at 26°C, while 1- $m_j$  spark can ignite any mixture between 6 and 11.5 vol % methane. These limits are known as limits of ignitability, indicating the igniting ability of the energy source. Limit mixtures that are essentially independent of the ignition source strength and give a measure of the flame to propagate away from the ignition source are defined as limits of flammability [53]. Considerably more spark energy is required to establish limits of flammability, and more energy is usually required to establish the upper rather than the lower limit.

## 9.59 MIXTURES OF FLAMMABLE GASES

### COMPOSITE FLAMMABILITY MIXTURES

Le Chatelier’s Rule allows the calculation of the lower flammability (explosibility) limits for flammable mixtures:

$$\text{Mixture composite LEL} = \frac{100 \text{ volume/volume}}{v_1/L_1 + v_2/L_2 + v_3/L_3 + \dots} \quad (9-54)$$

where

$v_1, v_2, \dots$  = volume percent of each combustible gas present in the mixture, free from air and inert gas

$$v_1 + v_2 + v_3 = 100 \quad (9-55)$$

$L_1, L_2, L_3, \dots$  = lower flammability limits, vol% for each flammable gas in the mixture.

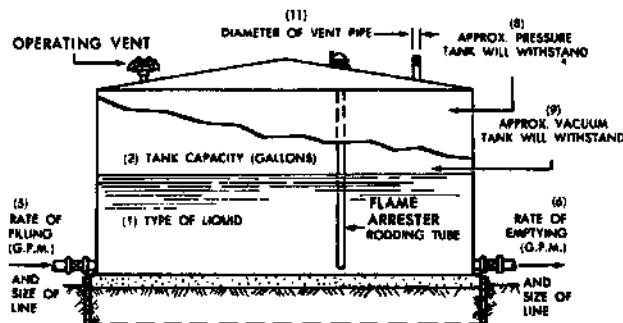
### OPERATING VENTS

**FILL IN THIS INFORMATION FOR NEW VENT INSTALLATIONS:**

1. Type of Liquid \_\_\_\_\_
- Specific Gravity \_\_\_\_\_ Flash Point \_\_\_\_\_ °F.
2. Tank Capacity in Gallons \_\_\_\_\_
3. Tank is Above Ground  or Below Ground
4. Tank is Vertical  or Horizontal  Tank Dia. \_\_\_\_\_ ft.
5. Rate of Filling (G.P.M.) \_\_\_\_\_ Size Line \_\_\_\_\_
6. Rate of Emptying (G.P.M.) \_\_\_\_\_ Size Line \_\_\_\_\_
7. Approximate operating pressure of tank \_\_\_\_\_
8. Approximate pressure tank will withstand \_\_\_\_\_
9. Approximate vacuum tank will withstand \_\_\_\_\_
10. Is tank inerted? \_\_\_\_\_ If yes, approximate blanketing pressure \_\_\_\_\_

**NOTES:**

(Specify any temperature conditions above or below ambient and/or special materials of construction required)



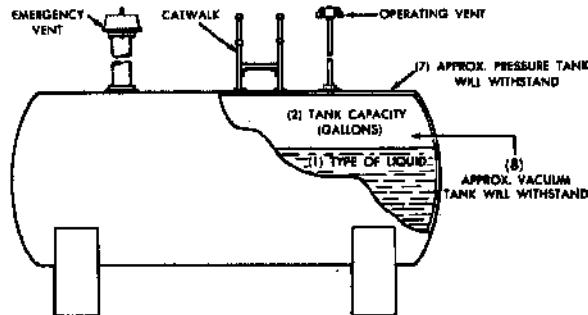
### EMERGENCY VENTS

**FILL IN THIS INFORMATION FOR NEW VENT INSTALLATION:**

1. Type of Liquid \_\_\_\_\_
- Specific Gravity \_\_\_\_\_ Flash Point \_\_\_\_\_ °F
- Molecular Weight \_\_\_\_\_ Latent Heat of Vaporization \_\_\_\_\_ BTU/lb.
2. Tank Capacity in: Gallons \_\_\_\_\_ or Barrels \_\_\_\_\_
3. Tank is Above Ground  or Below Ground
4. Tank is Vertical  or Horizontal
- Tank Diameter \_\_\_\_\_ Height \_\_\_\_\_ or Length \_\_\_\_\_
5. Approximate operating pressure of tank \_\_\_\_\_
6. Valve setting on Operating vent \_\_\_\_\_
7. Approximate pressure tank will withstand \_\_\_\_\_
8. Approximate vacuum tank will withstand \_\_\_\_\_
9. Is tank inerted? \_\_\_\_\_ If yes, what is approximate blanketing pressure? \_\_\_\_\_

**MODIFICATIONS (check, where applicable):**

	YES*	NO*
10. Tank drains to a remote area	<input type="checkbox"/>	<input type="checkbox"/>
11. Tank equipped with water spray	<input type="checkbox"/>	<input type="checkbox"/>
12. Tank is insulated	<input type="checkbox"/>	<input type="checkbox"/>



**NOTES:**

(Specify any temperature conditions above or below ambient and/or special materials of construction required)

Figure 9-40 Storage and process tank specifications work sheet. (By permission from The Protectoseal Co.)

**EXAMPLE 9-16**  
**Calculation of LEL for Flammable Mixture**

Assume mixture analysis (combustible with air)

Methane, 3.0%, LEL = 5.3%

Propane, 4.0%, LEL = 2.3%

Hexane, 1.0%, LEL = 1.1%

Subtotal 8.0% combustible

Balance is air (92%).

For each component:  $v$ , combustible only on air-free basis.

$$\text{Methane: } 3/8(100) = 37.5\%$$

$$\text{Propane: } 4/8(100) = 50.0\%$$

$$\text{Hexane: } 1/8(100) = 12.5\%$$

$$\text{Percentage combustible only} = 100.0\%$$

$$\text{Mixture composite LEL} = \frac{100}{37.5/5.3 + 50/2.3 + 12.5/1.1} \\ = 2.48\% \text{ volume of the mixture.}$$

**TABLE 9-21 Environmental Factors for Refrigerated Tanks**

Datum	F Factor
Bare metal vessel	1.0
Insulation thickness <sup>a</sup>	
6 in. (152 mm)	0.05 <sup>b</sup>
8 in. (203 mm)	0.037 <sup>b</sup>
10 in. (254 mm)	0.03 <sup>b</sup>
12 in. (305 mm) or more <sup>c</sup>	0.025 <sup>b</sup>
Concrete thickness	<sup>d</sup>
Water-application facilities <sup>e</sup>	1.0
Depressurizing and emptying facilities <sup>f</sup>	1.0
Underground storage	0
Earth-covered storage above grade	0.03

By permission from API-Std-2000, 3rd ed., 1982, reaffirmed Dec 1987 [48].

<sup>a</sup> To take credit for reduced heat input, the insulation shall resist dislodgment by a fire-hose stream, shall be non-combustible, and shall not decompose at temperatures up to 1000° F. If the insulation does not meet these criteria, the F factor for a bare vessel shall be used.

<sup>b</sup> These F factors are based on an arbitrary thermal conductivity of 4 British Thermal units per hour per square foot (per °F per inch of thickness) and a temperature differential of 1600° F when using a heat input value of 21,000 British thermal units per hour per square foot in accordance with the conditions assumed in API Recommended Practice 520. When these conditions do not exist, engineering judgment should be exercised either in selecting a higher F factor or in providing other means of protecting the tank from fire exposure.

<sup>c</sup> The insulation credit is arbitrarily limited to the F factor shown for 12 in. of insulation, even though greater thicknesses may be used. More credit, if taken, would result in a relieving device that would be impractically small but that might be used if warranted by design considerations.

<sup>d</sup> Twice the F factor for an equivalent thickness of insulation.

<sup>e</sup> Under ideal conditions, water films covering the metal surfaces can absorb substantially all incident radiation. However, the reliability of effective water applications depends on many factors. Freezing temperatures, high winds, system clogging, unreliability of the water supply, and adverse tank surface conditions are a few factors that may prevent adequate or uniform water coverage. Because of these uncertainties, the use of an F factor other than 1.0 for water application is generally discouraged.

<sup>f</sup> Depressurizing devices may be used, but no credit for their use shall be allowed in sizing safety valves for fire exposure.

F = environmental factor from Table 9.10.

A = wetted surface area, in square feet (see Table 9.20, footnote a)

Note: The formula above is based on

$$Q = 21,000A^{0.82}$$

As given in API Recommended Practice 520. The total heat absorbed, Q, is in British thermal units per hour. The constant 1107 is derived by converting the heat input value of 21,000 British thermal units per hour per square foot to standard cubic feet of free air by using the latent heat of vaporization at 60° F and the molecular weight of hexane. When the molecular weight, latent heat of vaporization, and temperature of relief conditions for refrigerated hydrocarbons are substituted in the formula based on hexane, the venting requirements are about equal to the values for hexane. Hexane has therefore been used as a basis for simplification and standardization (see Appendix B for additional information about the derivation of the formula).

The UEL for a composite is determined in the same manner, using the respective component UEL values. For the overall mixture, the above can be used to calculate the composition. Also see [56].

## 9.60 PRESSURE AND TEMPERATURE EFFECTS

The temperature and pressure of a liquid system are important in determining the effects created that result in a fire and explosion hazard. Because this relates to the flash point and flammability limits, see Tables 9-24, 9-25 and Figures 9-48 and 9-49a, and b [6].

- An increase in pressure raises the flash point, while a decrease lowers the flash point [6, 52].
- As the temperature increases on a liquid, its vapor pressure also increases and will therefore tend to vaporize at a greater rate [6].

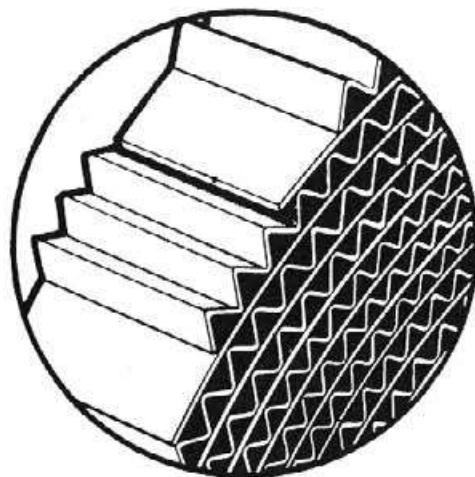
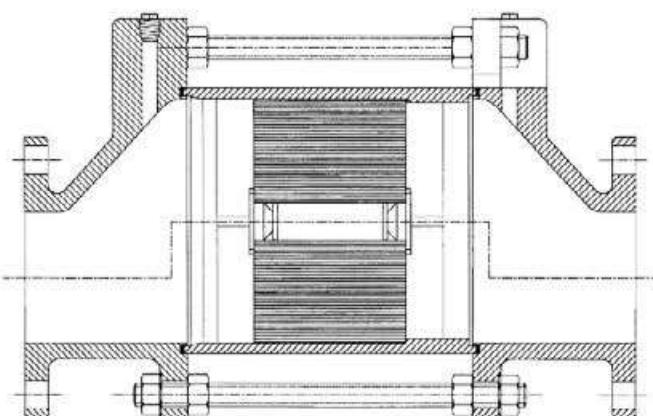
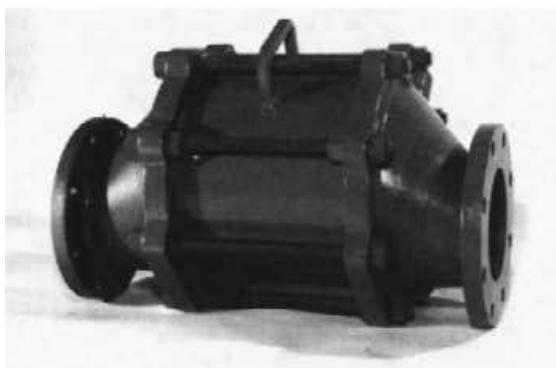
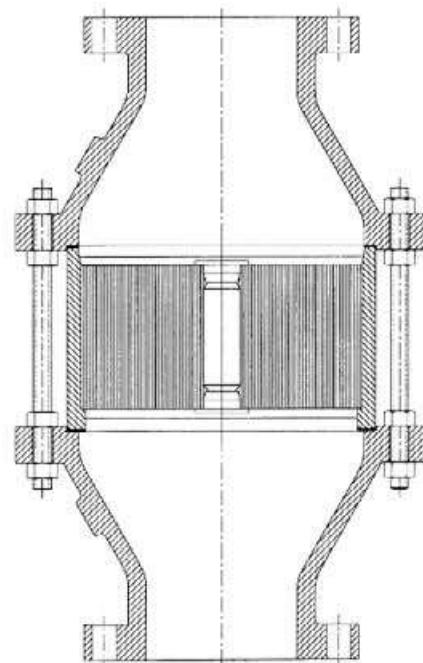
In a closed container, equilibrium develops at any given temperature and pressure, while in an open condition (not enclosed), the liquid will continue to vaporize in air until the liquid is completely vaporized. In that situation, the effects of temperature and pressure are valid only for enclosed conditions, such as tanks, vessels, piping, and other processing equipment. See Figures 9-49a, b and Table 9-25 for LEL or UEL showing a variation of limits with temperatures and pressures.

Extreme care must be exercised in designing potentially flammable systems to use reliable flammability limits data and to recognize the effects of pressure/temperature on the data and its implications to the safety of the system in question. Unless otherwise indicated, most published data are at atmospheric pressure and ambient temperature, and therefore should be corrected for other conditions.

Figure 9-47 illustrates a gas-freeing system using gasoline-air-water-vapor (the water vapor could be steam). The mixture "A" represents a saturated gasoline-vapor-air-water-vapor mixture at 70° F. In a closed tank, a more volatile gasoline than the one diagrammed would give a saturated mixture with gasoline vapor and less air. A less volatile gasoline would give less gasoline vapor and more air. If a continuous supply of air saturated with water vapor is added to a tank containing mixture A, all compositions between A and B (air plus water vapor) will be formed until all the gasoline vapor has been flushed from the tank and only steam remains (at 212° F or higher). If the tank is cooled, the steam will condense and air will be drawn into the tank giving mixtures along CB. At 70° F only air plus a small amount of water vapor will remain.

If hot water and water vapor at 175° F are used to flush mixture A from the tank, the mixture composition can only shift along AC to E. Mixtures between A and E are flushed from the tank, mixed with air to give a mixture between parts AE and B. After examining several other variations with air and water vapor, the conclusion is that mixture A cannot be flushed from a tank without forming flammable mixtures unless steam or some other inert vapor or gas is used [53].

Figure 9-48 [53] shows the effects of temperature on limits of flammability at a constant initial pressure. As the temperature increases, the lower limit decreases and the upper limit increases. This is extremely important in evaluating the explosive potential for a mixture, recognizing that just reading numbers for UEL and LEL from a table is inadequate when temperatures are above normal atmospheric (or the temperature recorded for the data). Due to this situation, even a non-flammable mixture such as part A (Figure 9-47) can propagate a flame a short distance from a source because it can become flammable as the temperature is raised, part B. The compositions along the "saturated vapor-air-mixture



**Figure 9-41** Typical individual flame arresters for tank or pipe mounting where conservation breathing vents may or may not be required, size 2–60 in. (By permission from Groth Equipment Co., Tank Protection Division.)

line and to the right of it make up the saturated and unsaturated combustible-oxidant system at a specified pressure" [53]. The engineer is urged to study Reference [53] because it deals with many more important topics of this subject than can be included here.

Figure 9-50 shows the effect of pressure on the limits of flammability of an ethane system.

### 9.61 IGNITION OF FLAMMABLE MIXTURES

Ignition can take place for any flammable mixture within the concentration ranges for the respective LEL and UEL. The conditions for ignition may vary with the specific mixture, the type of oxidant (usually air or pure oxygen), the temperature and pressure of the system. Ignition may result from electrical spark, static spark, contact with hot surfaces (autoignition) (see Figures 9-48, 9-51, and 9-52a and b) or other means. Established references include [15, 57, and 58].

Figures 9-51, 9-52a and b are convenient diagrams for studying the flammability of various compounds.

Where Autoignition Temperature (AIT) = the minimum temperature at which a material begins to self-heat at a high enough rate to result in combustion. Reported in the Data Guide (Figure 9-51) as the temperature in air at one atmosphere.

### 9.62 AQUEOUS SOLUTIONS OF FLAMMABLE LIQUIDS

Some organic compounds can be in solution with water and the mixture may still be a flammable mixture. The vapors above these mixtures such as ethanol, methanol, or acetone can form flammable mixtures with air. Bodurtha [52], Albaugh, and Pratt [20] discuss the use of Raoult's law (activity coefficients) in evaluating the effects. Figures 9-52a and b illustrate the vapor-liquid data for ethyl alcohol and the flash point of various concentrations, the shaded area of flammability limits, and the UEL. Note that some of the plots are calculated and bear experimental data verification.

### 9.63 BLAST PRESSURES

Deflagrations and detonations produce pressures associated with the resulting shock/pressure waves. These pressures can be sufficiently large to damage and/or demolish enclosed vessels, equipment, and buildings. A deflagration can produce pressure rises in excess of 8:1 and rises of 40:1 when a reflected wave develops from a detonation, referenced to the initial pressure of the system. In a containment vessel, the peak pressure from a detonation beginning at atmospheric pressure may rise to about 19.7 times or 289.8 psia. Figure 9-53 illustrates detonation velocities and pressures. Barker et al. [59] discuss blast-loaded structures.

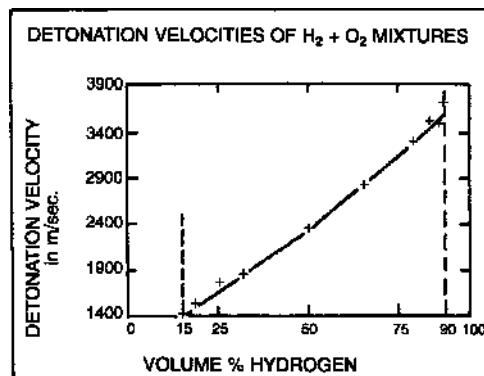
Table 9-26 presents US Atomic Energy Commission data [53], which express the overpressure above normal required to do the damage indicated. Table 9-27 presents a collection of industrial overpressure damage situations [60]. The overpressuring ( $P - P_o$ ) is expressed as [53]:

$$(P - P_o) = P_o [2k/(k-1)][v/v_s - 1] \quad (9-56)$$

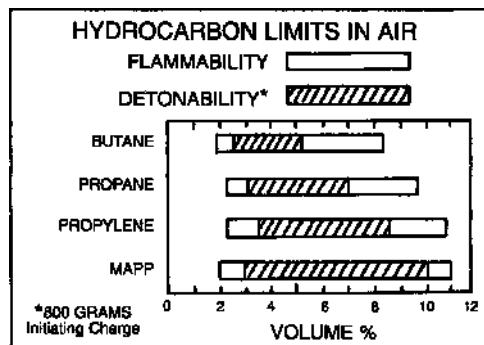
**TABLE 9-22 Comparison Data for Selected Hydrocarbon-Air Mixtures for Deflagrations and Detonations**

	Deflagrations		Detonation	
	Limits (%)	Velocity (m/s)	Limits (%)	Velocity (m/s)
CH <sub>4</sub> -Air	5.3–15	0.37	?	1540
C <sub>3</sub> H <sub>8</sub> -Air	2.2–9.5	0.40	3–7	1730
C <sub>2</sub> H <sub>2</sub> -Air	2.5–80	1.31	4.2–50	1870

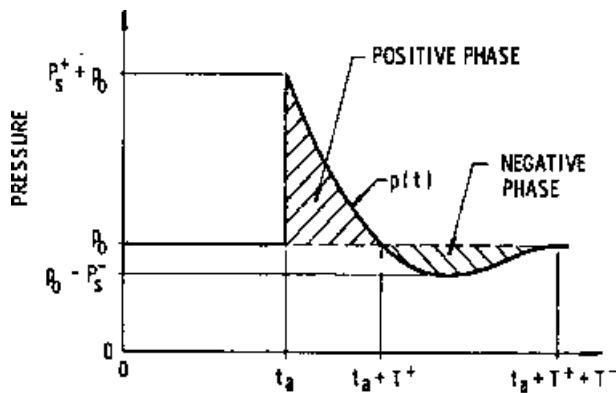
By permission from Stull [29] Dow Chemical Co. and AIChE, Monograph Series No. 10, Vol. 73 (1977).



**Figure 9-42a** Detonation velocities for hydrogen-oxygen mixtures. Note detonation range compared to flammability range of 4–95%. (By permission from [29], Stull, The Dow Chemical Co. and AIChE, Monograph No. 10, Vol. 73 (1977).)

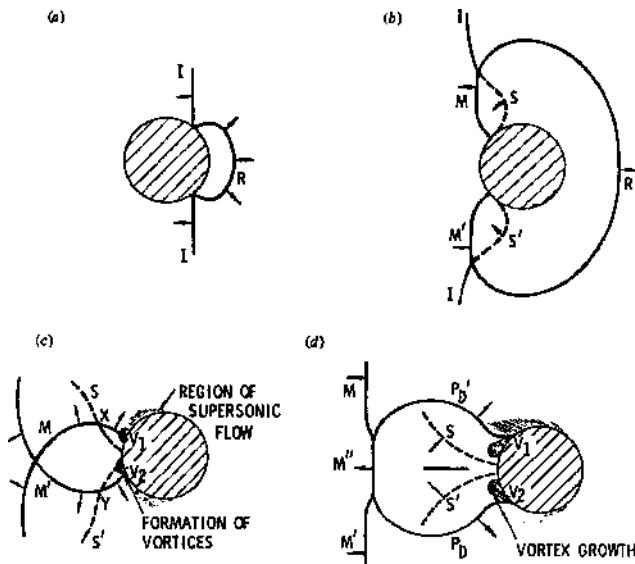


**Figure 9-42b** Comparison of flammability and detonation range limits in air for selected hydrocarbons and mixtures. (By permission from [29], Stull, The Dow Chemical Co., and AIChE, Monograph No. 10, Vol. 73 (1977).)



The gauge records ambient pressure  $P_0$ . At arrival time  $t_a$ , the pressure rises quite abruptly (discontinuously, in an ideal wave) to a peak value  $P_s^+ + P_0$ . The pressure then decays to ambient in total time  $t_a + T^+$ , drops to a partial vacuum of amplitude  $P_s^-$ , and eventually returns to  $P_0$  in total time  $t_a + T^+ + T^-$ . The quantity  $P_s^+$  is usually termed the peak side-on overpressure, or merely the peak overpressure. The portion of the time history above initial ambient pressure is called the positive phase, of duration  $T^+$ . That portion below  $P_0$ , of amplitude  $P_s^-$  and duration  $T^-$ , is called the negative phase.

**Figure 9-43** Ideal blast wave from gaseous explosion in air. (By permission from Wilfred Baker Engineering, Inc., *Explosions in Air*, 2nd printing (1983), Wilfred E. Baker, San Antonio, TX, USA [55].)

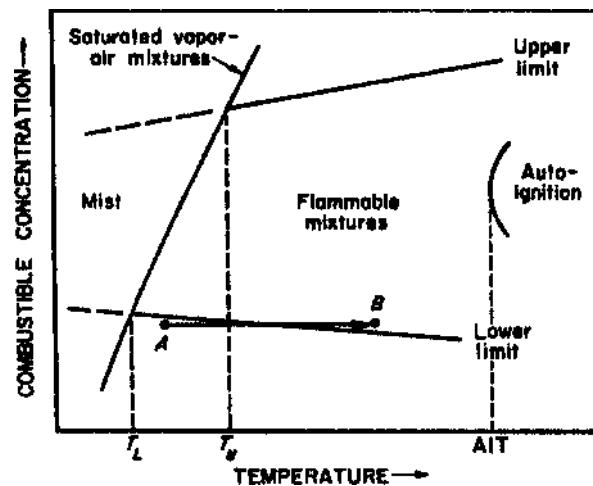


**Figure 9-44** Interaction of a shock wave with a cylinder. (Source: Bishop and Rowe, 1967) (By permission from Wilfred Baker Engineering, Inc., *Explosions in Air*, 2nd printing (1983), Wilfred E. Baker, San Antonio, TX, USA [55].)

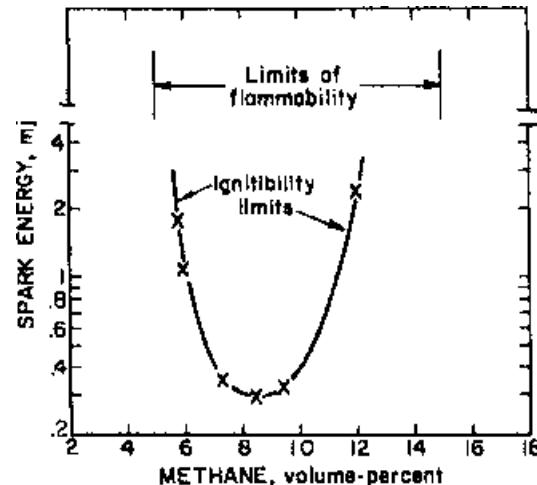
where

$k$  = ratio of specific heats

$v_s$  = velocity of sound in the medium through which the shock wave travels, ft/s



**Figure 9-45** Effect of temperature on limits of flammability of a combustible vapor in air at constant initial pressure. (By permission from US Bureau of Mines, Bulletin 627 [53].)



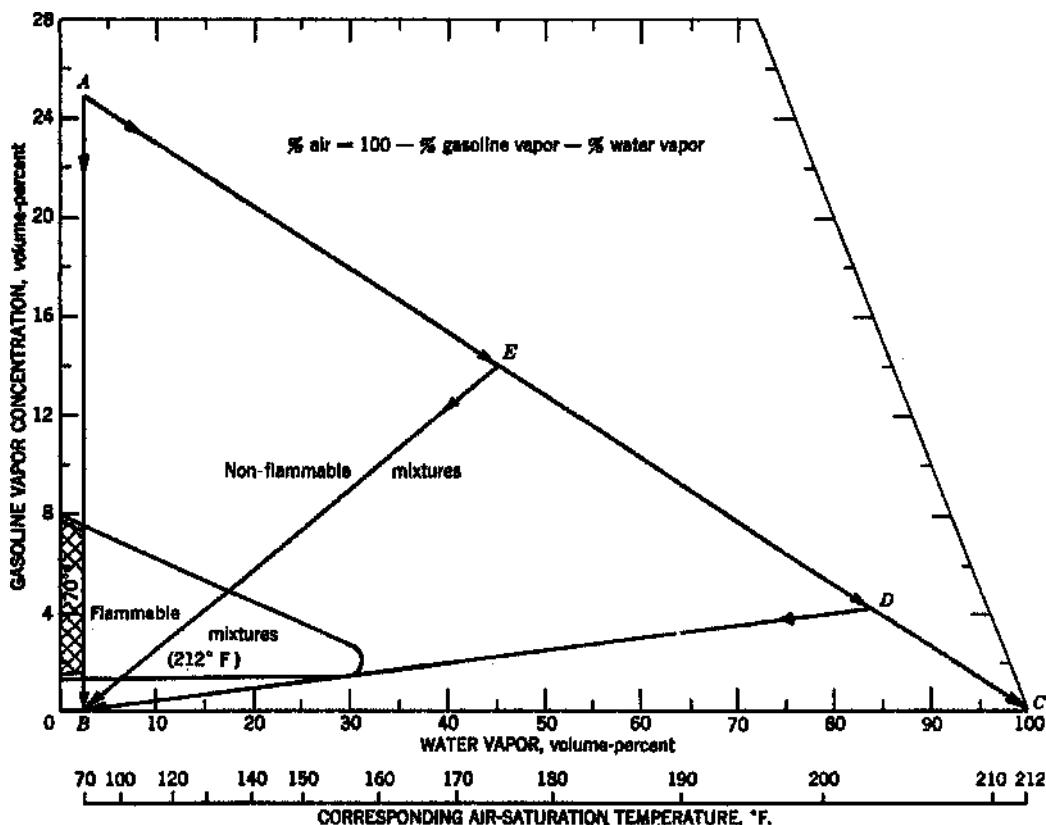
**Figure 9-46** Ignitability curve and limits of flammability for methane-air mixtures at atmospheric pressure and 26°C. (By permission from US Bureau of Mines, Bulletin 627 [53].)

$v$  = shock velocity (from test data or specific calculations), ft/s  
 $P'_i = P_o$  = initial pressure of system, psia.

$P_{\max} = P$  = peak pressure of blast, psia

Further information on the subject of blast damage are given in References [18, 19, and 21]. Tables 9-28a and b provide several miscellaneous conditions, including physiological effects of blast pressures on humans and structural damage to facilities.

Pressures of deflagration or detonation shock waves build upon the existing system pressure at the time of the initial blast. When a deflagration starts and then builds to a detonation, the resulting peak pressure can be quite high because the final pressure of the detonation builds on the peak pressure of the deflagration.



**Figure 9-47** Flammability diagram for the system gasoline vapor–water vapor–air at 70° F and at 212° F and atmospheric pressure. (By permission from US Bureau of Mines, Bulletin 627 [53].)

In an enclosure, a peak for initial pressure ratio for a deflagration generally can exceed 8:1 of the initial pressure. When a detonation develops, the ratio of the pressure developed (reflected pressure) to that of the initial pressure is 40:1. This is the reason detonations can be so disastrous. Their final pressure, when built on a deflagration peak pressure as a base or initial pressure, can be extremely high.

For example, see Figures 9-53 [53] and 9-54.

The overpressure levels, that is final peak less initial starting pressure, shown in Table 9-26 [53] are low referenced to the magnitude of overpressures attainable from many industrial blasts.

The speed of a combustion reaction will be at a maximum at a certain fuel–air ratio that is generally close to the stoichiometric composition. It will be lower, however, for compositions closer to each of the explosive limits. The rate of pressure rise is a measure of the speed of flame propagation and accordingly the violence of the explosion. The rate of pressure rise is the slope of the tangent line (Figure 9-69) through the rising branch of the pressure, which is the time curve of an explosion [60]. The greatest rate of pressure rise will occur when an explosive mixture ignites in the center of a vessel. Ignition at any other location will result in a lower rate and somewhat reduced explosion pressure. The volume of the vessel influences the violence of the explosion (Figures 9-55 and 9-56). For example, Bartknecht's data [30] show that for propane exploding in three separate but different sized vessels, the magnitude of the final maximum pressure is the same for each of about 7 bar, but the time to reach this maximum pressure is greater the larger the vessel (Figure 9-57).

The influence of vessel volume on the maximum rate of pressure rise for a specified gas is characterized by the Cubic Law, that is, [30] the rate of pressure rise varies for each gas.

$$\left( \frac{dp}{dt} \right)_{\max} (V)^{1/3} = \text{Constant} = K_G \quad (9-57)$$

where

$V$  = vessel volume, m<sup>3</sup>

$K_G$  = constant (bar) (m/s).

This is valid for the same degree of gas mixture turbulence and the same ignition source and is illustrated in Figure 9-58. Influence of the vessel shape is shown in Figure 9-57. The behavior of propane is considered representative of most flammable vapors including many solvents [30]. The maximum explosion pressure does not follow the cubic law and is almost independent of the volume of a vessel greater than one liter. For propane, town gas, and hydrogen, the volume relationship can be expressed as:

$$\frac{1}{V^{1/3}} < 5.5, \text{ or } \frac{\text{surface}}{\text{volume}} < \frac{1}{30 \text{ m}} \cong \frac{dp}{dt} \quad (9-58)$$

The violence of an explosion is influenced by the initial pressure or pressure of the system in which the explosion takes place. Figure 9-56 illustrates this point for propane and a constant ignition energy source. For low pressure below atmospheric, the explosion reactions are reduced until they do not propagate through the fuel–air mixture [30].

**TABLE 9-23 Properties of Flammable Liquids, Gases, and Solids**

Name	Synonym	Formula	Flash point (° F)	
			Closed Cup	Open Cup
Ethyleneimine	Dimethlenimine	NH(CH <sub>2</sub> ) <sub>2</sub>	12	—
Ethylene oxide	1,2-Epoxyethane	CH <sub>2</sub> OCH <sub>2</sub>	Gas	-4
Ethylethanolamine-1	Ethylaminoethanol	C <sub>2</sub> H <sub>5</sub> NHCH <sub>2</sub> CH <sub>2</sub> OH	—	160
Ethyl ether	See Diethyl ether		—	—
Ethyl formate	Ethyl methanoate	HCOOC <sub>2</sub> H <sub>5</sub>	-4	10
Ethyl hexaldehyde	—	C <sub>4</sub> H <sub>9</sub> CH(C <sub>2</sub> H <sub>5</sub> )CHO	—	125
2-Ethylhexanediol-1,3	Ethylhexylene glycol	C <sub>2</sub> H <sub>7</sub> CH(OH)CH(C <sub>2</sub> H <sub>5</sub> )CH <sub>2</sub> OH	—	260
2-Ethylhexanoic acid	Octoic acid	CH <sub>3</sub> (CH <sub>2</sub> ) <sub>2</sub> CH(C <sub>2</sub> H <sub>5</sub> )CH <sub>2</sub> OH	—	260
Ethyl hexanol	2-Ethyl hexyl alcohol	C <sub>4</sub> H <sub>9</sub> CH(C <sub>2</sub> H <sub>5</sub> )CH <sub>2</sub> OH	—	185
Ethylhexyl acetate	Octyl acetate	CH <sub>3</sub> COOCH <sub>2</sub> CH(C <sub>2</sub> H <sub>5</sub> )C <sub>4</sub> H <sub>9</sub>	180	190
Ethyl lactate	Ethyl 2-hydroxy propanoate	CH <sub>3</sub> CHOHCOOC <sub>2</sub> H <sub>5</sub>	115	158
Ethyl mercaptan	Ethanethiol	C <sub>2</sub> H <sub>5</sub> SH	<80	—
Ethylmonobromoacetate	—	BrCH <sub>2</sub> COOC <sub>2</sub> H <sub>5</sub>	118	—
Ethylmonochloroacetate	—	CClH <sub>2</sub> COOC <sub>2</sub> H <sub>5</sub>	100	—
Ethyl morpholine	—	CH <sub>2</sub> CH <sub>2</sub> OCH <sub>2</sub> CH <sub>2</sub> NCH <sub>2</sub> CH <sub>3</sub>	—	90
Ethyl nitrate	Nitric ether	C <sub>2</sub> H <sub>5</sub> ONO <sub>2</sub>	50	50
Ethyl nitrite	Nitrous ether	C <sub>2</sub> H <sub>5</sub> ONO	-31	—
Ethyl oxalate	Diethyl oxalate	(COOC <sub>2</sub> H <sub>5</sub> ) <sub>2</sub>	168	—
Ethyl phenyl ethanolamine	—	C <sub>6</sub> H <sub>5</sub> NC <sub>2</sub> H <sub>5</sub> CH <sub>2</sub> CH <sub>2</sub> OH	—	270
Ethyl phthalyl ethyl glycolate	—	C <sub>2</sub> H <sub>5</sub> OCOC <sub>6</sub> H <sub>4</sub> COOCH <sub>2</sub> COOC <sub>2</sub> H <sub>5</sub>	365	385
Ethyl propionate	Propionic ether	C <sub>2</sub> H <sub>5</sub> COOC <sub>2</sub> H <sub>5</sub>	54	—
2-Ethyl-3-propylacrolein	2-Ethylhexanal	C <sub>3</sub> H <sub>7</sub> CH : C(C <sub>2</sub> H <sub>5</sub> )CHO	—	155
Ethyl <i>n</i> -propyl ether	1-Ethoxypropane	C <sub>2</sub> H <sub>5</sub> OC <sub>3</sub> H <sub>7</sub>	—	—
Ethyl silicate	Ethyl orthosilicate	(C <sub>2</sub> H <sub>5</sub> ) <sub>2</sub> SiO <sub>4</sub>	—	125
Ethyl <i>p</i> -toluen sulfonamide	—	C <sub>7</sub> H <sub>7</sub> SO <sub>2</sub> NHC <sub>2</sub> H <sub>5</sub>	260	380
Ethyl <i>p</i> -toluen sulfonate	—	C <sub>7</sub> H <sub>7</sub> SO <sub>3</sub> C <sub>2</sub> H <sub>5</sub>	316	—
Fish oil	—	—	420	—
Fluorine	—	F <sub>2</sub>	Gas	Gas
Formal	See Methylal	—	—	—
Formaldehyde gas	Methanal	HCOH	Gas	Gas
Formaldehyde-37% in water	Formalin	—	130	200
Formic acid	Methanoic acid	HCOOH	156	—
Fuel oil No. 1	Range oil; kerosene	—	114-185	—
Fuel oil No. 1-D	Diesel fuel, light	—	>100	—
Fuel oil No. 2	Domestic fuel oil	—	126-230	—
Fuel oil No. 2-D	Diesel fuel, medium	—	>100	—
Fuel oil No. 4	Light industrial fuel	—	154-240	—
Fuel oil No. 5	Medium industrial fuel	—	130-310	—
Fuel oil No. 6	Heavy industrial fuel; Bunker C	—	150-430	—
Furan	Furfurane; oxole	HC:CHCH:CHO	-32	—
Furfural	Fural:2-furaldehyde	C <sub>4</sub> H <sub>3</sub> OCHO	140	155
Furfuryl alcohol	Furfuryl carbinol	C <sub>4</sub> H <sub>3</sub> OCH <sub>2</sub> OH	—	167
Furfurylamine	2-furanmethylamine	C <sub>4</sub> H <sub>3</sub> OCH <sub>2</sub> NH <sub>2</sub>	—	99
Gas, blast furnace	—	—	Gas	Gas
Gas, coal gas	—	—	Gas	Gas
Gas, coke oven	—	—	Gas	Gas
Gas, manufactured, 540 Btu	—	—	Gas	Gas
Gas, manufactured, 815 Btu	—	—	Gas	Gas
Gas, natural, 1035 Btu	—	—	Gas	Gas
Gas oil gas	—	—	Gas	Gas
Gas, producer	—	—	Gas	Gas
Gas, water	—	—	Gas	Gas
Gas, water-carbureted	—	—	Gas	Gas
Gas oil	(65 Diesel Index)	—	150-165	—
Gasoline, automotive-premium	—	—	-50±	—
Gasoline, automotive-regular	—	—	-50±	—
Gasoline, aviation-commercial	—	—	-50±	—
Gasoline, aviation-military	—	—	-50±	—
Glycerol	Glycerin	HOCH <sub>2</sub> CHOHCH <sub>2</sub> OH	320	350
Glyceryl triacetate	Triacetin	(C <sub>3</sub> H <sub>5</sub> )(OOCH <sub>2</sub> ) <sub>3</sub>	280	295
Glycol diformate	See Ethylene glycol diacetate			
Glycol diformate	Ethylene glycol diformate	HCOOCH <sub>2</sub> CH <sub>2</sub> OOCH	—	200
Heat transfer oils				
Heavy paraffinic	—	—	—	550
Light paraffinic	—	—	—	400
Light aromatic	—	—	—	280
Medium aromatic	—	—	—	360
Heptadecanol- <i>n</i>	Heptadecyl alcohol	C <sub>17</sub> H <sub>35</sub> OH	—	310
Heptane- <i>n</i>	Dipropylmethane	CH <sub>3</sub> (CH <sub>2</sub> ) <sub>5</sub> CH <sub>3</sub>	25	30
Heptane-iso	—	(CH <sub>3</sub> ) <sub>2</sub> CHCH <sub>2</sub> CH <sub>2</sub> CH <sub>2</sub> CH <sub>3</sub>	<0	—

TABLE 9-23—(continued)

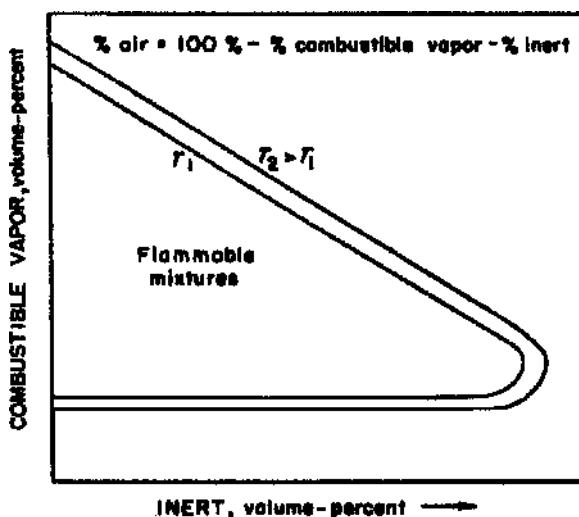
Explosive limits in air, % by vol.		Auto-ignition temperature (°F)	Specific gravity (water = 1.0)	Vapor density (air = 1.0)	Melting point (°F)	Boiling point or range (°F)	Water solubility (miscibility)	Suitable extinguishing agents	Hazard
Lower	Upper								
3.6	46	612	0.832	1.48	-97	132	∞	1	D,G,I
3	100	804	0.887	1.52	-168	51	∞	1, 3, 4	B,D,G,I
		.....	0.918	3.00	18	322	∞	1, 3	B,D,G
2.7	16.5	851	0.924	2.55	-122	130	s	2, 3	B,D,G
		.....	0.821	4.42	.....	325	sl	2, 3	B,D,G,
		.....	0.942	5.03	-40	472	sl	1, 2, 3	B
		.....	0.908	4.98	-180	440	sl	1, 2, 3	B
		.....	0.834	4.49	-105	359	sl	1, 2, 3	B,D,G
		.....	0.873	5.93	-37	390	sl	1, 2, 3	B,D,G,
1.5 at 212°F		752	1.03	4.07	.....	309	∞	1, 3	B,D,G
2.8	18.2	570	0.839	2.11	-234	96	sl	3, 4	D,G
		.....	1.484	.....	-40	318	i	2a, 3	D,G
		.....	1.156	.....	-15	295	i	2a, 3	D,G
		.....	0.916	4.00	.....	280	∞	1, 2, 3	B,D,G
3.8		.....	1.105	3.14	-152	190	sl	2, 3	D,G
3.0	>50	194	0.900	2.59	.....	63	sl	2, 3, 4	D,G
		.....	1.090	5.04	-41	367	d	1, 3	B,D,G
		.....	1.040	5.35	99	514	....	2, 3	B
		.....	1.180	9.60	.....	608	....	1, 2, 3	B
1.8	11	890	0.891	3.52	-99	210	sl	2, 3	B,D,G
		.....	0.851	4.35	.....	347	sl	1, 2, 3	B,D,G
1.9	24	.....	0.747	.....	<-110	147	s	2a, 3	B,D,G,I
		.....	0.936	7.22	-114	334	d	3	D,G
		.....	1.253	5.60	.....	207	....	1, 3	B,D
		.....	1.17	6.91	93	439	i	1, 2, 3	B,D
		.....	.....	.....	.....	.....	....	1, 2, 3	C
		.....	.....	1.31	-369	-305	d	4	A,E,F
7	73	806	.....	1.07	-134	-6	vs	4	D,G,H,I
		.....	795	1.080	1.03	207	s	1, 3	D,G
		.....	1114	1.218	1.59	213	∞	1, 3	D,G
0.6	5.6	445–560	0.78–0.85	.....	.....	340–355	i	2, 3	B,D,G
1.3	6.0	350–625	<1.0	.....	.....	<590	i	2, 3	B,D,G
		500–705	0.80–0.90	.....	.....	340–640	i	2, 3	B,D,G
1.3	6.0	490–545	0.81	.....	.....	380–650	i	2, 3	B,D,G
1	5	505	0.84–0.98	.....	.....	425–760	i	1, 2, 3	B,D,G
1	5	.....	0.92–1.06	.....	.....	.....	i	1, 2, 3	B,D,G
1	5	765	0.92–1.07	.....	.....	.....	i	1, 2, 3	B,D,G
2.3	14.3	.....	0.937	2.35	-122	88	i	3	B,D,F1,F2,G,H,I
2.1 at 257°F		600	1.161	3.31	-34	322	s	1, 2, 3	B,D,F6
1.8	16.3	736	1.129	3.37	-18	340	∞	1, 2a, 3	B,D,H
		915	1.050	3.35	-94	295	∞	2a, 3	B,D
35	74	.....	.....	.....	.....	.....	....	4	D,H,I
5.3	32	1200	.....	0.47	.....	.....	....	4	D,H,I
4.4	34	.....	.....	0.44	.....	.....	....	4	D,H,I,
		.....	.....	0.38	.....	.....	....	4	D,H,I,
		.....	.....	0.50	.....	.....	....	4	D,H,I
3.8–6.5	13–17	>1000	.....	0.61	.....	.....	....	4	D,H,I
4.7	33	637	.....	0.47	.....	.....	....	4	D,H,I
17–35	70–80	.....	.....	0.86	.....	.....	....	4	D,H,I
7.0	72	.....	.....	0.57	.....	.....	....	4	D,G,H,I
5.5	36	.....	.....	0.63	.....	.....	....	4	D,G,H,I
6.0	13.5	640–690	.....	.....	.....	599–649	i	1, 2, 3	B,D,H
1.3–1.4	6.0–7.6	700	0.71–0.76	3.0–4.0	<-76	91–403	i	2, 3	B,D,G
1.3–1.4	6.0–7.6	700	0.70–0.75	3.0–4.0	<-76	91–401	i	2, 3	B,D,G
1	6.0–7.6	800–880	0.70–0.71	3.0–4.0	<-76	108–318	i	2, 3	B,D,G
1	6.0–7.6	800–880	0.70–0.71	3.0–4.0	<-76	107–319	i	2, 3	B,D,G
		739	1.26	3.17	64	554	∞	1, 3	B,D
		812	1.161	7.52	-108	496	s	1, 2a, 3	B,D
		.....	1.227	4.07	14	350	d	3	B,D
		600	0.92	.....	.....	>550	i	1, 2, 3	B,D
		550	0.89	.....	.....	>550	i	1, 2, 3	B,D
		700+	0.99	.....	.....	>550	i	1, 2, 3	B,D
		700+	0.97	.....	.....	>550	i	1, 2, 3	B,D
		.....	0.848	8.84	130	588	sl	1, 2, 3	B
1.2	6.7	433	0.688	3.45	-131	209	i	2, 3	B,D,G
1	6	.....	0.725	3.45	-182	176–195	i	2, 3	B,D,G

By permission from Factory Mutual Engineering Corp./McGraw-Hill book Co., *Handbook of Industrial Loss Prevention*, 2nd Ed., 1967, pp. 42–16 and 42–17 [4]. This is only partial selected listing of reference.

**TABLE 9-24** Sample Listing of Properties of Flammable Liquids

Name	Flash Point (°F)		Explosive Limits in Air % by Volume		Autoignition Temperature (°F)	Vapor Density (Air = 1.0)
	Closed Cup	Open Cup	Lower	Upper		
Acetaldehyde	-36	-	4.0	55.0	365	1.52
Acetone	0	15	2.1	13.0	1000	2.00
Ammonia (Anhydrous)	Gas	Gas	15	28	1204	0.596
Amyl acetate- <i>n</i>	76	80	1.1	7.5	714	4.49
Amyl Alcohol- <i>n</i>	91	120	1.2	-	572	3.04
Benzene (Benzol)	12	-	1.4	7.1	1044	2.77
Benzine	<0	-	1.4	5.9	550	2.50
Butyl Acetate- <i>n</i>	72	90	1.4	7.6	790	4.00
Butyl Alcohol- <i>n</i>	84	110	1.4	11.2	693	2.55
Camphor	150	200	-	-	871	5.24
Carbon Disulfide	-22	-	1.0	50	257	2.64
Carbon Tetrachloride	None	None	-	-	-	-
Cellosolve	104	120	2.6	15.7	460	3.10
Chloroform	None	None	-	-	-	4.13
Coal Tar Oil	80-160	-	-	-	-	-
Coal Tar Pitch	405	490	-	-	-	-
<i>o</i> -Cresol	178	-	1.3 at 300° F	-	1038	3.72
Cyclohexanol	154	-	-	-	572	3.45
Denatured Alcohol - 95%	60	-	-	-	750	1.60
Ethyl Acetate	24	30	2.2	11	800	3.04
Ethyl Alcohol (Ethanol)	55	70	3.5	19	737	1.59
Ethylene Glycol	232	240	3.2	-	775	2.14
Ethyl <i>n</i> -propyl ether	-	-	1.9	24	-	-
Formaldehyde, 37% in water	130	200	-	-	795	1.03
Fuel oil No. 1	114-185	-	0.6	5.6	445-560	-
Fuel oil No. 1-D	100 min	-	1.3	6	350-625	-
Fuel oil No. 2	126-230	-	-	-	500-705	-
Fuel oil No. 2-D	100 min	-	1.3	6	490-545	-
Fuel oil No. 4	154-240	-	1	5	505	-
Fuel oil No. 5	130-310	-	1	5	-	-
Fuel oil No. 6	150-430	-	1	5	765	-
Gasoline Automotive premium	-50±	-	1.3-1.4	6.0-7.6	770	3.0-4.0
Gasoline Automotive regular	-50±	-	1.3-1.4	6.0-7.6	700	3.0-4.0
Gasoline Aviation, commercial	-50±	-	1	6.0-7.6	800-880	3.0-4.0
Gasoline Aviation, military	-50±	-	1	6.0-7.6	800-880	3.0-4.0
Hexane- <i>n</i>	-7	-	1.2	7.5	453	2.91
Hexane-iso	<20	-	1	7	-	3.00
Hydrogen sulfide	Gas	Gas	4.3	45.5	500	1.18
Jet fuel JP-1	110-125	-	0.6	5.6	442-560	-
Jet fuel JP-4	26-36	-	0.8	6.2	468	-
Kerosene	110-130	-	0.6	5.6	440-560	4.5
Lacquer	0-80	-	-	-	-	-
Maleic Anhydride	218	240	-	-	890	3.38
Methyl Acetate	15	20	3.1	16	935	2.56
Methyl Alcohol (Methanol)	54	60	5.5	36.5	878	1.11
Methyl Ethyl Ketone	30	-	1.8	10	960	2.48
Mineral spirits	100 min	110	0.7 at 212° F	-	475	3.9
Naphtha	100-110	-	0.8	5	440-500	-
Naphtha VM&P	20-45	-	0.9	6.0	450-500	3.75
Naphthalene	174	190	0.9	5.9	979	4.42
Petroleum crude	20-90	-	-	-	-	-
Petroleum ether	<0	-	1.4	5.9	550	2.50
Phenol	175	185	-	-	1319	3.24
Phthalic Anhydride	305	330	1.7	10.5	1083	5.10
Pine oil	172	175	-	-	-	-
Propane	≤100	Gas	2.2	9.6	871	1.56
Propyl Acetate- <i>n</i>	58	70	1.7	8.0	842	3.52
Propyl Alcohol-iso	53	60	2.5	12	750	2.07
Quenching oil	365	405	-	-	-	-
Stoddard solvent	100-110	-	0.8	5	440-500	-
Styrene	90	-	1.1	6.1	914	3.60
Sulfur	405	440	-	-	450	-
Toluene	40	45	1.3	7.0	997	3.14
Trichloroethylene		Weakly flammable	10 in. O <sub>2</sub>	65 in. O <sub>2</sub>	-	4.53
Turpentine	95	-	0.8	-	488	4.84
<i>p</i> -Xylene	77	-	1.1	7.0	984	3.66

By permission from The Protectoseal Co., Technical Manual.



**Figure 9-48** Effect of initial temperature on limits of flammability of a combustible vapor–inert–air system at atmospheric pressure. (By permission from US Bureau of Mines, Bulletin 627 [53].)

For some mixtures, unusual conditions seem to develop in the rate of pressure rise at peak explosion pressures due to possible changes in the violence of the reaction [30]. Similar results are reported for the level of energy required for ignition of a mixture related to the concentration range for ignition. Without examining the energy level versus concentration at various initial pressures, it might erroneously be concluded that some fuel–air mixture were “not explodable.” For a pipeline of flammable gas/vapor with one end closed and the other open, an explosion originating at the closed end will produce a higher velocity and higher pressure explosion than if ignition originates at the open end [30].

The violence of an explosion increases when the shape of the vessel changes from spherical to a more elongated shape or has a ratio of length: diameter of 1:1 to 1:5, with length several times the diameter due to the flame front of the explosion moving swiftly in the axial direction and compressing at the end of the vessel thereby energizing the violence of the explosion (see Figures 9-57). Accordingly, the shape of the vessel is an important consideration when evaluating the explosion potential of a mixture. All remarks above are related to non-turbulent mixtures, because turbulence increases the violence of the explosion (Figure 9-58).

Figure 9-58 shows the differences in  $K_G$  value and maximum explosion pressures for turbulent and non-turbulent systems for the same fuel–air mixtures.

Mixtures of flammable gas/vapors plus oxygen when ignited can form more violent explosions with greater peak pressures than when ignited in air.

Somewhat unusual and/or unexpected mixtures of gases/vapors can form explosive mixtures. Some, but not all inclusive, of these include the following [6].

- Chlorine with hydrogen, ammonia, acetylene, turpentine, or powdered metals. Steel will burn in the presence of chlorine.
- Bromine causes fire in contact with combustible materials.
- Iodine is explosive with ammonia, turpentine, or lead triethyl.
- Fluorine reacts spontaneously with almost all elements, hydrogen, water vapor, and many organic compounds. Steel will melt and ignite in fluorine with a violent reaction.

See [6] for a more complete listing of corrosive chemicals, water and air-reactive chemicals, unstable chemicals, combustible chemicals, and oxidizing chemicals.

## 9.64 TRI-NITRO TOLUENE (TNT) EQUIVALENCE FOR EXPLOSIONS

The explosion of a quantity of TNT has been established as the standard for defining or comparing the blast effects of other explosive materials [19]. The blast waves generated by other chemical explosives generally differ somewhat from the waves of TNT explosions, including peak pressure and impulse, but are similar in other effects. The equivalent energy of TNT is 1120 cal/g (this value varies with some data). Other types of explosives are related by their charge weights and can be converted to their TNT equivalent,  $e_t$ :

$$W_{\text{TNT}} = (e_t)(W_c) \quad (9-59)$$

where

$$\begin{aligned} W_{\text{TNT}} &= \text{equivalent charge weight of TNT, lb} \\ e_t &= \text{equivalence factor, see Table 9-29} \\ W_c &= \text{charge weight of explosives of interest, lb.} \end{aligned}$$

Table 9-29 [19] has been developed by ratio of relative heats of explosion. For close explosion, that is ( $Z < 3.0 \text{ ft/lb}^{1/3}$ ) and for shapes other than spherical, the TNT equivalent factor can be much greater than those from relative heats of explosion [19].

## 9.65 PRESSURE PILING

If two or more systems are connected together (such as a pipe length with an orifice plate, two or more vessels connected with pipe or duct, or a compartmented vessel) and an explosion develops in No. 1 area, which generally may be at equilibrium pressure with compartments Nos 2 and 3 in equilibrium with No. 2, it can cause a pressure rise in front of the flame front in the unburnt gases in the interconnecting spaces (pipe, compartment). The increased pressure in compartment or area No. 1 becomes the starting pressure for an explosion in compartment No. 2 and, by the same analysis, this increased pressure in No. 2 becomes the starting pressure for an explosion in No. 3. This pressure buildup under these types of conditions is known as pressure piling [51]. From a pressure buildup standpoint,

1. when the initial pressure in compartment No. 1 is  $p_1$ , the final pressure will be  $(p_1)(x)$
2. the final pressure in compartment No. 2 could be  $x^2 p_1$
3. the final pressure in compartment No. 3 could be  $x^3 p_1$ .

where  $x$  is ratio of pressure increase, often with a value between 2 and 8 for a deflagration.

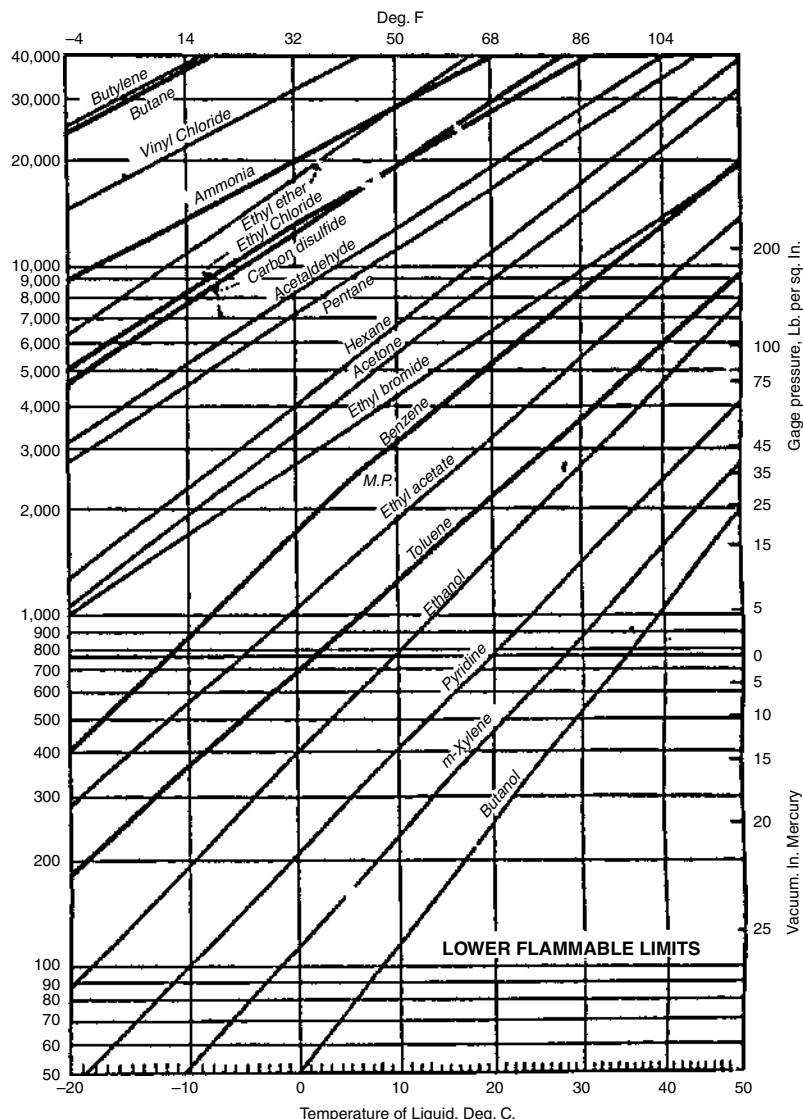
For example, if  $p_1 = 20 \text{ psig} + 14.7 = 34.7 \text{ psia}$ , assume  $x = 6.5$ .

Thus, final pressures in compartment No. 2 would be  $= (6.5)^2(34.7) = 1466 \text{ psia}$ .

Thus, it is easy to recognize that the pressure buildup in a process system can be dangerously large and requires attention to both pressure relieving and the design pressures for vessels/equipment. This also helps explain why some process vessels fragment during an explosion, and fragments impact on personnel, buildings, and so on to do damage. It also helps to explain the shock wave effects.

## 9.66 BLAST SCALING

The Universal Hopkinson-Cranz and Sachs Laws of Blast Scaling have both been verified by experiment. These laws state that self-similar blast (shock) waves are produced at identical scaled



This chart is applicable only to flammable liquids or gases in equilibrium in a closed container. Mixtures of vapor and air will be too lean to burn at temperatures below and at pressures above the values shown by the line on the chart for any substance. Conditions represented by points to the left of and above the respective lines are accordingly non-flammable. Points where the diagonal lines cross the zero gauge pressure line (760 mm of mercury absolute pressure) indicate flash point temperatures at normal atmospheric pressure.

**Figure 9-49a** Variation of lower flammable limits with temperature and pressure. (Reprinted by permission from *Fire Protection Handbook*, 17th ed. (1991), National Fire Protection Association, Quincy, MA 02269 [6].)

#### EXAMPLE 9-17

##### Estimating Blast Pressures and Destruction

A process petrochemical plant producing a synthesis gas high in hydrogen experiences an explosion that results in the destruction of a 1500 ft<sup>3</sup> storage vessel normally held at 50 psig. Unprotected glass windows (i.e., no wire mesh reinforcing, nor tempered) in the plant area 150 ft away from the tank are broken. What pressures were involved?

Using the equation for isentropic expansion of an ideal gas:

Using the equation for isentropic expansion of an ideal gas:

$$W_c = (P_1 V_1) / (k - 1) \left[ 1 - (P_2/P_1)^{(k-1)/k} \right], \text{ ft lb} \quad (9-60)$$

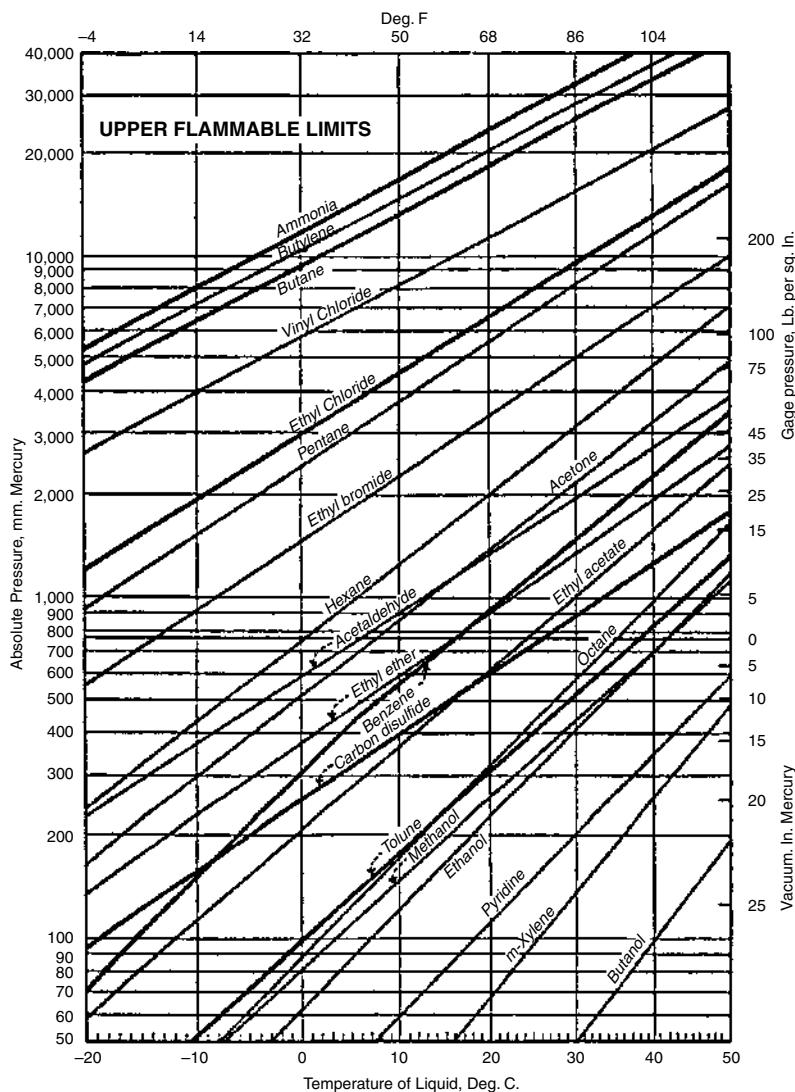
where

$$V_1 = 1500 \text{ ft}^3$$

$$k = 1.41 \text{ for hydrogen}$$

$$P_1 = 50 + 14.7 = 64.7 \text{ psia } (144 \text{ in.}^2/\text{ft}^2) = 9316.8 \text{ lb/ft}^2$$

(continued)



This chart is applicable only to flammable liquids or gases in equilibrium in a closed container. Mixtures of vapor and air will be too "rich" to be flammable at temperatures above and pressures below the values shown by the lines on the chart for any substance. Conditions represented by points to the right of and below the respective lines are accordingly nonflammable.

**Figure 9-49b** Variation of upper flammable limits with temperature and pressure. (Reprinted by permission from *Fire Protection Handbook*, 17th ed. (1991), National Fire Protection Association, Quincy, MA 02269 [6].)

**EXAMPLE 9-17—(continued)**  
 $P_2 = 14.7 \text{ psia}$  final pressure.

$$\begin{aligned}
 W_c &= \left[ \frac{(9316.8)(1500)}{(1.41 - 1)} \right] \left[ 1 - \left( \frac{14.7}{64.7} \right)^{\frac{(1.41-1)}{1.41}} \right] \\
 &= 1.193 \times 10^7 \text{ ft lb} \\
 &= (1.193 \times 10^7)(0.3241 \text{ cal}/\text{ft lb}) \\
 W_c &= 0.38674 \times 10^7 \text{ cal}
 \end{aligned}$$

$$\text{Mass of TNT} = m_{\text{TNT}} = 0.38674 \times 10^7 / 1120 \text{ cal/g TNT} = 3453 \text{ g TNT}$$

$$\text{Mass of TNT in lb} = 3453 / 453 \text{ g/lb} = 7.623 \text{ lb TNT}$$

$$\text{Using scaling: } Z = 150 / (7.623)^{1/3} = 76.22 \text{ ft/lb}^{1/3}$$

(See following paragraph.)  
 Reading chart, Figure 9-59

overpressure,  $p^\circ = 0.37 \text{ psi}$  at a distance of 150 ft.

This overpressure would produce some glass breakage, ceiling damage, and minor structural damage. See Tables 9-27 and 28a and b.

**TABLE 9-25 The Effect of Elevated Temperature on the Lower Flammable Limit of Combustible Solvents as Encountered in Industrial Ovens\***

Solvent	Flash Pt Closed Cup	Lower Flammable Limit Percent Vapor by Volume at Initial Temperature (° F)					
		Room	212	392	437	482	572
Acetone	3	2.67	2.40	2.00 <sup>†</sup>	—	—	—
Amyl Acetate, Iso	77	—	1.00	0.82	—	0.76 <sup>†</sup>	—
Benzene	-4	1.32	1.10	0.93	—	—	0.80 <sup>†</sup>
Butyl Alcohol, normal	100	—	1.56	1.27	1.22 <sup>†</sup>	—	—
Cresol, Meta-Para	202	—	1.06 <sup>††</sup>	0.93	—	0.88 <sup>†</sup>	—
Cyclohexane	-4	1.12	1.01	0.83 <sup>†</sup>	—	—	—
Cyclohexanone	111	—	1.11	0.96	0.94	0.91 <sup>†</sup>	—
Ethyl Alcohol	54	3.48	3.01	2.64	—	2.47	2.29 <sup>†</sup>
Ethyl lactate	131	—	1.55	1.29	—	1.22 <sup>†</sup>	—
Gasoline	-45	1.07	0.94	0.77 <sup>†</sup>	—	—	—
Hexane, Normal	-15	1.08	0.90	0.72 <sup>†</sup>	—	—	—
High-Solvency Petroleum Naphtha	36	1.00	0.89	0.74	0.72	0.69 <sup>†</sup>	—
Methyl Alcohol	52	6.70	5.80	4.81	—	4.62	4.44 <sup>†</sup>
Methyl Ethyl Ketone	21	1.83	1.70	1.33 <sup>†</sup>	—	—	—
Methyl Lactate	121	—	2.21	1.86	1.80	1.75 <sup>†</sup>	—
Minerals Spirits, No. 10	104	—	0.77	0.63 <sup>†</sup>	—	—	—
Toluene	48	1.17	0.99	0.82	—	—	0.72 <sup>†</sup>
Turpentine	95	—	0.69	0.54 <sup>†</sup>	—	—	—
V.M. and P. Naphtha	28	0.92	0.76	0.67 <sup>†</sup>	—	—	—

Reprinted by permission from *Fire Protection Handbook*, 17th ed. (1991) p. 4-32. National Fire Protection Association [6].

\* From NFPA Quarterly, Apr 1950; UL Bulletin of Research No. 43.

† Rapid and extensive thermal decomposition and oxidation reactions in vapor-air mixture at this temperature.

‡ Lower limit determined at 302°F.

distances when two explosive charges of similar geometry and the same explosive composition, but of different size, are detonated in the same atmosphere [19].

The scaled distance  $Z$ , which is proportionality constant, is as follows.

$$Z_{\text{TNT}} = \frac{R}{m_{\text{TNT}}^{1/3}}, \text{ for TNT only, ft/lb}^{1/3} \quad (9-61)$$

$$Z = R/W_c^{1/3}, \text{ ft/lb}^{1/3} = R_0/W_0^{1/3} \quad (9-62)$$

or  $R_{\text{exp}} = R_0 \lambda$   
where

$R_{\text{exp}}$  = distance from the center of the explosion source to the point of interest, ft

$m_{\text{TNT}}$  = mass of TNT, lb

$Z_{\text{TNT}}$  = scaled distance to the point of interest,  $\text{ft/lb}^{1/3}$

$W_c$  = explosive charge weight, lb

$\lambda$  = yield factor  $= (W/W_0)^{1/3}$

Subscript, 0, refers to reference value.

When accounting for effective charge weight at ground reflection the conversion is given as [19]:

$$W_e = 1.8W_c \quad (9-63)$$

where  $W_e$  = effective charge weight in pounds of TNT for estimating surface burst effects with free air.

It is assumed that the energy released is proportional to the mass of a specific explosive [51].

At times it is necessary to have a feel for overpressure as it relates to shock front velocity [19] (Figure 9-60). Note especially

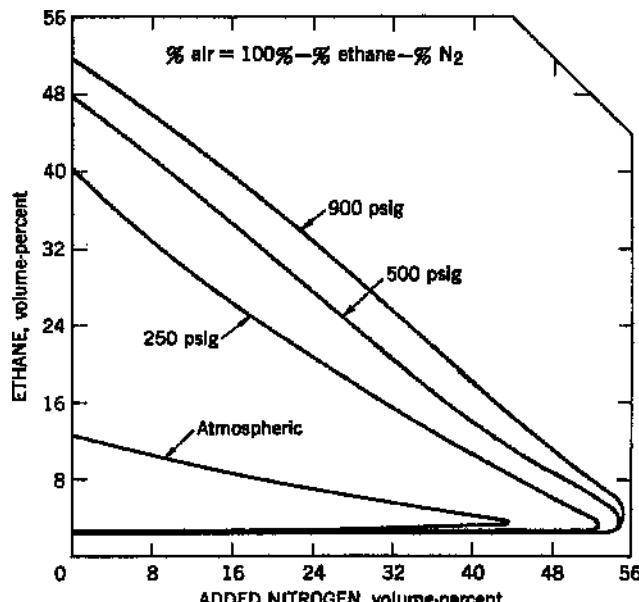


Figure 9-50 Effects of pressure on limits of flammability of ethane-nitrogen-air mixture at 26°C. (By permission from US Bureau of Mines Bulletin 627 [53].)

that for a reasonable detonation velocity the peak overpressure could be in the range of 700–1000 psi and when referenced to Figure 9-60, the extent of industrial damage would be catastrophic. The use of scaled distance is illustrated in [29].

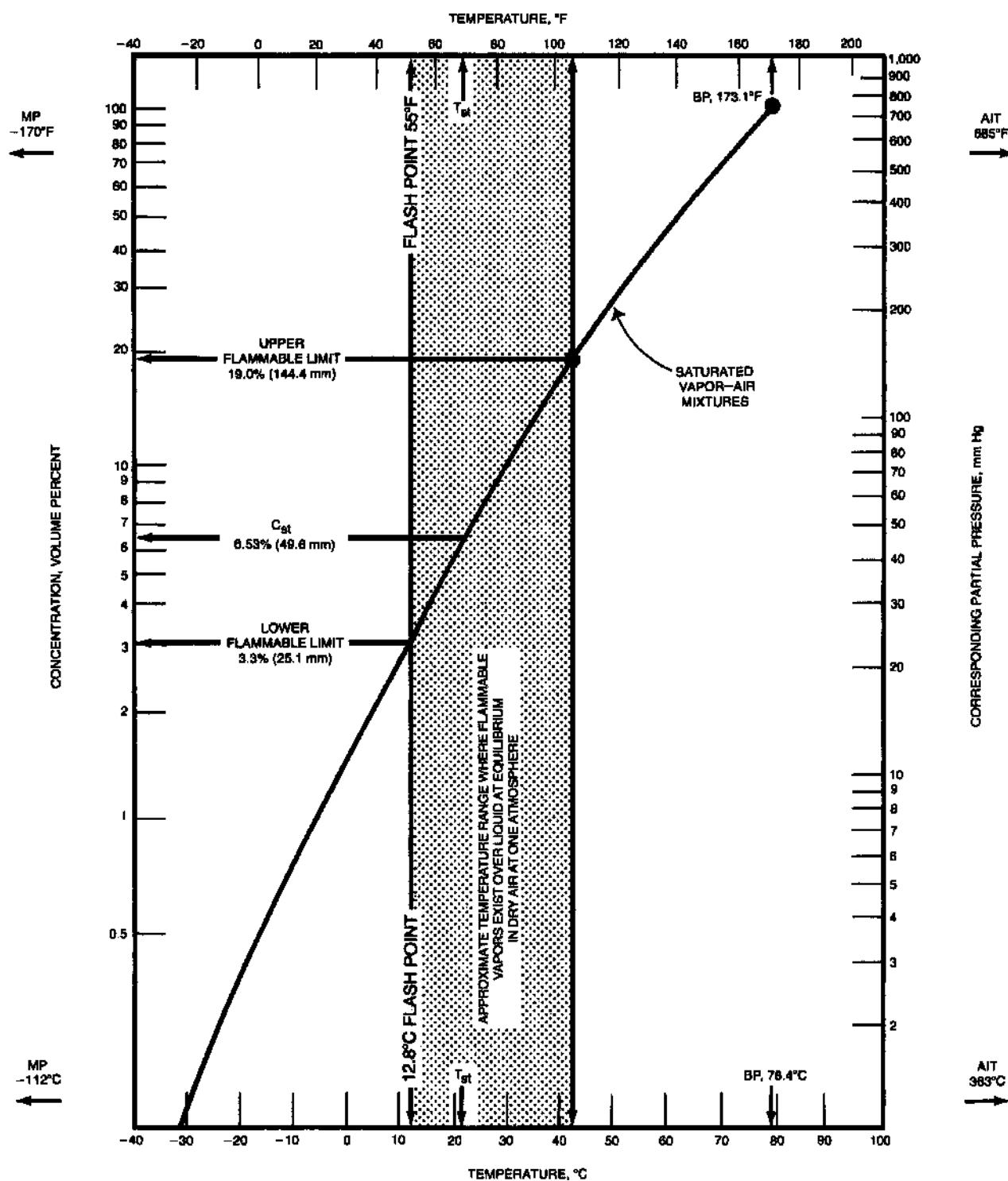
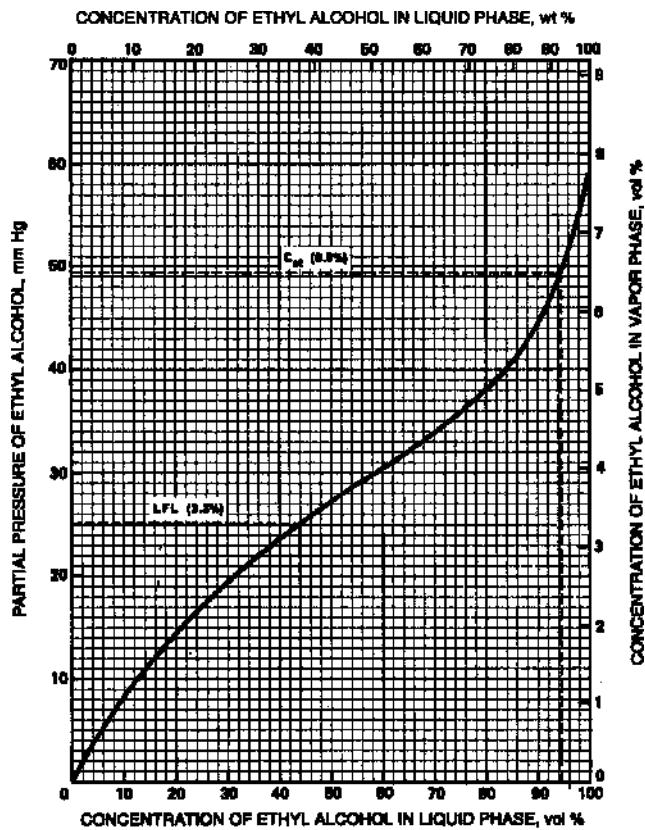


Figure 9-51 Example flammability guide for ethanol. (By permission from Hercules, Inc.)

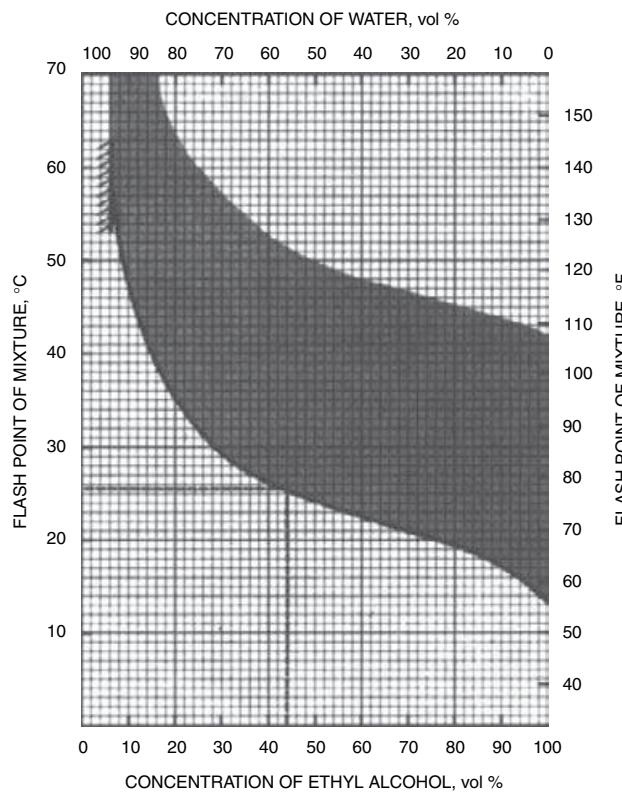
### 9.67 EXPLOSION VENTING FOR GASES/VAPORS (NOT DUSTS)

Unless there is sufficient explosion data for a specific chemical system, including air-mixing or runaway reactions, very few devices can be effective in relieving a confined vessel explosion

other than a carefully designed rupture disk, with a good factor of over capacity. Because detonation explosions initiate and travel so fast (see previous tables/charts), there is a limited chance to relieve the overpressure. Some fast microsecond electronic detectors based on the rate of pressure rise or the rate of temperature rise can be and have been used to anticipate runaway reactions



**Figure 9-52a** Vapor-liquid data for solutions of ethyl alcohol relating to mixture flash points. (By permission from Hercules, Inc.)



**Figure 9-52b** Flash points of ethyl alcohol–water mixtures as a function of liquid phase concentration at one atmosphere total pressure. (By permission from Hercules, Inc.)

#### EXAMPLE 9-18 Blast Scaling

Compare two different explosive charge weights of the same material. For an observed overpressure of 40 psi from a specific charge using the scaling equation above, the scaled distance is  $Z = 5 \text{ ft/lb}^{1/3}$ . What is the distance for an overpressure of 40 psi with a charge of 500 lb?

$$Z = 5 = 5/(1 \text{ lb})^{1/3} = x/(500 \text{ lb})^{1/3}$$

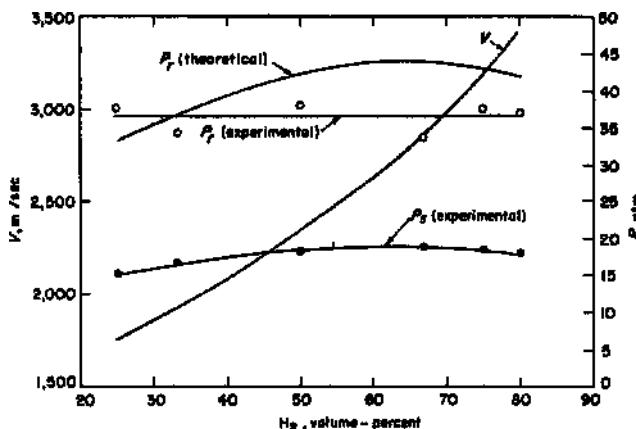
$$x = (500)^{1/3}(5) = 7.93(5) = 39.6 \text{ ft}$$

The distance in which a 500 lb charge will develop a 40 psi overpressure is approximately 40 ft.

and thus create an “early” anticipated release or opening of the relieving device. When such explosive conditions are anticipated, special studies such as the DIERs technology [22, 13] can be justified to provide information for a safe response and control of an explosion. Other methods such as injection of inerting or reaction suppressants have proven to be beneficial (Figures 9-61 and 9-62):

#### 9.68 BLEVES (BOILING LIQUID EXPANDING VAPOR EXPLOSIONS)

This particular type of explosion is less known and understood, but nevertheless is an important type for damage consideration.



**Figure 9-53** Detonation velocity,  $V$ , static pressure,  $P_s$ , and reflected pressure,  $P_r$ , developed by detonation wave propagating through hydrogen–oxygen mixtures in a cylindrical tube at atmospheric pressure at  $18^\circ \text{ C}$ . (By permission from US Bureau of Mines, Bulletin 627 [53].)

This is a type of pressure release explosion and there are several descriptions.

Kirkwood [2] describes BLEVEs referenced to flammable liquids as occurring when a confined liquid is heated above its atmospheric boiling point by an external source of heat or fire and is suddenly released by the rupture of the container due to overpressurization by the expanding liquid. A portion of the superheated

**TABLE 9-26 Conditions of Failure of Peak Overpressure for Selected Structural Components**

Structural Element	Failure	Approximate Incident Blast Overpressure (psi)
Glass windows, large and small	Usually shattering occasional frame failure	0.5–1.0
Corrugated asbestos siding	Shattering	1.0–2.0
Corrugated steel or aluminum paneling	Connection failure, followed by buckling	1.0–2.0
Wood siding panels, standard house Construction	Usually failure occurs at main connections, allowing a whole panel to be blown in	1.0–2.0
Concrete or cinder block wall panels, 8 or 12 in. thick (not reinforced)	Shattering of the wall	2.0–3.0
Brick wall panel, 8 or 12 in. thick (not reinforced)	Shearing and flexure failures	7.0–8.0

By permission from Zabetakis, M.G., US Bureau of Mines, Bul. 627 [53].

#### EXAMPLE 9-19

##### Estimating Explosion Damage

An overpressure after an explosion is noted as 0.5 psi. The calculated scaled distance  $Z$  is  $75 \text{ ft}/(\text{lb})^{1/3}$ . Thus for a one pound charge, windows are broken at a distance of 75 ft. How far will windows be broken for a 500 lb charge?

$$Z = 75 = 75/(1)^{1/3} = x/(500)^{1/3}$$

$x = 7.937(75) = 595 \text{ ft}$ , windows broken for a 500 lb charge.

In general, a reflected shock wave of 55 psi on a human for 400 ms would be just about the tolerance limit [29] (see Table 9-28b). For a more detailed discussion of blast scaling and overpressure, see [51].

liquid immediately flashes to vapor and is ignited by the external heat source (see [61]).

The NFPA [6] contains extensive descriptions of BLEVEs (also see [62]) and describes them in summary as paraphrased here with permission: liquefied gases stored in containers at temperatures above their boiling points at NTP will remain under pressure only as long as the container remains closed to the atmosphere. If the pressure is suddenly released to atmosphere due to failure from metal overstress by external fire or heat, corrosion penetration, or external impact (for examples), the heat stored in the liquid generates very rapid vaporization of a portion of the liquid proportional to the temperature difference between that of the liquid at the instant the container fails and the normal boiling point of the liquid. Often this can generate vapor from about one-third to one-half of the liquid in the container. The liquid vaporization is accompanied by a large liquid to vapor expansion, which provides the energy for propagation of vessel cracks, propulsion of pieces of the container, rapid mixing of the air and vapor resulting in a characteristic fireball upon ignition by the external fire or other source that caused the BLEVE to develop in the first place, with atomization of the remaining cold liquid. Often the cold liquid from the vessel is broken into droplets that can burn as they fly out of the vessel. Often this cold liquid can escape ignition and may be propelled 1.5 mi or more from the initial site. In most BLEVEs, the failure

originates in the vapor space above the liquid, and it is this space that is most subject to external overheating of the metal.

The transportation industry is subject to more impact failures of vessels that lead to BLEVEs. The BLEVE occurs simultaneously with the impact in most recorded cases, but not all. Sometimes there is a delay due to lack of total penetration of the vessel with a hole or crack, and time is needed for the temperature in the container to rise. Failures of the vessel can lead to fireballs of more than several hundred feet in diameter.

The application of water externally to the vapor space of a vessel or application of insulation can often protect against BLEVEs.

A relief valve will not usually handle the vapor generated because its set pressure is usually higher than the boiling point pressure created by the hole or crack in the vessel; therefore, it will not relieve at the lower pressure.

Lees [51] points out that the effects of a BLEVE depend on whether the liquid in the vessel is flammable. The initial explosion may generate a blast wave and fragments from the vessel. For a flammable material, the conditions described in [6] above may result, and even a vapor cloud explosion (VCE) may result.

#### 9.69 LIQUID MIST EXPLOSIONS

When a flammable liquid is sprayed as fine droplets into the air, a flammable mixture can result, which may burn or explode. The mist or spray may be formed by condensation of saturated vapors or by mechanical means [51]. As the particle sizes of the liquid become greater than 0.01 mm diameter, the lower flammability limit of the material becomes above 0.01 mm and the LEL is about the same as the vapor. Mechanical engine crankcase explosions of oil mist in air are hazardous, and current practice is to apply explosion-relief valves to the crankcase.

Compressed air system explosions in engines, pipelines, separators, and so on are characteristic of this same type of mist explosion.

#### 9.70 RELIEF SIZING: EXPLOSIONS OF GASES AND VAPORS

The NFPA has extensive codes that relate to fire and explosion prevention and protection for all major industries and/or

TABLE 9-27 Typical Damage Caused by Overpressure Effects of an Explosion

Equipment	Overpressure (psi)																									
	85	10	15	20	25	30	35	40	45	50	55	60	65	70	75	80	85	90	95	100	120	140	160	180	200	220
Control house steel roof	a	c	d				n																			
Control house concrete roof	a	e	p	d			n																			
Cooling tower	b		f			o																				
Tank cone roof	d			k								u														
Instrument caboose	a		m					t																		
Fired heater	g	i					t																			
Reactor chemical	a			i				p				t														
Filter	h			f								v	t													
Regenerator	i				p			t																		
Tank floating roof	k				u																d					
Reactor cracking	i											t														
Pipe supports	p			s	o																					
Utilities electric transformer			h				i				t															
Electric motor			h					i			t										v					
Blower			q								t											v				
Fractionation column				r	t																					
Pressure vessel horizontal					pl						t															
Steam turbine						i					t			m	s						v					
Heat exchanger					i	t																				
Tank sphere						l					t															
Pressure vessel vertical																	i	t								
Pump															i	v										
	.89	.87	.81	.74	.57	.42	.34	.29	.23	.19	.15	.11	.08	.06	.04	.03	.02	.01	.00	.60	.83	.97	1.1	1.2	1.4	1.5
	Overpressure (bar)																									

- a. Windows and gauges break
- b. Louvres fall at 0.3–0.5 psi
- c. Switchgear damaged by roof collapse
- d. Roof collapses
- e. Instruments damaged
- f. Inner parts damaged
- g. Brick cracks
- h. Debris-missile damage occurs
- i. Unit moves and pipes break
- j. Bracing fails
- k. Unit uplifts (half-filled)
- l. Power lines severed
- m. Controls damaged
- n. Block walls fail
- o. Frame collapses
- p. Frame deforms
- q. Case damaged
- r. Frame cracks
- s. Piping breaks
- t. Unit overturns or is destroyed
- u. Unit uplifts (0.9 filled)
- v. Unit moves on foundation

By permission from Wells, G. L., *Safety in Process Plant Design*. George Godwin Ltd, London: John Wiley & Sons, NY, 1980 [60].

**TABLE 9-28a Selected Overpressure Failure Situation**

Damage Limits	Average Overpressure (psi)
Crater	280
Probable Total Destruction	$\geq 10$
Limit Serious Structural Damage	2.3
Limit Earth Wave Damage	1.2
Limit Minor Structural Damage	0.4
Missile Limit	0.3
Typical Glass Failure	0.15
Limit Glass Breakage	0.006

By permission from Stull [29] Dow Chemical Co. and AIChE, Monograph Series, No. 10, Vol. 73 (1977).

**TABLE 9-28b Physiological Effects of Blast Overpressure**

Physiological Effects of Blast Pressures	Peak Overpressure (psi)						
Knock Personnel Down	1						
Eardrum Rupture	<table> <tr> <td>{ Threshold</td><td>5</td></tr> <tr> <td>50%</td><td>15</td></tr> </table>	{ Threshold	5	50%	15		
{ Threshold	5						
50%	15						
Lung Damage	<table> <tr> <td>{ Threshold</td><td>30–40</td></tr> <tr> <td>Severe</td><td><math>\geq 80</math></td></tr> </table>	{ Threshold	30–40	Severe	$\geq 80$		
{ Threshold	30–40						
Severe	$\geq 80$						
Lethality	<table> <tr> <td>{ Threshold</td><td>100–120</td></tr> <tr> <td>50%</td><td>130–180</td></tr> <tr> <td>Near 100%</td><td>200–250</td></tr> </table>	{ Threshold	100–120	50%	130–180	Near 100%	200–250
{ Threshold	100–120						
50%	130–180						
Near 100%	200–250						

By permission from Stull [29] Dow Chemical Co. and AIChE, Monograph Series, No. 10, Vol. 73 (1977).

occupations, for example [5, 8, 10, 12, and 48]. The NFPA code No. 69 [12] relates to explosion prevention systems and the “design, construction, operation, maintenance and testing of systems for the prevention of deflagration explosions” and is a valuable document for designers. One significant requirement of this code relates to establishing the design pressure of an ASME code pressure vessel if the vessel contains a deflagration and its explosion pressure is internal.

From NFPA-69 [12], extracted by permission, design Pressure

$$P_r = \frac{1.5[R(P_i + 14.7) - 14.7]}{F_u}, \text{ psig} \quad (9-64)$$

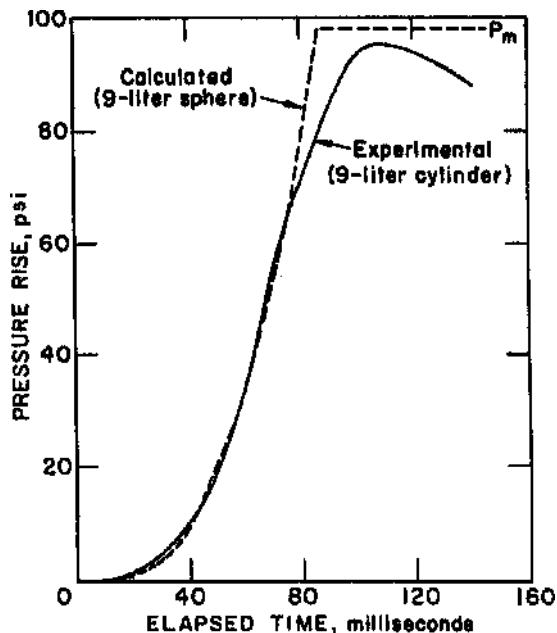
$$P_d = \frac{1.5[R(P_i + 14.7) - 14.7]}{F_y}, \text{ psig} \quad (9-65)$$

where

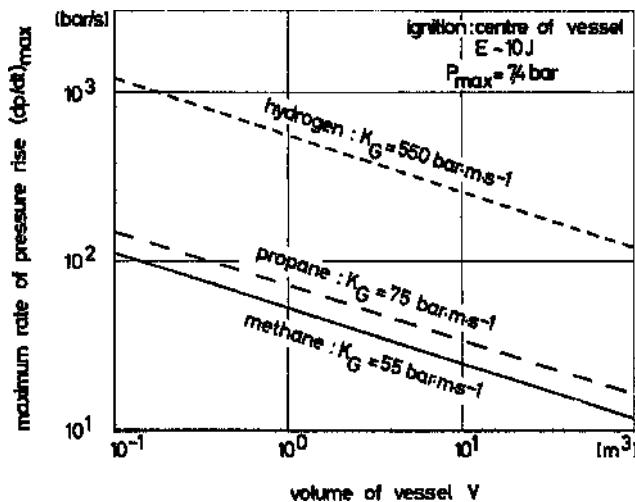
$P_r$  = the design pressure to prevent (vessel) rupture due to internal deflagration, psig

$P_d$  = the design pressure to prevent deformation due to internal deflagration, psig

$P_i$  = the maximum initial pressure at which the combustible atmosphere exists, psig



**Figure 9-54** Pressure produced by ignition of a 9.6 vol % methane-air mixture in a 9 L cylinder (experimental). (By permission from US Bureau of Mines, Bulletin 627 [53].)



**Figure 9-55** Influence of vessel volume on violence of explosion of flammable gases. Ignition at zero turbulence. (By permission from Bartknecht, W., *Explosions*, 2nd ed. (1980), Springer-Verlag [30].)

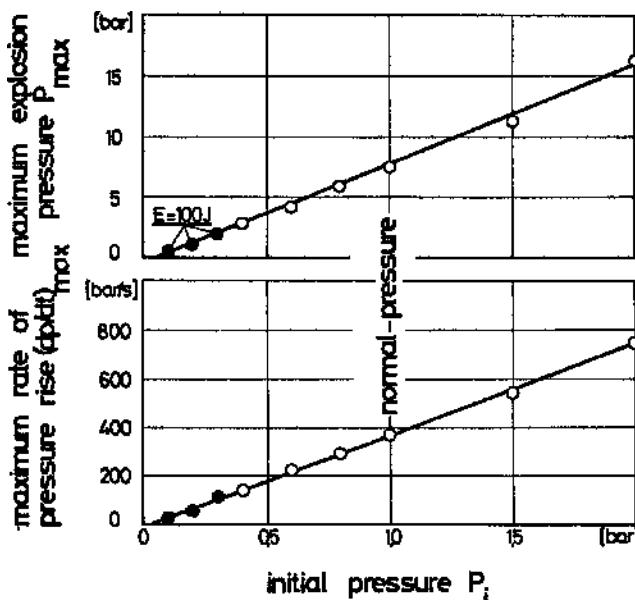
$R$  = the ratio of the maximum deflagration pressure to the maximum initial pressure, as described in code Par 5-3.3.1

$F_u$  = the ratio of the ultimate stress of the vessel to the allowable stress of the vessel

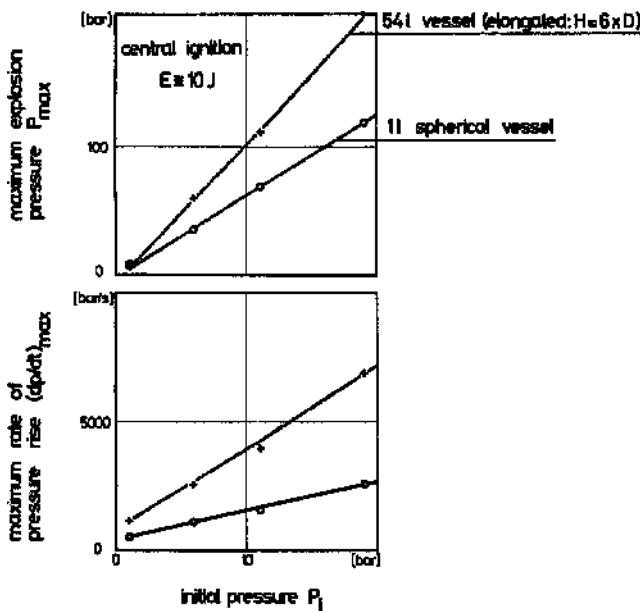
$F_y$  = the ratio of the yield stress of the vessel to the allowable stress of the vessel

Note: 1 psi = 6.89 kPa.

**5-3.3.1** The dimensionless ratio  $R$  is the ratio of the maximum deflagration pressure, in absolute pressure units, to the maximum



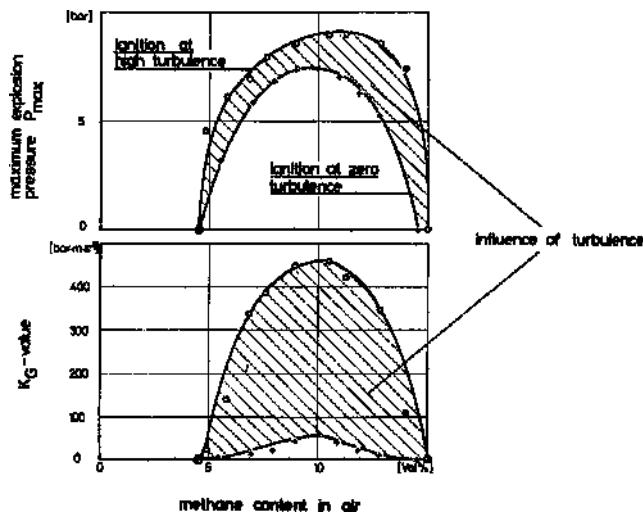
**Figure 9-56** Influence of initial pressure on explosion data of propane, 7 L vessel E = 10J. (By permission from Bartknecht, W., *Explosions*, 2nd ed. (1980). Springer-Verlag [30].)



**Figure 9-57** Influence on vessel shape on the explosion data of methane. (By permission from Bartknecht, W., *Explosions*, 2nd ed. (1980). Springer-Verlag [30].)

initial pressure, in consistent absolute pressure units. For gas/air mixtures,  $R$  shall be taken as 9.0; for dust/air mixtures,  $R$  shall be taken as 10.0.

**Exception:** A different value of  $R$  may be used if appropriate test data or calculations are available to confirm its suitability.



**Figure 9-58** Influence of turbulence on the explosion data of methane at 10J. (By permission from Bartknecht, W. *Explosions*, 2nd ed. (1980), Springer-Verlag [30].)

**TABLE 9-29 TNT Equivalence Factors for Chemical Explosives**

Explosive	$e_t$ (TNT Equivalent)
Amatol 60/40 (60% ammonium nitrate, 40% TNT)	0.586
Baronal (50% barium nitrate, 35% TNT, 15% aluminum)	1.051
Comp B (60% RDX, 40% TNT)	1.148
C-4 (91% RDX, 9% plasticizer)	1.078
Explosive D (ammonium picrate)	0.740
H-6 (45% RDX, 30% TNT, 20% Al, 5% D-2 wax)	0.854
HBX-1 (40% RDX, 38% TNT, 17% Al, 5% D-2 wax)	0.851
HMX	1.256
Lead Azide	0.340
Lead Styphnate	0.423
Mercury Fulminate	0.895
Nitroglycerine (liquid)	1.481
Nitroguanidine	0.668
Octol, 70/30 (70% HMX, 30% TNT)	0.994
PETN	1.282
Pentolite, 50/50 (50% PETN, 50% TNT)	1.129
Picric Acid	0.926
RDX (Cyclonite)	1.185
Silver Azide	0.419
Tetryl	1.00
TNT	1.00
Torpex (42% RDX, 40% TNT, 18% Al)	1.667
Tritonal (80% TNT, 20% Al)	1.639

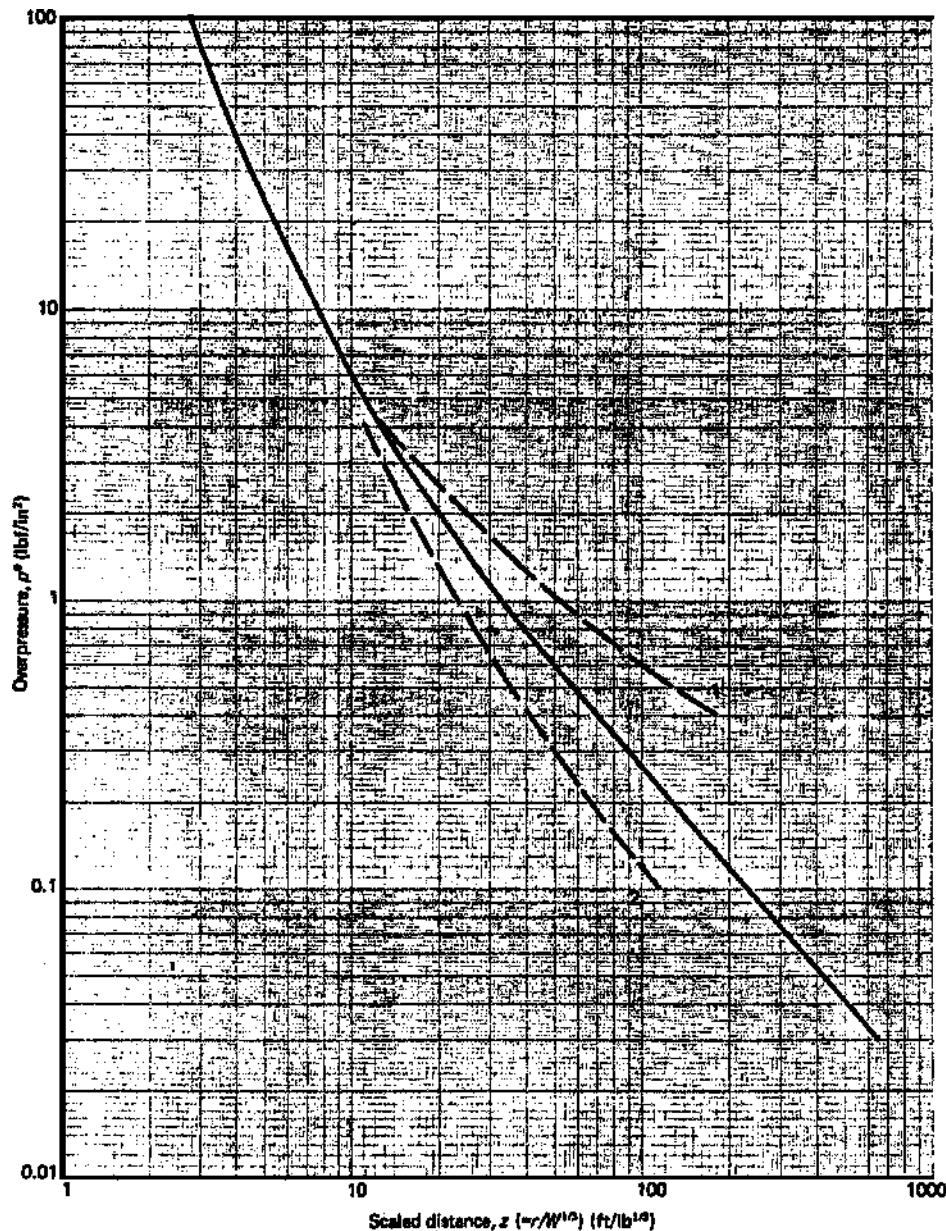
(Refs 3-1 and 3-4 original source)

By permission from US Army Corps of Engineers, Report HNDM-1110-1-2 (1977) [19].

**5-3-3.2** For operating temperatures below 25°C, the value of  $R$  shall be adjusted according to the following formula:

$$R' = R [298 / (273 + T_i)] \quad (9-66)$$

where  $R$  is either 9.0 or 10.0 and  $T_i$  is the operating temperature in °C.



**Figure 9-59** Peak overpressure vs. scaled distance for a blast wave from an explosion of TNT. (By permission from the publishers, Butterworth-Heinemann, Ltd, Lees, F.P. *Loss Prevention in the Process Industries*, Vol. 1, p. 574 [51].)

**5-3.3.3** For vessels fabricated of low carbon steel and low alloy stainless steel  $F_u = 4.0$  and  $F_y = 2.0$ .

**5-3.4** The presence of any pressure-relief device on the system shall not cause the design pressure calculated by 5-3.3 to be reduced.

**5-3.5\*** For systems handling gases and liquids, the maximum initial pressure,  $P_i$ , shall be the maximum pressure at which a combustible mixture can exist, but not higher than the setting of the pressure-relief device plus its accumulation. For systems handling dusts, this maximum initial pressure shall be the maximum possible discharge pressure of the compressor or blower that is suspending or transporting the material or the setting of the pressure-relief device on the vessel being protected plus its accumulation, whichever

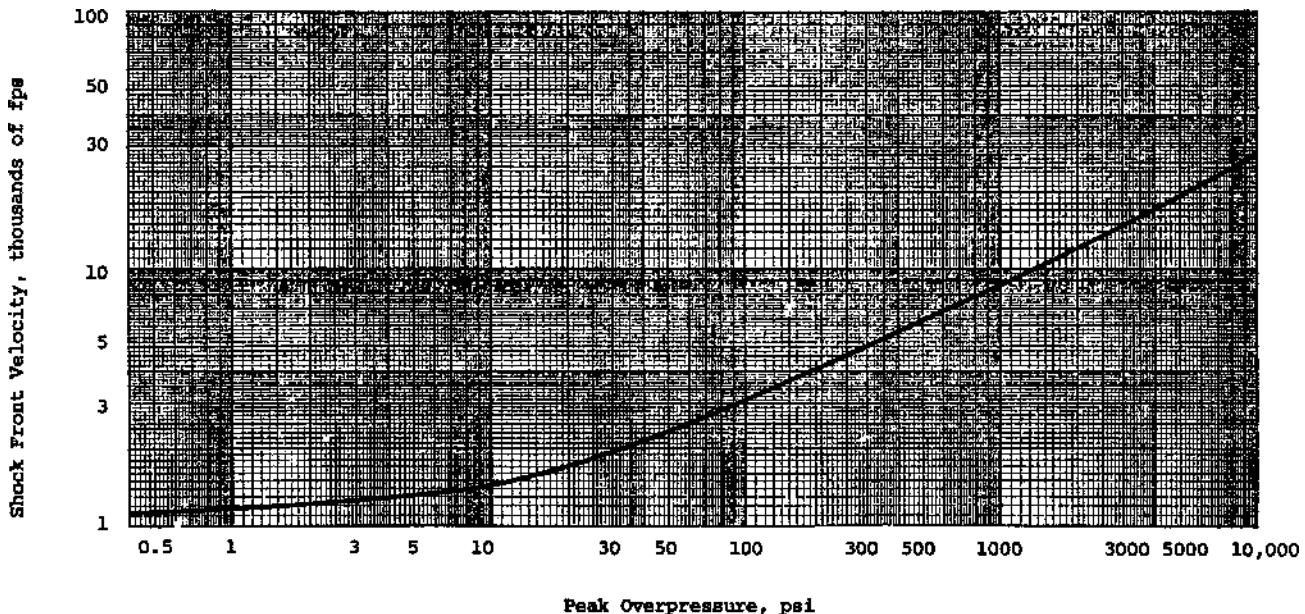
is greater. For gravity, discharge of dusts, the maximum initial pressure shall be taken as 0.0 psig (0.0 kPa gage).

**5-3.6** For systems operating under vacuum, the maximum initial pressure shall be taken as no less than atmospheric pressure (0.0 psig or 0.0 kPa gage).

**5-3.7** The vessel design pressure shall be based on the wall thickness of the vessel, neglecting any allowance for corrosion or erosion.

**5-3.8** The design must take into consideration the minimum operating temperature at which a deflagration may occur. This minimum temperature must be compared with the temperature characteristics of the material of construction of the vessel to ensure that brittle fracture will not result from a deflagration.

*Note:* \* refers to Appendix A of the NFPA code.



**Figure 9-60** Shock front velocity as a function of peak overpressure at sea level. (By permission from Report HNDM-1110-1-2, US Army Corps of Engineers, 1977, Huntsville, AL [19].)

The NFPA-No. 68 [10] presents a design procedure for venting deflagrations within an enclosure in order to minimize the structural and mechanical damage. This deflagration may result from ignition of a combustible gas, vapor, mist, or dust, but is not necessarily considered a detonation, because it acts so much more rapidly and with greater force than the deflagration. It does not apply to bulk autoignition of gases or unconfined deflagrations such as open-air or VCEs. Also, it is not applicable to process situations where internal pressure develops from fire external to the vessel (see NFPA 30 [63] Flammable and Combustible Liquids Code) nor to runaway reactions.

Note that the venting design may not necessarily prevent a deflagration, but is intended to relieve the overpressure developed (see [64]).

### 9.71 VENT OR RELIEF AREA CALCULATION [10] FOR VENTING OF DEFLAGRATIONS IN LOW-STRENGTH ENCLOSURES

Low-strength enclosures notwithstanding more than 1.5 psig (not applicable to end of elongated enclosure) are applicable more to rooms, buildings, and certain equipment enclosures (see Figure 9-8r).

$$A = CA_s / (P_{\text{red}})^{1/2}, \text{ ft}^2 \quad (9-67)$$

where

$A_v$  = vent area,  $\text{ft}^2$

$A_s$  = internal strength-containing surface area of enclosure (or suspended ceilings, etc.)  $\text{ft}^2$

$C$  = venting equation constant, Table 9-30, NFPA-68, pp. 68–14.

$P_{\text{red}}$  = maximum internal overpressure that can be withheld by the weakest structural element, psi.

For elongated enclosures, vent area should be applied evenly relative to the longest dimension. Length-to-diameter ratio should not exceed 3 [10].

For other length-to-diameter ratio, refer to [10]. For cross sections other than circular or square, use the hydraulic diameter:

$$4A/P_{\text{er}}$$

where  $A$  is cross-sectional area and  $P_{\text{er}}$  is the perimeter of the cross section.

For vent area limited or restricted to one end of an elongated enclosure (vertical tank, silo, etc.), the venting equation is limited to:

$$L_3 \leq 12A/P_{\text{er}}, \text{ ft} \quad (9-68)$$

where

$L_3$  = longest dimension of the enclosure, ft

$A$  = cross section area,  $\text{ft}^2$

$P_{\text{er}}$  = perimeter of cross section, ft.

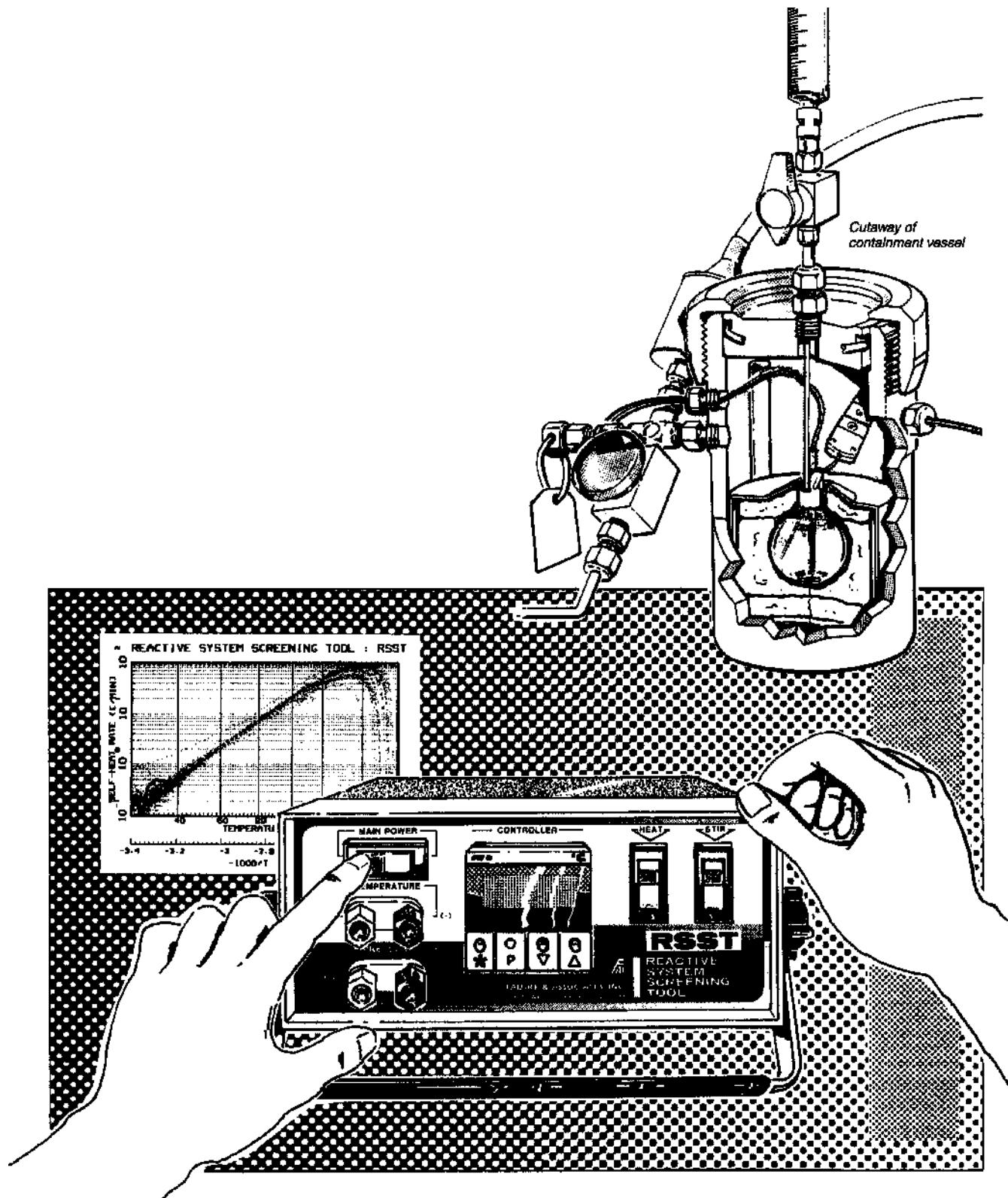
For highly turbulent gas mixing in an enclosure and the vent area is restricted to one end of the elongated enclosure, ratio of length-to-diameter should not exceed 2, or

$$L_3 \leq 8A/P_{\text{er}}, \text{ ft} \quad (9-69)$$

For other conditions, refer to the NFPA-68 code. For the above relations to apply, the constant,  $C$ , should be referred to fuels having the characteristics of one of those in Table 9-30.

Reference [10] presents a thorough discussion of limits to structural components strengths, and these should be observed. Ductile design practices should be used. The maximum allowable design stress should not exceed 25% of the ultimate strength. The strength of the enclosure should exceed the vent relief pressure by at least 0.35 psi.

The vent design must provide at least the area required to satisfy the volume of the enclosure (see NFPA-68) [10].



**Figure 9-61** Reactive System Screening tool (RSST) for evaluating runaway reaction potential. (By permission from Fauske and Associates, Inc.)

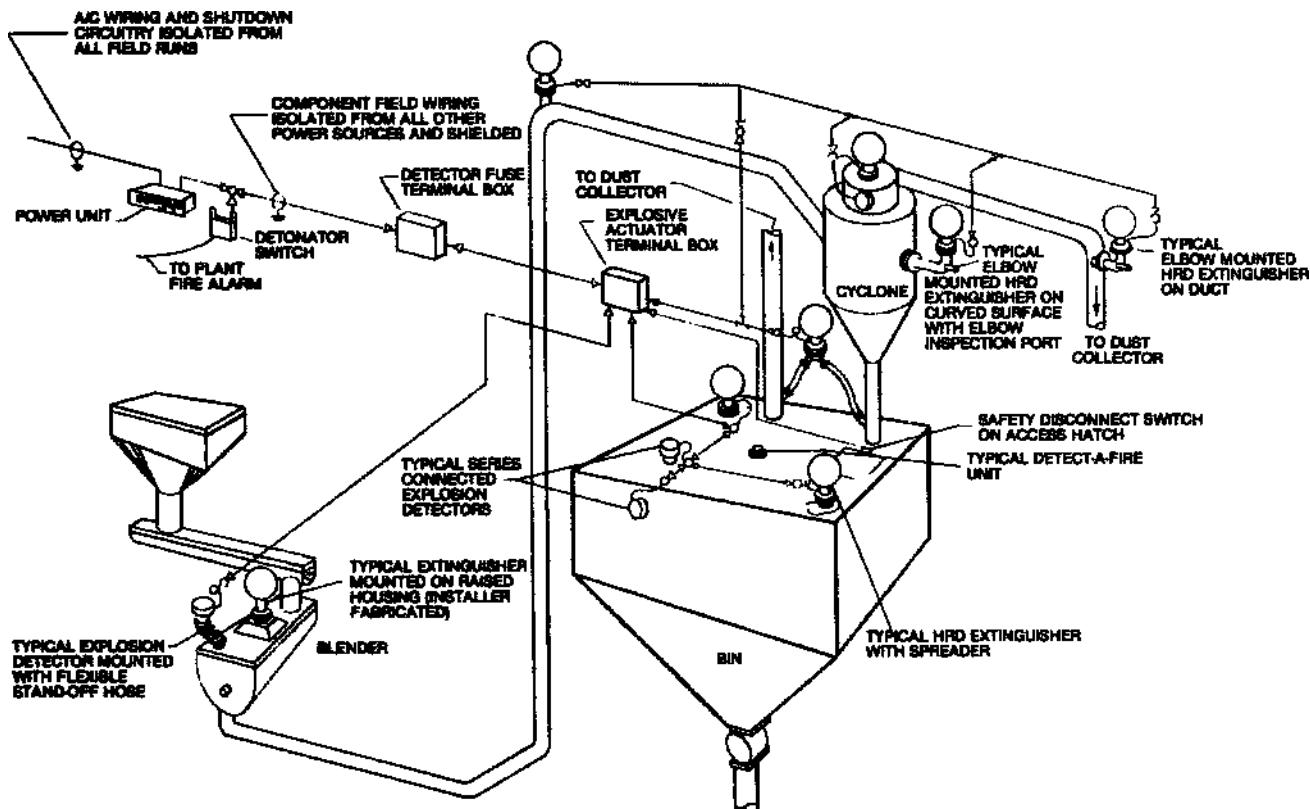


Figure 9-62 Typical Fenwall explosion suppression system. (By permission from Fenwall Safety Systems, Inc.)

TABLE 9-30 Fuel Characteristic Constant for Venting Equation

Fuel	$C(\text{psi})^{1/2}$	$C(\text{kPa})^{1/2}$
Anhydrous Ammonia	0.05	0.13
Methane	0.14	0.37
Gases with fundamental burning velocity less than 1.3 times that of propane	0.17	0.45
St - 1 dusts	0.10	0.26
St - 2 dusts	0.12	0.30
St - 3 dusts	0.20	0.51

Supporting material on explosion protection methods against dust explosions is available for review at NFPA Headquarters, Batterymarch Park, Quincy, MA 02269. Reprinted with permission from NFPA Code 68, *Venting of Deflagrations* (1988), National Fire Protection Association, Quincy, MA 02269 [10].

Note: This reprinted material is not the official position of the National Fire Protection Association on the referenced subject which is represented only by the standard in its entirety.

## 9.72 HIGH-STRENGTH ENCLOSURES FOR DEFLAGRATIONS

This section and Chapters 6 and 7 of the code [10] apply to vessels and equipment capable of withstanding more than 1.5 psig internal pressure. These design procedures do not apply to a detonation that is not believed to be capable of being vented successfully [10].

The vent devices used to relieve the overpressure from the deflagration must be structurally sound, low in weight, and should not fragment to form missiles when the force hits the device.

The discharge from high pressure vessels must be vented out of the building to avoid fires and explosions and overpressure in the building, and the back pressure or pressure loss through the vent duct/pipe must be recognized as affecting the relieving pressure of the vessel. Never locate a vent duct discharge to atmosphere in an area where the discharge might be drawn into (a) fresh air intake to a building or, (b) fresh air intake to a compressor or gas fired engine.

Rupture-disks when properly sized and located on the potentially overpressure vessel have been shown to provide the best protection for a deflagration but not a detonation [30].

**EXAMPLE 9-20****Low Strength Enclosure Venting**

Design to protect a large warehouse containing plastic materials that can emit ethylene and propylene vapors. The dimensions on the rectangular building with a flat roof are as follows.

100 ft long  $\times$  50 ft wide  $\times$  20 ft tall

The building design has been selected as good for 0.4 psi overpressure. See Table 9-26 for glass window shattering. Use known design figures when available.

Areas are

$$\begin{array}{lll}
 \text{Floor} & = 100 \times 50 & = 5,000 \text{ ft}^2 \\
 \text{Roof} & = \text{same} & = 5,000 \text{ ft}^2 \\
 2 \text{ ends} & = 2 \times 50 \times 20 & = 2,000 \text{ ft}^2 \\
 2 \text{ sides} & = 2 \times 100 \times 20 & = 4,000 \text{ ft}^2 \\
 & & \hline
 & & = 16,000 \text{ ft}^2
 \end{array}$$

Vent area, given by Eq. (9-67)

$$A_v = CA_s / (P_{\text{red}})^{1/2}$$

From Table 9-30:

$$\begin{aligned}
 C &= 0.17(\text{psi})^{1/3} \\
 P_{\text{red}} &= 0.40 = 0.40 \text{ psi} \\
 A_v &= 0.17(16,000) / (0.40)^{1/2} = 4300 \text{ ft}^2.
 \end{aligned}$$

This will require roof bursting panels, side wall bursting panel, or an end panel that could be hinged to blow out. The panel relieving pressure should be set for 0.40 psi  $- 0.35 \text{ psi} = 0.05 \text{ psi}$  to burst or relieve per code.

### 9.73 DETERMINATION OF RELIEF AREAS FOR DEFLAGRATIONS OF GASES/VAPORS/MISTS IN HIGH-STRENGTH ENCLOSURES

The nomographs of Figures 9-63a-d [10] were developed by Bartknecht [30] for the conditions of:

- no turbulence in vessel at time of ignition
- low ignition energy of 10 J or less
- atmospheric pressure.

To utilize the charts in Figures 9-63a-d, enter volume, read up to selected  $P_{\text{red}}$  value and across to vent pressure  $P_{\text{stat}}$ , and down to vent area required.

In order to calculate the same area results as the above noted charts for methane, propane, coke gas, and hydrogen, the following equation is presented by NFPA-68 [10]:

$$A_v = a(V)^b (e^{c(P_{\text{sat}})}) (P_{\text{red}})^d \quad (9-70)$$

where

$A_v$  = required vent area,  $\text{m}^2$

$V$  = enclosure volume,  $\text{m}^3$

$e$  = 2.718 (base natural log)

$a-d$  = coefficients, see the table below

$P_{\text{red}}$  = maximum pressure developed during venting, barg.

$P_{\text{stat}}$  = vent closure release pressure, barg.

Coefficients\* /exponents [10]

	<b>a</b>	<b>b</b>	<b>c</b>	<b>d</b>
Methane	0.105	0.770	1.230	-0.823
Propane	0.148	0.703	0.942	-0.671
Coke gas	0.150	0.695	1.380	-0.707
Hydrogen	0.279	0.680	0.755	-0.393

\* Do not use for extrapolation beyond the nomographs.

The nomographs (Figures 9-63a-d) apply for vessel/ equipment with length/diameter ratio ( $L/D$ ) of 5 or less; otherwise, a danger of detonation may exist. For equipment with  $L/D > 5$ , refer to NFPA-68, Chapter 8 [10], and NFPA-69 [12].

### FOR SYSTEMS WITHOUT TEST DATA OR LITERATURE DATA

1. When test data for a particular gas-air system is not available, the nomograph for hydrogen air system, Figure 9-63d, can be conservatively used. Reference [10] reports the additional vent area determined in this manner will usually be small.
2. Interpolation of existing nomographs to determine deflagration vent area of different gas-air mixtures:

Area required for new gas or mixture =  $A_{\text{ng}}$

Assume vessel volume =  $V'$

Establish\* maximum allowable value for  $P_{\text{red}} = x$ :

Establish†  $P_{\text{stat}} = y$

Maximum rate of pressure rise,  $K_G$ , for gas in question in a specific vessel =  $z$  bar/sec.

Note:  $K_G$  values are not constant and vary with test conditions.  $K_G$  provides means of comparing maximum rates of pressure rise for various gases, but should only be used for deflagration vent sizing if test data come from test vessels of similar configuration and same type of ignitor and ignition energy [10].

Refer to the propane and hydrogen nomographs (Figures 9-63b and d). The required vent area to protect the vessel specified from each of the charts will be a and b, square meters, respectively.

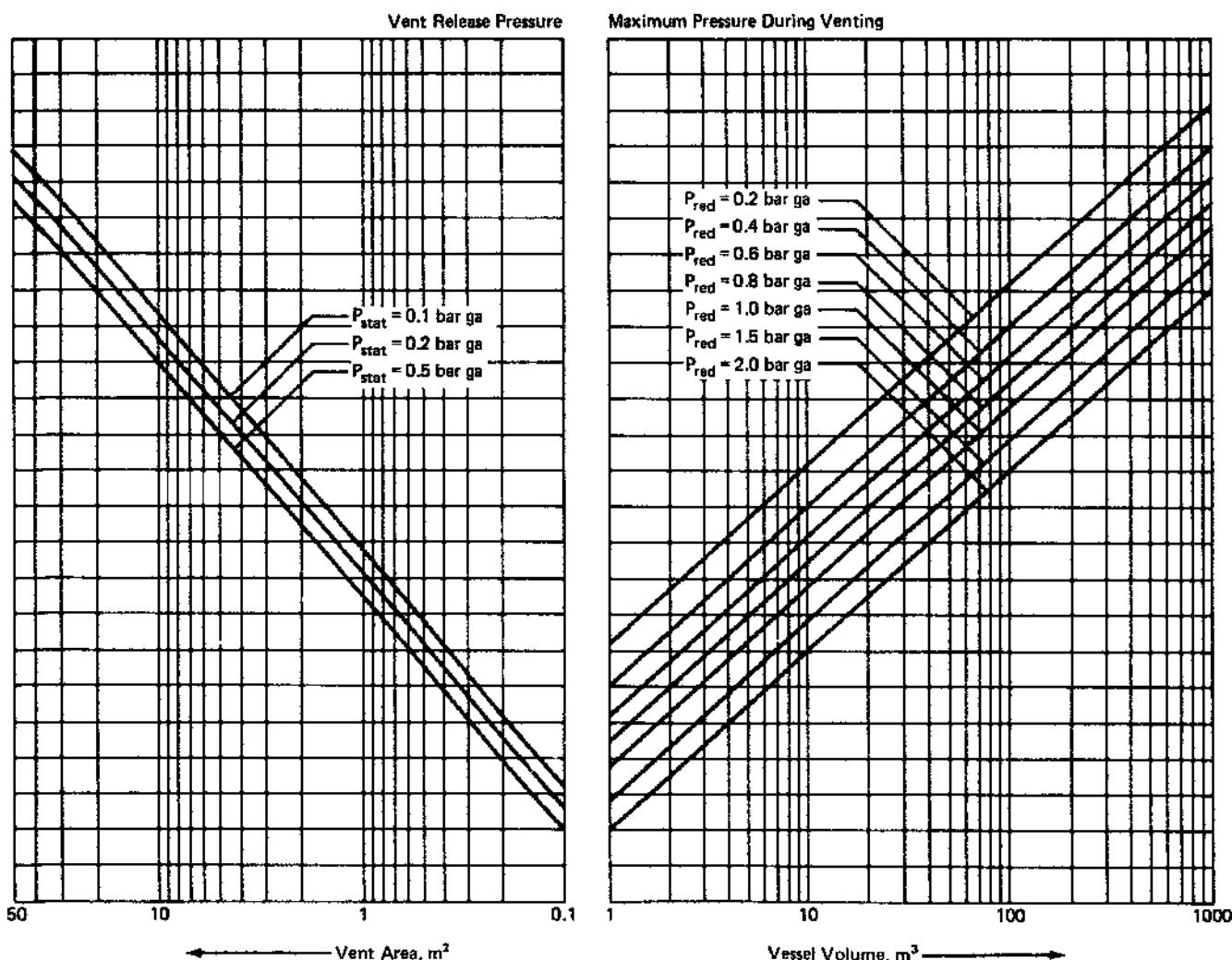
The maximum rates of pressure rise for propane and hydrogen are  $r$  and  $r'$ , respectively, bar/sec in the same vessel. For linear interpolation:

$$\begin{aligned}
 \text{Required vent area for new gas} &= \left[ a + \left[ \frac{(z-r)}{(r'-r)} \right] (b-a) \right] \\
 &= \text{vent area, } \text{m}^2 \quad (9-71)
 \end{aligned}$$

Using the cubic equation presented earlier, Bartknecht [30] developed vessels of different sizes for the same process system; in

\* Must not exceed mechanical strength of vessel; use design or MAWP.

† Must be lower than  $x$  or not more than 10% above MAWP.



**Figure 9-63a** Venting nomograph for methane. (Reprinted with permission from NFPA 68-1988, *Deflagration Venting* (1988), National Fire Protection Association, Quincy, MA 02269). Note: This material is not the complete and official position of the National Fire Protection Association on the referenced subject, which is represented only by the standard in its entirety. Note from E.E. Ludwig that this statement applies to all material referenced for use that originates with the National Fire Protection Association [10].

closed or vented vessels, valid for flammable gases and combustible dusts:

$$A_2 = \left[ \left( \frac{V_1^{1/3}}{V_2^{1/3}} \right) \left( \frac{A_1}{A_2} \right) \right] (V_2) = \left[ \frac{V_1^{1/3}}{V_2^{1/3}} \right] (f)(V_2)(f_1)(V_2) \quad (9-72)$$

where

$A_1$  = initial vessel relief area,  $\text{m}^2$

$A_2$  = second vessel relief area,  $\text{m}^2$

$V$  = volume of vessels 1 and 2, respectively,  $\text{m}^3$

$f$  = specific relief area,  $\text{m}^2/\text{m}^3$ , from  $F = fV$  and  $f_1 V_1^{1/3} = f_2 V_2^{1/3}$  at same  $P_{\text{stat}}$  and  $P_{\text{red}}$ .

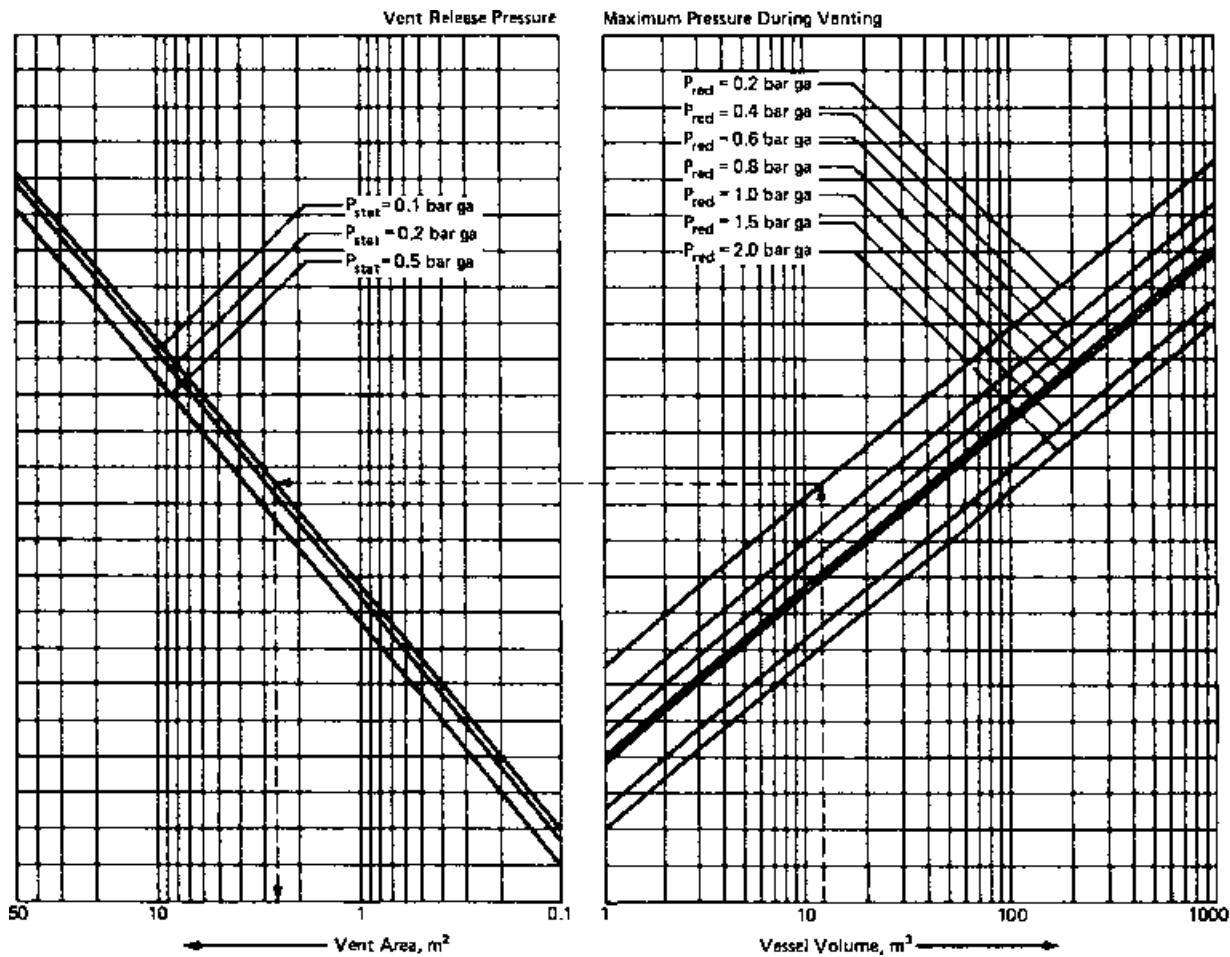
When the relief device relieves, the explosion pressure falls off, but then it increases faster than in the beginning due to the opening of the relief when the flame front is distorted creating an acceleration of the combustion process [30]. Thus, there are two pressure peaks

in the course of the relieving: (1) at the activation of the relief device due to pressure buildup in the vessel, and (2) at the end of the combustion reaction. The first pressure is always greater than the second. The second pressure rise is created by turbulence during the venting process [30].

$K_G$  or  $K_{\text{st}}$  values, which are a measure of the characteristics for the course of an explosion (gas or dust) of vessels roughly cubic in shape, are termed deflagration indices.

A few selected values for zero turbulence are [30] as follows.

$K_G, \text{Bar m/s (average values)}$	
Methane	55
Propane	75
Coke Gas	140
Hydrogen	550



**Figure 9-63b** Venting nomograph for propane. (Reprinted with permission from NFPA 68-1988, *Deflagration Venting* (1988), National Fire Protection Association, Quincy, MA 02269). See note in Figure 9-63a.

The nomograms, Figure 9-63a-d, are based on an operating pressure of 1 bar (absolute), but may be used without correction up to 1.2 bara [30]. There is insufficient pressure data available to recommend using much above normal pressure. When the operating pressure is raised above normal, the reduced explosion pressure will show a proportional increase for a given constant relief venting area [30]. The nomographs are not applicable for detonations.

When systems involving solvent vapor are considered, use the nomogram for propane-air mixtures because most common solvent vapors have maximum explosion pressures of 7.1 to 7.6 bar, and the  $K_G$  falls between 40 and 75 bar m/s (see [30]).

The effects of turbulence must be taken into account when sizing a relief area. For example, the explosion violence of turbulent methane-air mixture is comparable to that of zero turbulence of hydrogen-air mixtures. From the investigations [30], the nomograms from Figure 9-63 can be applied for turbulent gas mixtures under the following conditions [30]:

- For large relief areas:  $P_{red} \leq 1$  bar, valid but influence of ignition energy must be considered.
- For small relief areas,  $P_{red} > 1$  bar, independent of ignition energy. Relief area (vent area,  $A$ ) from nomograms must be increased by a constant factor,  $\Delta A = 0.08V^{2/3}$ .

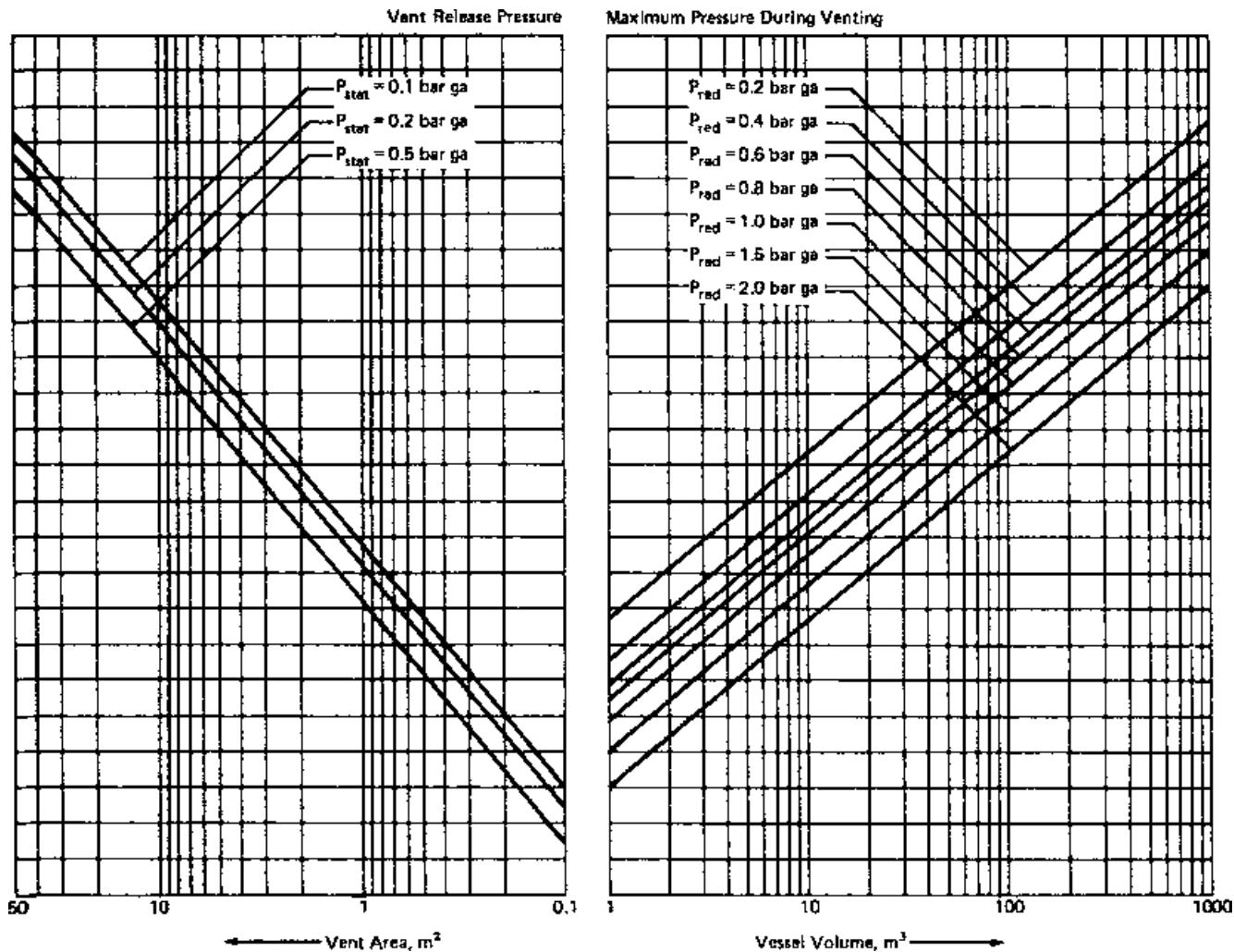
Recommend applying only for methane, propane, and solvent vapors due to insufficient data and for a static activation pressure of the relief device of  $P_{stat} \leq 0.1$  bar (see [30]).

Vessels with internal components may be susceptible to turbulence in the gas mixture, which can lead to detonations, which are not covered by this procedure. Hydrogen is particularly vulnerable to detonations; therefore, for such systems, a specialized expert should be consulted. The details of NFPA-68 should also be consulted as there are many factors that must be recognized (also see [30]).

The range of the nomographs can be extrapolated or extended for a specific vessel volume by cross-plotting as shown in Figure 9-64, limiting to a constant  $P_{red}$  for each chart at varying  $P_{stat}$ . Do not extrapolate below  $P_{stat}$  of 0.05 barg, nor below  $P_{red}$  of 0.1 barg.  $P_{red}$  should not be extrapolated above 2.0 barg;  $P_{stat}$  can be extrapolated but it must always be less than  $P_{red}$  by at least 0.05 bar [10].

## 9.74 DUST EXPLOSIONS

It has been somewhat of a surprise to many engineers that fine dust particles are combustible and will explode for many specific dust types. And in spite of extensive research and development of more than a century to prevent and mitigate dust explosions in the chemical process industries (CPI), this hazard continually



**Figure 9-63c** Venting nomograph for coke gas. (Reprinted with permission from NFPA 68-1988, *Deflagration Venting* (1988), National Fire Protection Association, Quincy, MA 02269). See note in Figure 9-63a.

threatens industries that manufacture, use and /or handle powders and dusts of combustible materials. Dust generation occurs when a bulk material is dropped under free-fall conditions to impact on a stockpile or inside equipment laden with dust and gas. An approach that may be taken to control the dust is to partially or fully enclose the process. An example is the case of conveyor transfer stations, where bulk material is dropped from the end of one conveyor onto the start of another. A very small fraction of the parent bulk material is liberated as fugitive dust, and as a result the total quantity of dust generated may be enormous due to the very large throughput of the parent material. It is essential for the designer to know [65]:

- the volume of dusty air that should be extracted from the enclosure
- the concentration of dust in the air.

Figure 9-65 illustrates examples of dust generation processes.

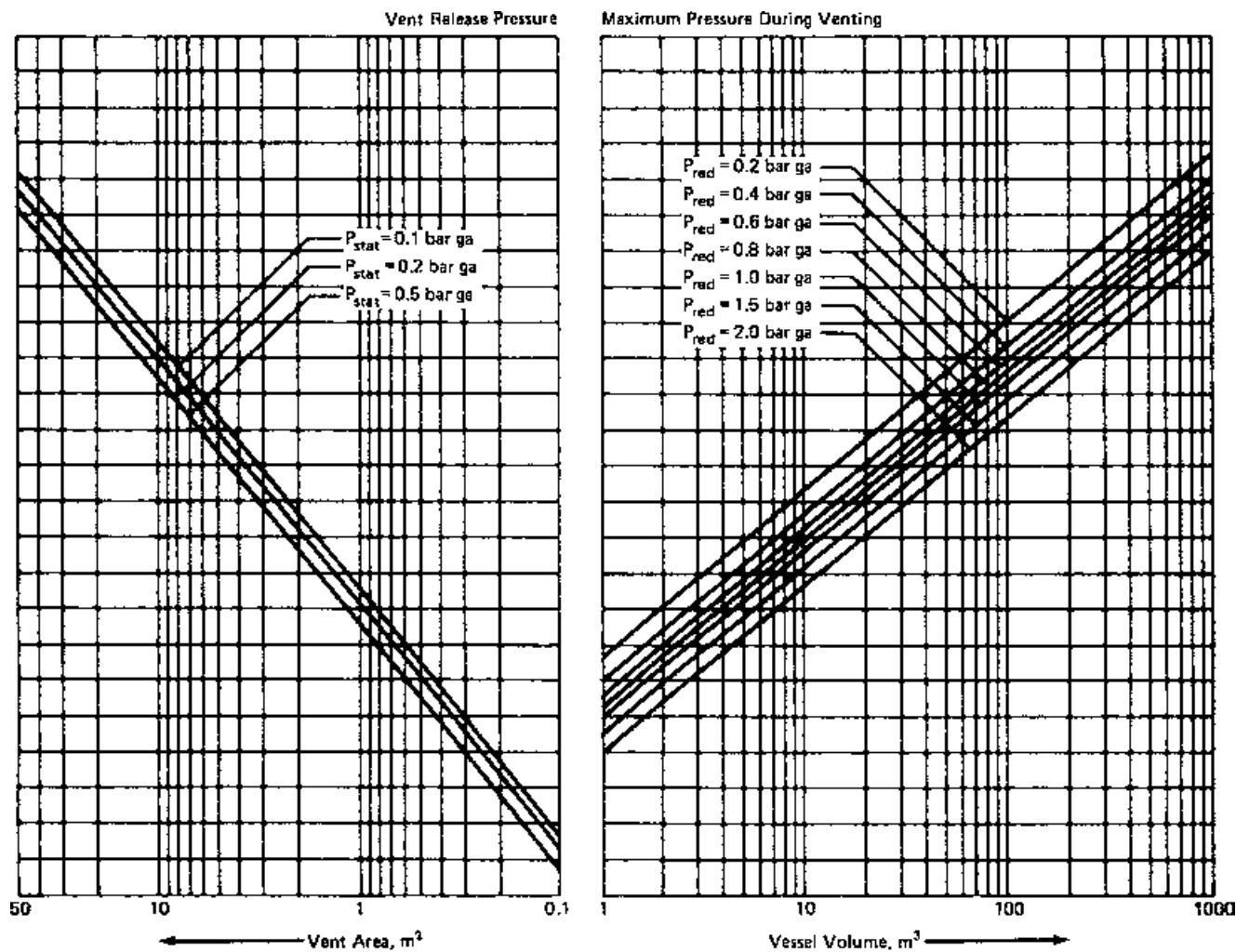
A dust explosion occurs when a finely divided combustible solid is dispersed into the air and subjected to an ignition source or is considered explosive if a dust flame becomes clearly detached from the ignition source. A dust explosion can only occur when the following conditions are satisfied [66].

- The dust must be combustible.
- The dust must be capable of becoming airborne.
- The dust must have a particle size distribution capable of propagating flame.
- The dust concentration must fall within the explosive range.
- The dust suspension must be in contact with an ignition source capable of initiating and sustaining flame propagation.
- The atmosphere into which the dust is suspended must contain sufficient oxygen to support combustion.

## 9.75 DUST EXPLOSION CHARACTERISTICS

Some of the more important factors that influence the initiation and course of a dust explosion are [66] as follows.

**Chemical composition:** During a dust explosion, a chemical reaction takes place between the dust particles and the available oxygen. The rate at which oxygen is consumed and thus the development of the explosion is being related to the chemical nature of the dust.



**Figure 9-63d** Venting nomograph for hydrogen. (Reprinted with permission from NFPA 68-1988, *Deflagration Venting* (1988), National Fire Protection Association, Quincy, MA 02269). See note in Figure 9-63a.

**Particle size:** The particle size of the dust is one of the most important factors to influence the course of the reaction. This is because the reactions take place at the solid surfaces and consequently the surface area available for oxidation, and heat transfer will have a significant effect on the progress of an explosion. The total contact surface area between a solid and air increases as it is reduced in size, which increases the combustion rate (Figure 9-66). Progressive size reduction will result in an increasingly more flammable/explosible dust, and hence, a given material becomes more hazardous and a consequence of an explosion more severe, as the particle size is reduced (Figure 9-67).

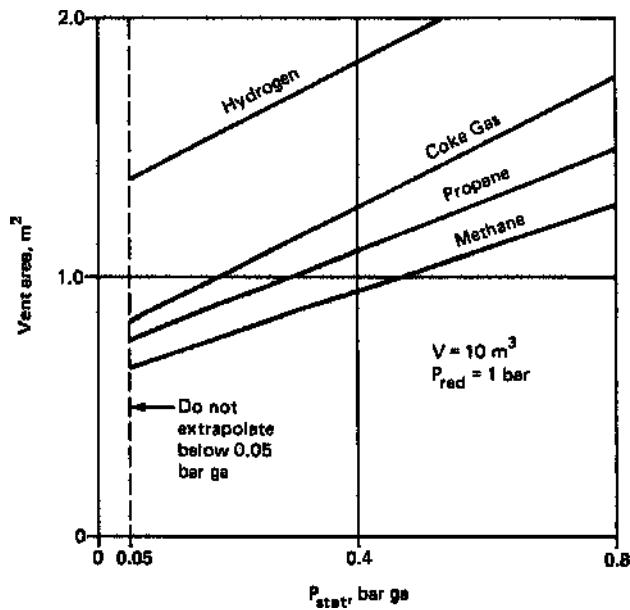
**Dust concentration:** Combustible dust when mixed with air exhibits an explosive range having upper and lower limits. For dusts, the upper explosion limit is not well defined, and difficult to determine. Therefore, the lower explosion limit or minimum explosive concentration is defined by the smallest amount of suspended dust in a given volume capable of propagating flame. This is favored by concentration within the explosive range.

**Moisture content:** Moisture is often present in dusts. In terms of dust explosion, it is generally beneficial since it tends to decrease the explosibility of a dust by increasing the proportion of agglomerates and reducing the effectiveness of an applied ignition source, since water has to be vaporized from the dust before it can ignite.

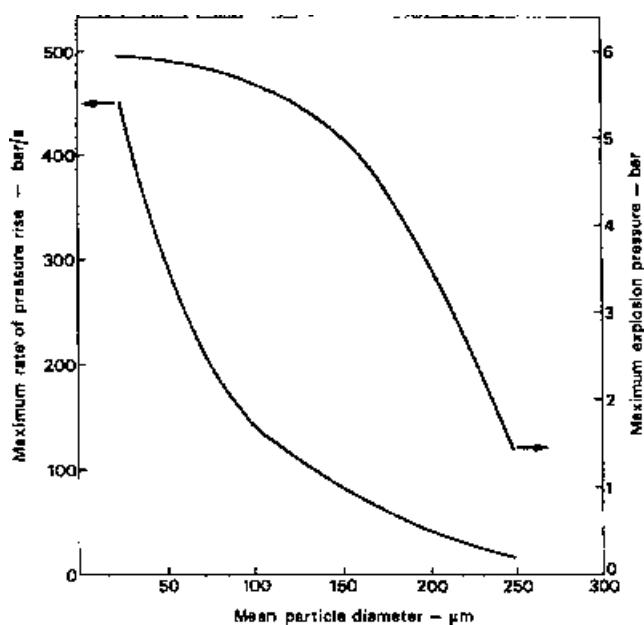
erates and reducing the effectiveness of an applied ignition source, since water has to be vaporized from the dust before it can ignite.

**Ignition:** A combustible dust suspension will only explode if all the above conditions are present and if in addition an ignition source of sufficient energy is present. Dust suspensions exhibit minimum ignition temperatures within the range 300–600°C, and minimum ignition energies which may range from a few millijoules to several joules. However, sources of ignition within these limits are often encountered and arise typically from flames, hot surfaces, electric and electrostatic sparks, welding and cutting operations, friction sparks and spontaneous heating.

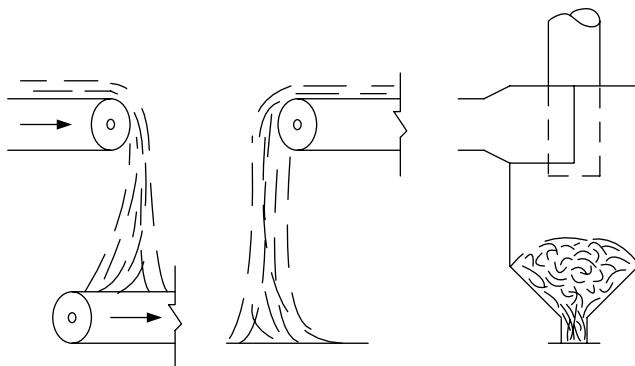
**Turbulence:** Ignition of a highly turbulent dust-air mixture will result in a severe explosion than in a similar mixture ignited under condition of low turbulence. The greater severity of the explosion is a measure of the increase in the rate of pressure rise and not the explosion pressure, which may not be significantly different for the two extreme conditions. The presence of a degree of turbulence is unavoidable especially where air is used to transport dust. It is important that some consideration is given to the likely turbulent conditions when studying explosion protection requirements.



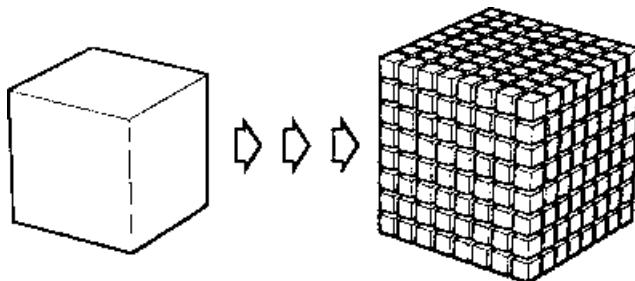
**Figure 9-64** Extrapolation of nomographs for gases. (Reprinted with permission from NFPA 68-1988, *Deflagration Venting* (1988), National Fire Protection Association, Quincy, MA 02269). See note in Figure 9-63a.



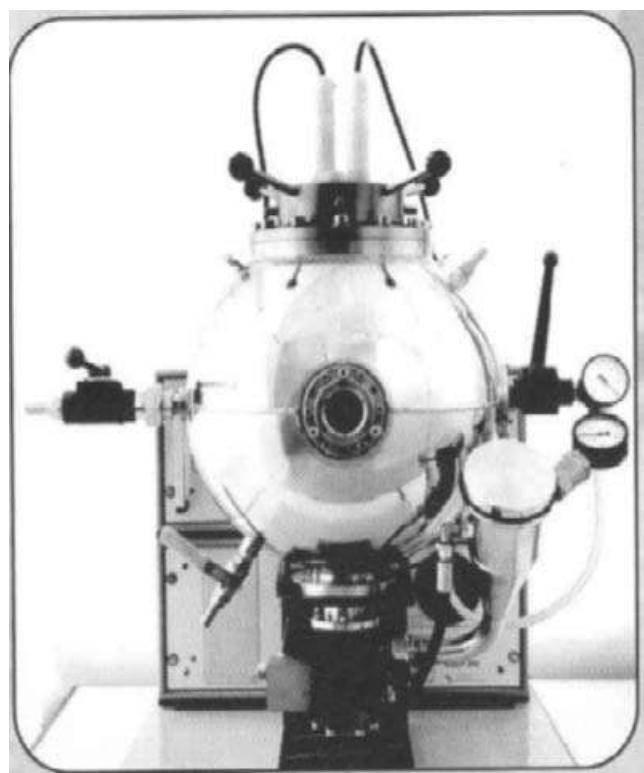
**Figure 9-67** Influence of particle size on the explosion hazard. (From P. Field [66].)



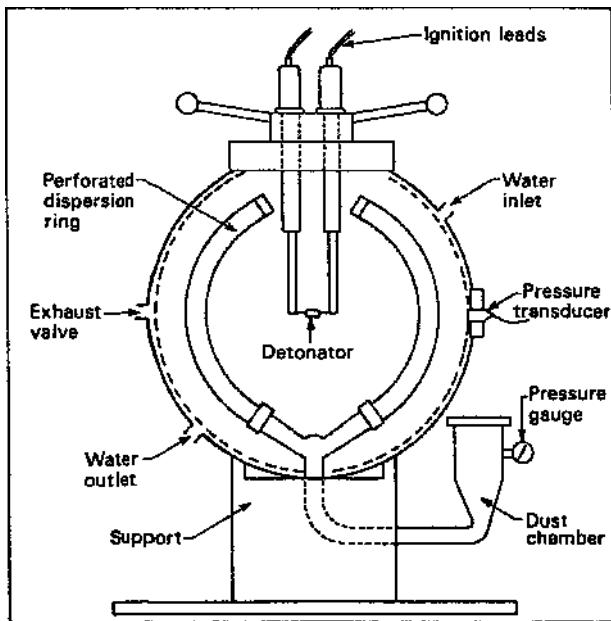
**Figure 9-65** Examples of dust-generation processes involving free fall of bulk material.



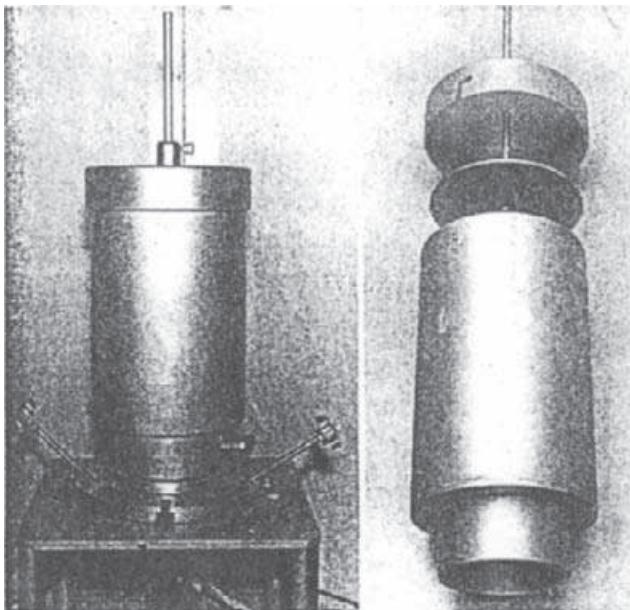
**Figure 9-66** Increase in surface area resulting from particle size reduction. (From P. Field [66].)



**Figure 9-68a(a)** A 20L spherical explosion apparatus to determine maximum explosion pressure and the maximum rate of pressure rise. (From Hazard Evaluation Laboratory, Ltd.)



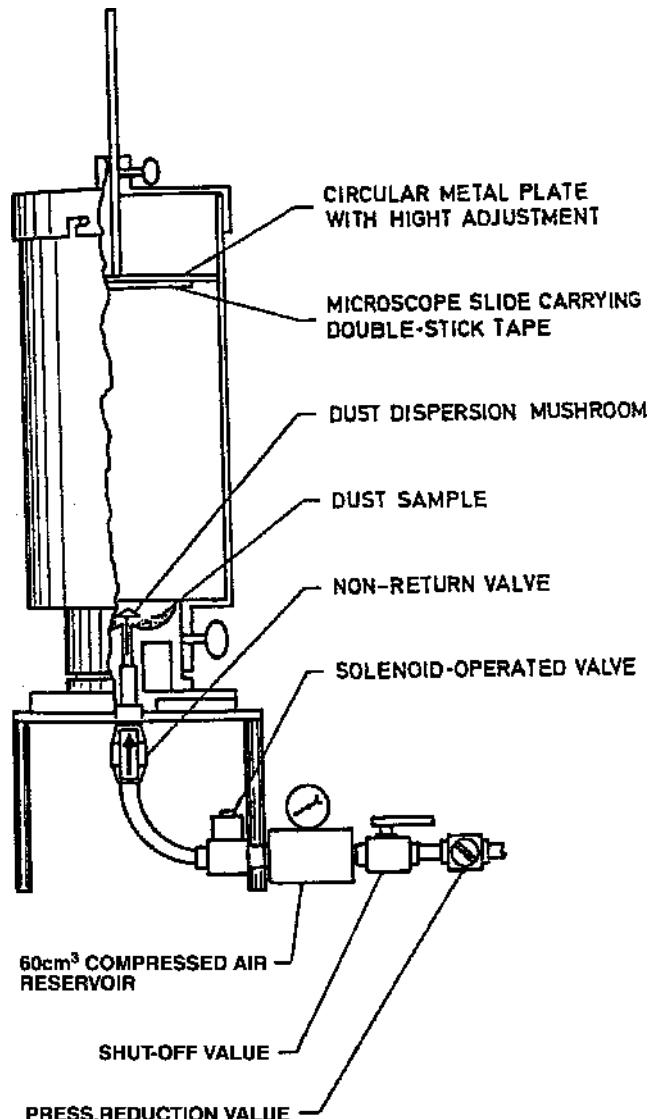
**Figure 9-68a(b)** A Schematic of the 20L spherical explosion apparatus. (From G. Lunn [67].)



**Figure 9-68b** Apparatus for assessing the degree of dispersion in dust clouds in the Hartmann bomb, assembled on the dust dispersion cup of the Hartmann bomb (left) and dismantled (right). (from R.K. Eckhoff [68].)

## 9.76 EVALUATING THE HAZARD

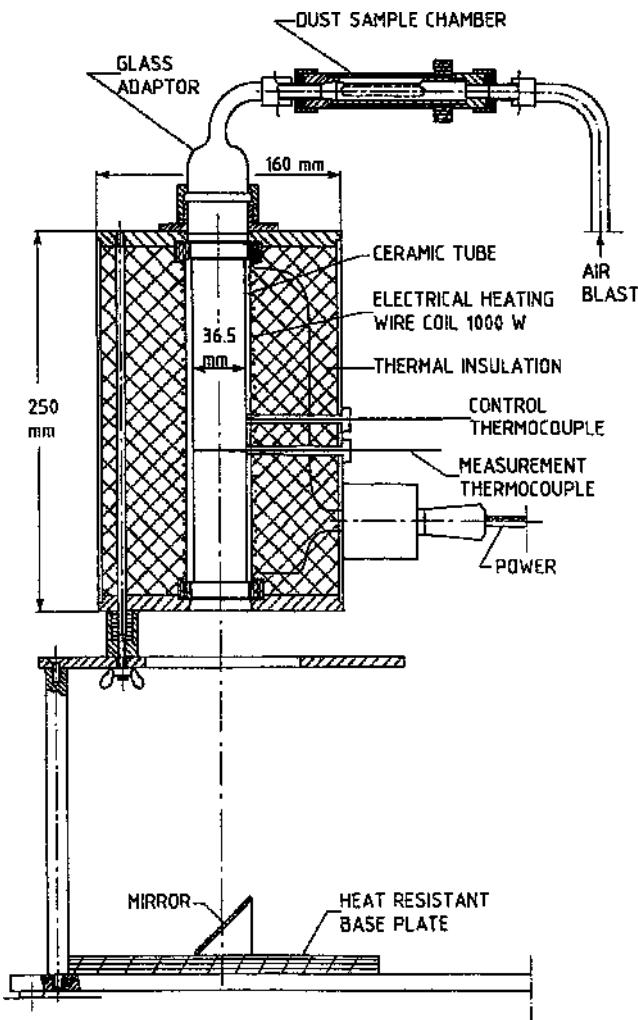
Properly evaluating a potential hazard in a chemical plant involves two steps: material testing and process review. The first stage in the design of explosion prevention and protection system is to determine a quality assessment of the ability of a dust to take part in an explosion. These are explosive tests, which are used to classify dusts into two main groups.



**Figure 9-68c** Apparatus for assessing the degree of dispersion in dust clouds in the 1.2 L Hartmann bomb. (From R.K. Eckhoff [68].)

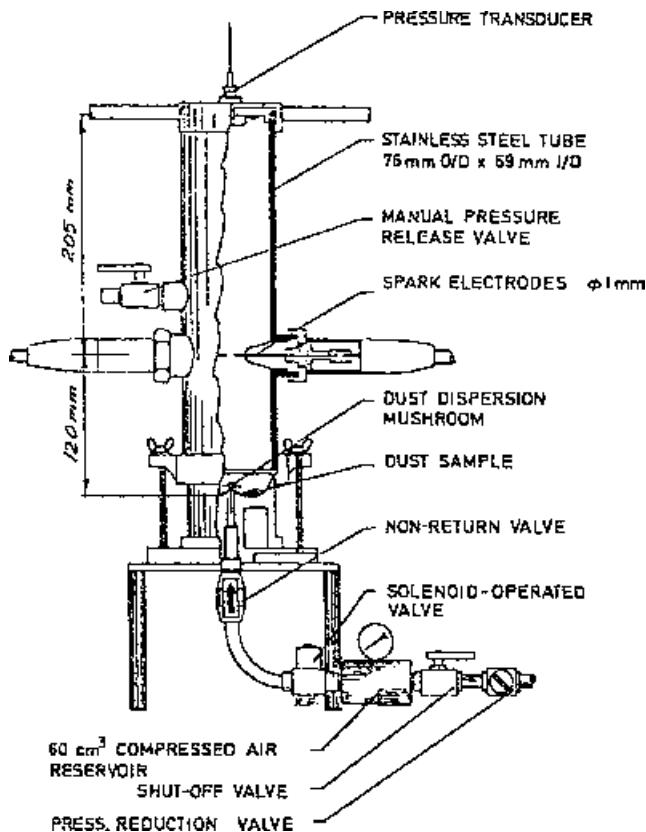
1. Group A – dusts which are able to ignite and propagate a flame.
2. Group B – dusts which do not propagate a flame.

It is essential that the classification relates to the conditions under which the dust will be handled. It will be futile to conduct an explosibility assessment at a room temperature when the process temperature is at several hundred degrees. Further, dusts which can be classified as Group B at room temperature can readily be shown to ignite if the temperature is higher. A series of tests which allows explosion classification under severe conditions are illustrated by Lunn [67]. Figures 9-68a-g show test methods that are employed in classifying dust explosion, and Table 9-31 shows some dust explosibility techniques with their advantages and limitations. Lunn [67] and Eckhoff [68] have provided detailed descriptions of various test methods for classifying dust explosions. The following illustrates the ASTM codes in relation to test methods on a range of dust explosion classification.



**Figure 9-68d** A schematic of the Godbert-Greenwald furnace for determination of the minimum ignition temperature of dust clouds. (From R.K. Eckhoff [68].)

Minimum ignition energy (MIE)	This test is performed as per ASTM E 2019-03, "Standard Test Method for Minimum Ignition Energy of a Dust Cloud in Air". This test method provides a procedure for performing laboratory tests to determine the minimum ignition energy of a dust cloud. The test is conducted in a 1.2-L Hartmann chamber made of clear plastic.
Limiting (minimum) oxygen content (LOC)	This test is performed as per ASTM proposed standard. The test method provides a procedure for performing laboratory tests to determine the lowest oxygen concentration for which the dust-air mixture will propagate a deflagration using a U.S. Bureau of Mines 20-L explosibility test chamber.
Minimum autoignition temperature (MAIT) of dust cloud	This test is performed as per ASTM E 1491-97 (Reapproved 2002), "Standard Test Method for Minimum Autoignition Temperature of Dust Clouds". This test method provides a procedure for performing laboratory tests to determine the minimum autoignition temperature (MAIT) of a dust cloud. This test is conducted in a U.S. Bureau of Mines 1.2-L furnace.



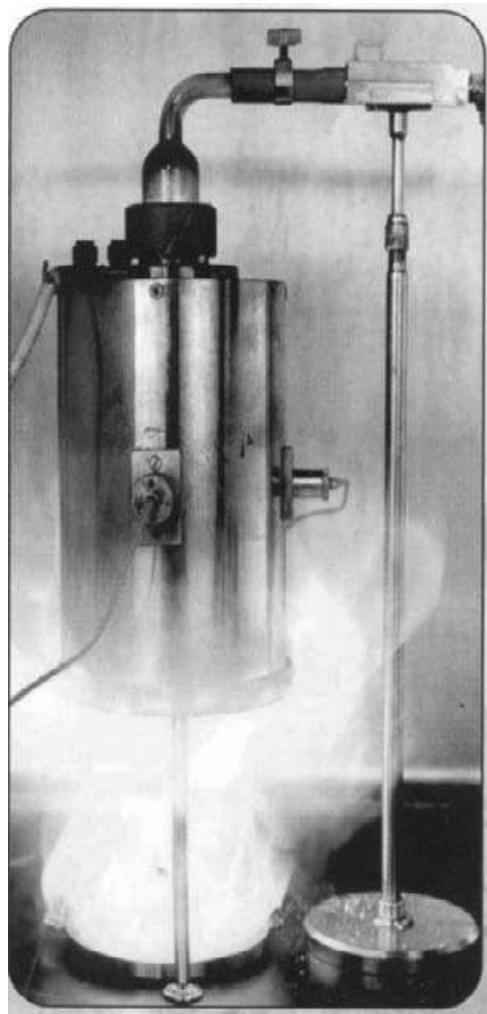
**Figure 9-68e** A schematic of 1.2 L Hartmann bomb to determine pressure development in dust explosions at constant volume. (From R.K. Eckhoff [68].)

Explosion severity test ( $K_{st}$ ,  $P_{max}$ , and  $dP/dt$ )

This testing is performed as per ASTM E 1226-00, "Standard Test Method for Pressure and Rate of Pressure Rise for Combustible Dust". This test method provides a procedure for performing laboratory tests to evaluate deflagration parameters of dusts. The parameters measured are the maximum pressure output and the maximum rate of pressure rise. These values are applicable to the design of protective measures, such as deflagration venting. The tests are performed using a U.S. Bureau of Mines 20-L explosibility test chamber.

(Source: Fauske & Associates LLC, *Process Safety Newsletter*, Vol. 13, No. 2, Spring 2006.)

Many dusts will ignite explosively at temperatures well below 500°C. Recently, infrared hot body detectors have been used in ordinary light without triggering. The temperature limits of the detectors are low, typically 400, 250, or 175°C. These low temperatures, well below the almost 700°C threshold of spark detectors, make it possible to use this technology to prevent dust explosions. This technology is especially important for very ignition sensitive dusts. Older spark detection technology is inadequate in many circumstances involving today's high technology products and large-scale production requirements [70].



**Figure 9-68f** Furnace apparatus for minimum ignition temperature determination. (From Hazard Evaluation Laboratory, Ltd.)

The same laws of flammability apply as were previously presented for flammable gases/vapors, following the “cubic law” [10] for explosive violence [30].

$$(dp/dt)_{\max, V1} (V_1)^{1/3} = (dp/dt)_{\max, V2} (V_2)^{1/3} \quad (9-73)$$

$= K_G$  (gases), bar m/s

or  $K_{st}$  (dust), bar m/s

$$(dp/dt)_{P_{red}, V1} (V_1)^{1/3} = (dp/dt)_{P_{red}, V2} (V_2)^{1/3} \quad (9-74)$$

$= (K_G)_{P_{red}}$  or  $(K_{st})_{P_{red}}$ , bar m/s

where

$G$  = for gas systems

$st$  = for dust system

$f$  = specific relief area

$(dp/dt)_{\max}$  = maximum rate of pressure rise, bar/s.

$$f_1 (V_1)^{1/3} = f_2 (V_2)^{1/3}$$

Both qualitative and quantitative assessments are necessary to measure explosion characteristics, which are essential to the design



**Figure 9-68g** Apparatus for classifying dusts into categories. (From Hazard Evaluation Laboratory, Ltd.)

of explosion protection methods. These explosion characteristics are as follows.

- The maximum explosion pressure,  $P_{\max}$ . This is the highest explosion pressure developed by an enclosed dust explosion. It is measured in a standard test at the optimum dust concentration.
- The maximum rate of pressure rise  $(dp/dt)_{\max}$ . This is the highest rate of pressure rise generated by an enclosed dust explosion. It is measured in a standard test at the optimum dust concentration.

Figure 9-69 shows pressure vs. time profiles of a vented and unvented explosion. The maximum rate of pressure rise is a parameter which is helpful in assessing the type of explosion suppression system and can be used to give guidance on the size of explosion relief required to safely vent an explosion. The maximum rate of pressure rise obtained from the Hartmann bomb apparatus is related to the required size of explosion relief of the vent ratio method. This is an empirical relationship as shown in Table 9-32 and illustrated in Figure 9-70.

Figures 9-71b–l show the venting area requirements for dust explosions. These are based on high energy ignition sources. The  $K_{st}$  value is derived only from measurements in either the 1 m<sup>3</sup> vessel or the 20 L sphere. Comparisons of results from these equipment testers show that [67]:

1. The values for the maximum explosive pressure,  $P_{\max}$  measured in the 20 L sphere are slightly lower than those measured in the 1 m<sup>3</sup> apparatus.
2. The  $K_{st}$  values are equal up to about 600 bar m/s.

**TABLE 9-31** Dust Cloud Explosibility Test Methods

Test	Standard	Applications	Advantages	Limitations
Minimum Explosible Concentration (MEC)	ASTM E1515 CEN prEN 14034-3	Prevention via dust concentration control, e.g., in pneumatic conveying	The measured MEC value can be compared to <i>in situ</i> measurements of suspended dust concentrations in conveyors and other process equipment, and can be used to check whether the concentration is below the MEC.	Measured MEC value is based on a pressure rise of one atmosphere above the pressure due to the igniter; explosions with smaller pressure rises can occur at concentrations below the reported MEC value.
Minimum Cloud Autoignition Temperature (MAIT)	ASTM E1491 IEC 1241-2-1	Safe operating temperatures in heated process equipment	MAIT is valuable data for both particulate material manufacturing and for post-production processing by other companies and facilities.	Horizontal ovens yield lower MAIT values than vertical furnaces. MAIT value depends on the residence time of the dust cloud in the heated equipment, and on the area of a heated surface.
Minimum Ignition Energy (MIE)	ASTM E2019 IEC 1241-3	Electrostatic ignition hazard evaluations; flexible intermediate bulk container (FIBC) material classes	MIE value determines what precautions are needed in silo/bag filling and other particulate handling operations.	Measured MIE value depends on the dust cloud turbulence level as well as the amount of inductance in the spark generation circuit.
Maximum Explosion Pressure ( $P_{\max}$ ) and $K_{st}$	ASTM E1226 ISO 6184	Deflagration containment, deflagration venting or suppression	$K_{st}$ is often considered the most important parameter to characterize dust material combustibility.	Measured $K_{st}$ value depends on both the ignition energy and the dust cloud turbulence level as determined by the time delay between the dust injection and the ignition.
Limiting Oxygen Concentration (LOC)	ASTM E2079 CEN prEN 14034-4	Inerting per NFPA 69 and NFPA 654	LOC provides valuable explosion prevention data.	Measured LOC values vary with ignition energy and with the particular inert gas used in the test.

(Source: Zalosh, R., et al. [69], Reproduced with permission from the AIChE, All rights reserved.)

**TABLE 9-32** Ratios for the Vent Area Method for Equipment Volume up to  $30\text{ m}^3$ 

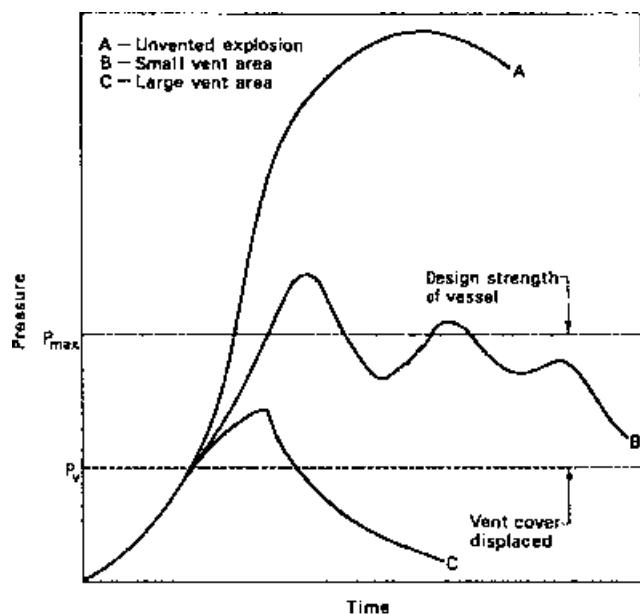
Maximum Rate of Pressure Rise ( $\frac{dp}{dt}_{\max}$ bar/s)*	Vent Ratio (Area of Vent per Volume of Plant) ( $\text{m}^2/\text{m}^3 = \text{m}^{-1}$ )
<350	1/6.1
350–700	1/4.6
>700	1/3.1

(Source: Lunn, G [67])

\*1 bar = 100 kN/m<sup>2</sup> = 14.5 lb<sub>f</sub>/in.<sup>2</sup>

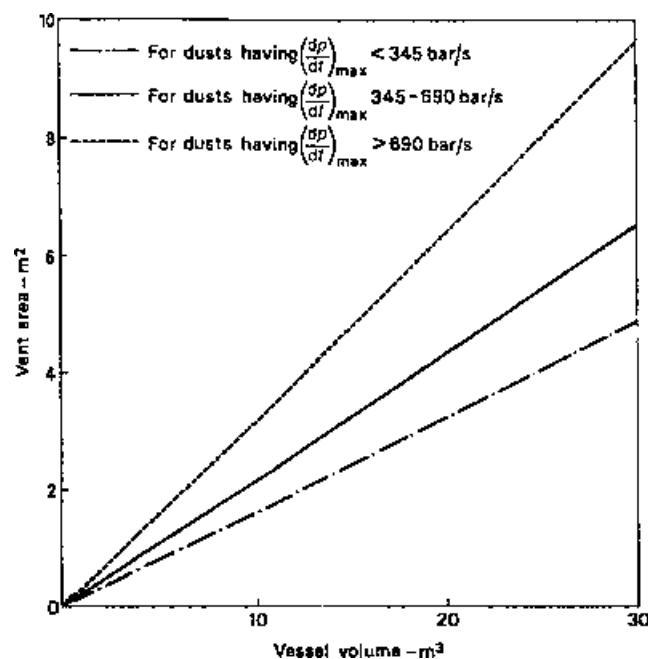
The  $K_{st}$  value can be used to classify dusts into one of several groups. Table 9-33 shows the classification that is generally adopted.

Equation (9-73) is known as the “cubic law” or “cube root law.”  $K_{st}$  is considered to be a constant for any dust, independent of vessel size, and Eqs (9-73) and (9-74) act as a simple scaling law. However, there are limitations to the conditions under which this scaling law is strictly applicable. In practical situation, this means that any apparatus used must be calibrated against the 1 m<sup>3</sup> or 20 L standard test vessels. Any deviation from the established procedure requires an alteration in the ignition delay,  $t_d$ , until concurrence with measurements in the 1 m<sup>3</sup> vessel is obtained. The 20-L spherical tester is recommended as a standard explosibility test in many countries including the US and the UK.



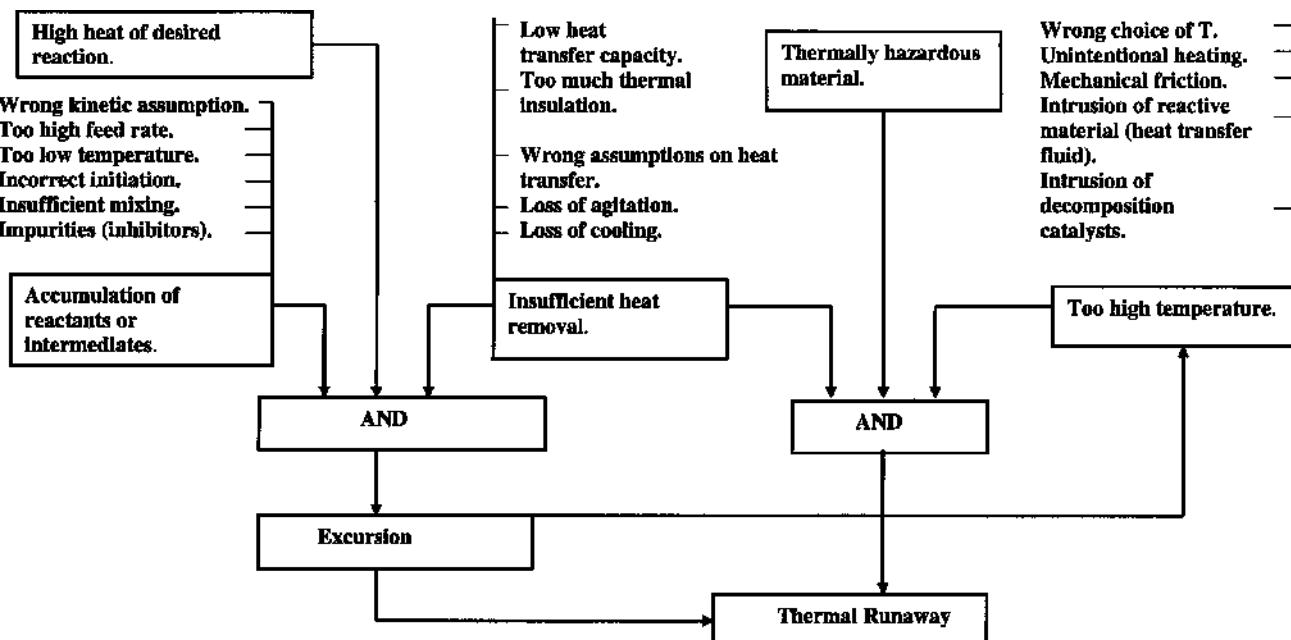
**Figure 9-69** Typical pressure-time history of a vented and unvented explosion. (From P. Field [66].)

The dust hazard class,  $K_{st}$ , vessel volume and strength, and the relieving pressure of the vent closure are the key components of the relief determination using the nomographs [10]. Although stated by reference [10] to be non-exact but sufficiently accurate for industrial use, care must be used in selecting the conditions for design. Classification of a dust in a specific class is not an assurance of the probability of a dust explosion [30].

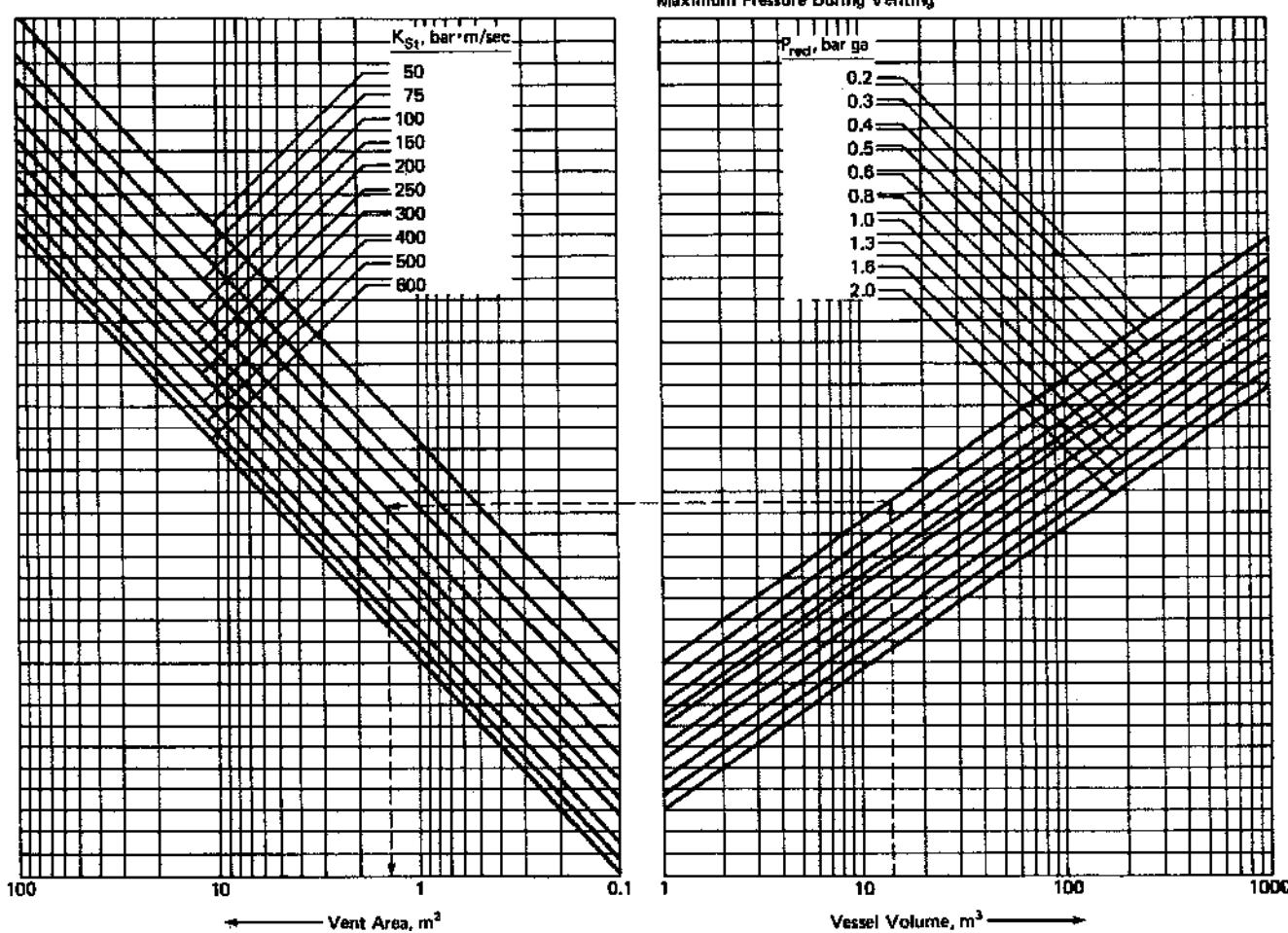


**Figure 9-70** Sizing of explosion relief vent areas by the vent-ratio method. (From P. Field [66].)

Explosions occur in various chemical processing equipment including silos, grinders, conveying equipment, filters, dryers, furnaces, and mixers. However, chemical plants can avoid many of these disasters by following NFPA standards. The NFPA has produced a comprehensive set of guidelines and codes that provide practical solutions to the explosion threats. These standards are followed by most regulatory bodies. A deflagration is a combination



**Figure 9-71a** Causes of runaways in industrial reactors. (Source: W. Regenass, "Safe Operation of Exothermic Reactions", Internal Ciba-Geigy Publication, 1984.)



**Figure 9-71b** Venting nomograph for dust,  $P_{\text{stat}} = 0.1 \text{ barg}$ . (Reprinted with permission from NFPA 68-1988, *Deflagration Venting* (1988), National Fire Protection Association, Quincy, MA 02269). See note Figure 9-63a.

of three elements, namely fuel, oxygen, and an ignition source. Fuel can come from a combustible gas, dust, mist, or vapor. Oxygen is plentiful in every chemical process, and all that is required is an ignition source to combine with the mixture. The ignition source can have energy as low as that released by a static discharge, or can be caused by friction, lightning, and sparks from a welding torch or other energy sources. When the fuel and air are premixed in a flammable concentration and an ignition source of sufficient energy to cause combustion is introduced, deflagration occurs. The flame can travel through the mixture at a rate governed by how quickly the fuel will burn. Deflagration propagation rates can range between 5 and 30 ft/s (1.5 and 9 m/s). This speed can increase dramatically when the gas or dust mixtures are pre-compressed or in a turbulent condition. Such a situation can produce flame speeds of more than 1000 ft/s.

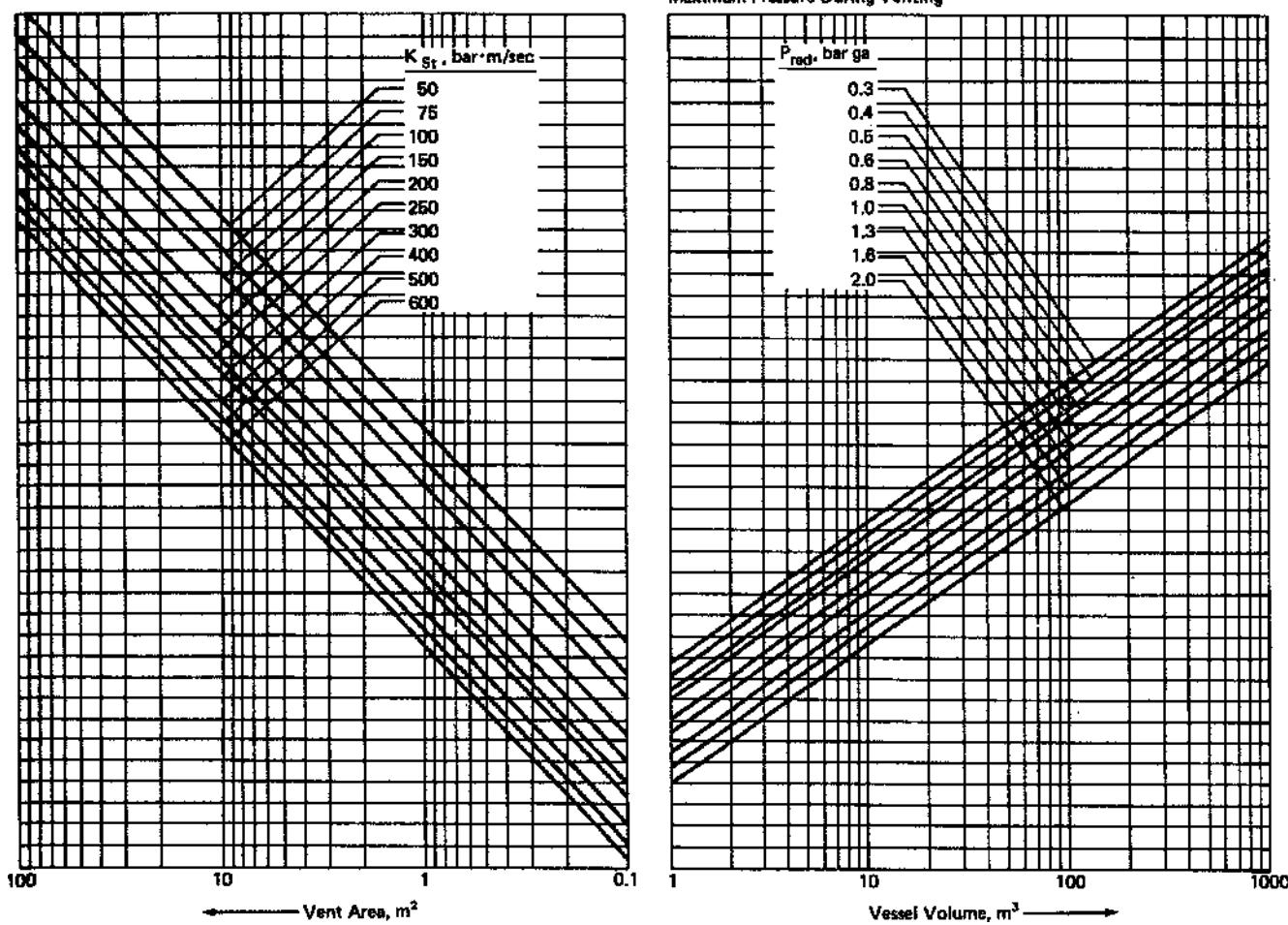
Garzia reports [71] that in the event of deflagration in a closed vessel, the rapidly expanding gases would create pressures between 100 and 150 psig, and most pressure vessels in the chemical process industry (CPI) rupture well below these pressure levels, often at pressures lower than 5 psig. Once the explosibility of the material is known, a process review can help to determine the explosion risk. Such an evaluation should assist engineers in providing information

critical to the design of the explosion protection system. A process review should identify the following.

- ignition sources
- existing explosion protection hardware
- process strength
- interconnections (ducts, pipes)
- mechanical barriers (screw conveyors, rotary gate valves)
- maximum and minimum process pressures and temperatures
- process volumes and dimensions
- process location (as it relates to outside walls and roofs)
- material concentrations and particle size distribution.

The dust hazard classes for deflagrations are given in Table 9-33 (for data see Tables 9-33 and 9-35).

Using a nomograph requires only the vessel volume in meters, selecting the dust class, St-1, St-2, or St-3 from Table 9-33. Using Table 9-34 or 9-35 select the  $K_{\text{st}}$  value determined experimentally. The reduced pressure,  $P_{\text{red}}$  (maximum pressure actually developed during a vented deflagration, termed “reduced explosion pressure”), must not exceed strength of vessel (see earlier discussion) and the  $P_{\text{stat}}$ , that is the vent device release pressure. Note that the static activation pressure,  $P_{\text{stat}}$ , must be determined from experimental tests of the manufacture of relief panels such as rupture disks.



**Figure 9-71c** Venting nomograph for dust,  $P_{\text{stat}} = 0.2 \text{ barga}$ . (Reprinted with permission from NFPA 68-1988, *Deflagration Venting* (1988), National Fire Protection Association, Quincy, MA 02269). See note in Figure 9-63a.

### 9.77 SIZING OF VENTS METHODS

Safe sizing of vent openings is the most essential design parameter in the application of explosion venting. Various techniques for sizing vents have been proposed; however, they are not applicable in all cases. Any attempt to apply a single technique indiscriminately may result either to uneconomic and impractical large vents or more seriously to inadequate vents which could result in extensive damage and fatal injury.

The procedure of choosing the best method of vent sizing can be complex as it requires sufficient information about both the plant and the process. Maximum information is required for the basic and commonly used vent sizing techniques for compact vessels, as the results from these techniques have been shown to be accurate and reliable in practice. Figure 9-72 outlines a general approach to practical dust explosion hazard in industry, and Figure 9-73 illustrates a logic diagram or decision trees with an assessment of the capability of a dust to take part in an explosion. Here, the flow chart considers pertinent questions required for the correct decisions that have to be taken before a final choice of vent sizing technique can be safely employed. Lunn [67] has provided various logic diagrams or trees that take the reader through the decisions that are necessary before either the option can be accepted or another option is considered.

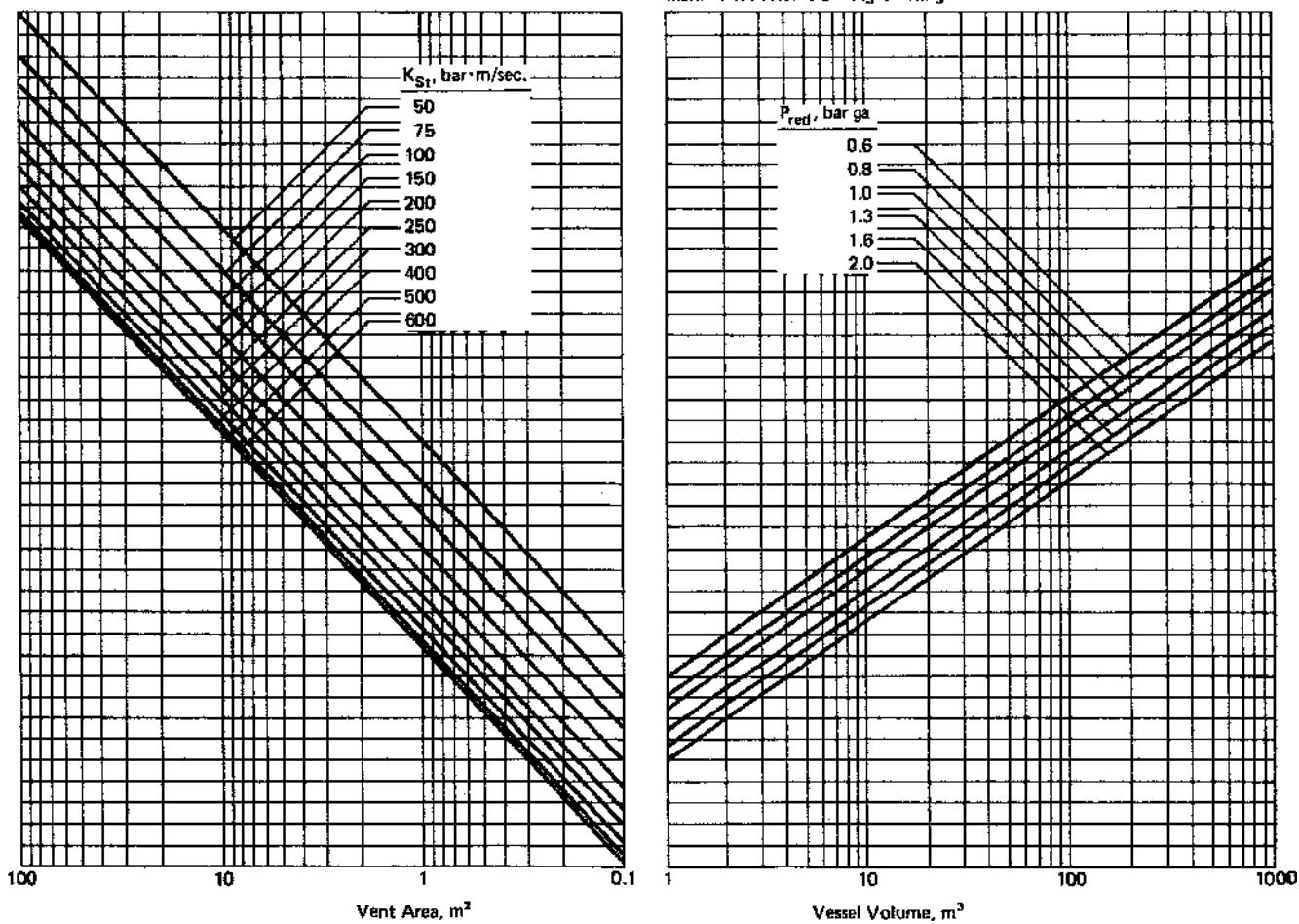
### 9.78 THE VDI NOMOGRAPH METHODS

This method was originally designed by Verein deutscher Ingenieure [72], and was later adopted for the United States by the NFPA [10]. The  $K_{\text{st}}$  nomograph method for assessment of vent area was derived from the theoretical work of Heinrich [73] and the experimental work of Bartknecht [74] and Donat [75]. The method has found widespread acceptance throughout Europe, the United States, and elsewhere.

The following parameters required for the  $K_{\text{st}}$  nomograph are as follows.

- The reduced explosion pressure,  $P_{\text{red}}$  (bara).
- The volume of the dust-handling plant ( $\text{m}^3$ ).
- The explosibility characteristic – the  $K_{\text{st}}$  value of the dust as measured in a standard test (bar m/s).
- The static opening pressure of the vent cover,  $P_{\text{stat}}$  (bara). It is assumed that the vent has low inertia with an area density less than  $10 \text{ kg/m}^3$ . No provision is made for vent covers of high inertia.

Heinrich [73] used experimental data of Donat [75] to develop a theoretical model, which was subsequently employed to design a series of nomographs which are reproduced in Figures 9-71j-l. Each of the nomographs is applicable for a given value of the



**Figure 9-71d** Venting nomograph for dust,  $P_{stat} = 0.5$  barga. (Reprinted with permission from NFPA 68-1988, *Deflagration Venting*, (1988) National Fire Protection Association, Quincy, MA 02269). See note in Figure 9-63a.

vent bursting pressure,  $P_{stat}$ . Example 9-21 illustrates the method of using the nomograph.

The published nomographs are applicable in the following conditions [67].

- Vent bursting pressure,  $P_{stat}$  greater than 1.1 bara
- Reduced explosion pressures,  $P_{red}$  greater than 1.2 bara
- $K_{st}$  values greater than 10 bar m/s and less than 600 bar m/s
- Values of  $P_{max}$  less than 11 bara for St 1 and St 2 dusts and  $P_{max}$  values less than 13 bara for St 3 dusts
- Vessel volumes less than 1000 m<sup>3</sup>
- Length-to-diameter ratio of the vessel less than 5:1
- No vent ducts are fitted to the vent.

### 9.79 THE ST GROUP NOMOGRAPH METHOD

These series of nomographs rank dusts in the three St groups namely: St 1, St 2, and St 3. The nomographs are reproduced in Figures 9-74a–c, and the method of their use is shown in Figure 9-74a. Example 9-22 shows a worked example of using the St nomograph.

The St nomographs are based on the experimental data rather than on Heinrich's theory. Therefore, there are differences between these nomographs and the  $K_{st}$  nomographs. For example, the St 1 line in the St nomographs will not exactly coincide with the

$K_{st} = 200$  bar m/s from the nomographs. The differences are slight; however, if the dust has a low  $K_{st}$  value within its specific St group, the St nomographs are bound to oversize the vent area, a result which may be of practical or economic significance [67].

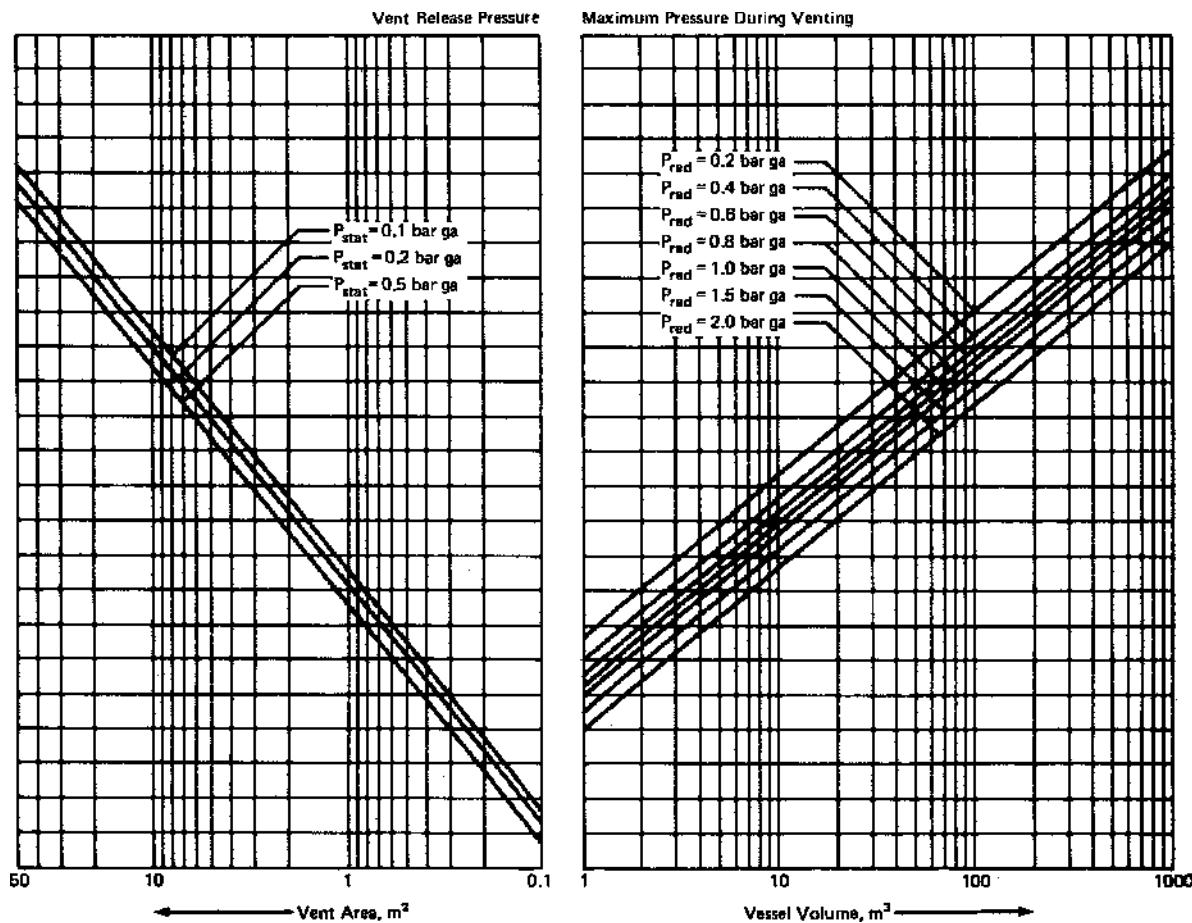
### 9.80 REGRESSION ANALYSIS FROM THE $K_{st}$ NOMOGRAPHS

Recently, equations have been fitted to the original  $K_{st}$  nomographs published in VDI 3673. These are not theoretical derivations, but are regression analyses, and the equations can be used for rapid calculation of vent area using computer program.

An equation developed by Simpson [76] is recommended in NFPA 68 [10].

An equation from reference [10] represents the dust explosion nomographs, Figures 9-71b–d. Because the equation was derived from the nomographs, it is no more accurate but may be more convenient:

$$A_v = (a) (V^{2/3}) (K_{st})^b (P_{red})^c \quad (9-75)$$



**Figure 9-71e** Venting nomograph for classes of dusts,  $P_{\text{stat}} = 0.1 \text{ barg}$ . (Reprinted with permission from NFPA 68-1988, *Deflagration Venting* (1988), National Fire Protection Association, Quincy, MA 02269). See note in Figure 9-63a.

where

$$a = 0.000571 e^{(2)(P_{\text{stat}})}$$

$$b = 0.978 e^{(-0.105)(P_{\text{stat}})}$$

$$c = -0.687 e^{(0.226)(P_{\text{stat}})}$$

$$A_v = \text{vent area, } \text{m}^2$$

$$V = \text{enclosure volume, } \text{m}^3$$

$$e = 2.718, \text{ natural logarithm}$$

$$P_{\text{red}} = \text{maximum pressure developed during venting, barg}$$

$$P_{\text{stat}} = \text{vent closure release pressure, barg}$$

$$K_{\text{st}} = \text{deflagration index for dust, bar m/s, Table 9-33.}$$

The limits of this equation are the same as the limits to the  $K_{\text{st}}$  nomographs.

$$2 \text{ barg} > P_{\text{red}} > P_{\text{stat}} + 0.1 \text{ barg}$$

$$0.5 \text{ barg} > P_{\text{stat}} > 0.1 \text{ barg}$$

$$600 > K_{\text{st}} > 10 \text{ bar m/s}$$

$$1000 \text{ m}^3 > V > 1 \text{ m}^3:$$

### 9.81 EQUATIONS TO REPRESENT THE NOMOGRAPHS

The NFPA-68 presents equations to represent the respective nomographs for dusts (Figures 9-71b-i) but they are stated to be no more accurate than the nomographs. The following equations can be used as alternatives to the nomographs.

For St 1 dusts

$$\log A_v = 0.77957 \log V - 0.42945 \log P_{\text{red}} - 1.24669 \quad (9-76)$$

For St 2 dusts

For vessel volumes in the range  $1\text{--}10 \text{ m}^3$ , that is,  $1 \text{ m}^3 < V < 10 \text{ m}^3$ ,

$$\log A_v = 0.64256 \log V - 0.46527 \log P_{\text{red}} - 0.99461 \quad (9-77)$$

For vessel volumes in the range  $10\text{--}1000 \text{ m}^3$ , that is,  $10 \text{ m}^3 < V < 1000 \text{ m}^3$ ,

$$\begin{aligned} \log A_v = & 0.74461 \log V - 0.50017 \log (P_{\text{red}} + 0.18522) \\ & - 1.02406 \end{aligned} \quad (9-78)$$

where

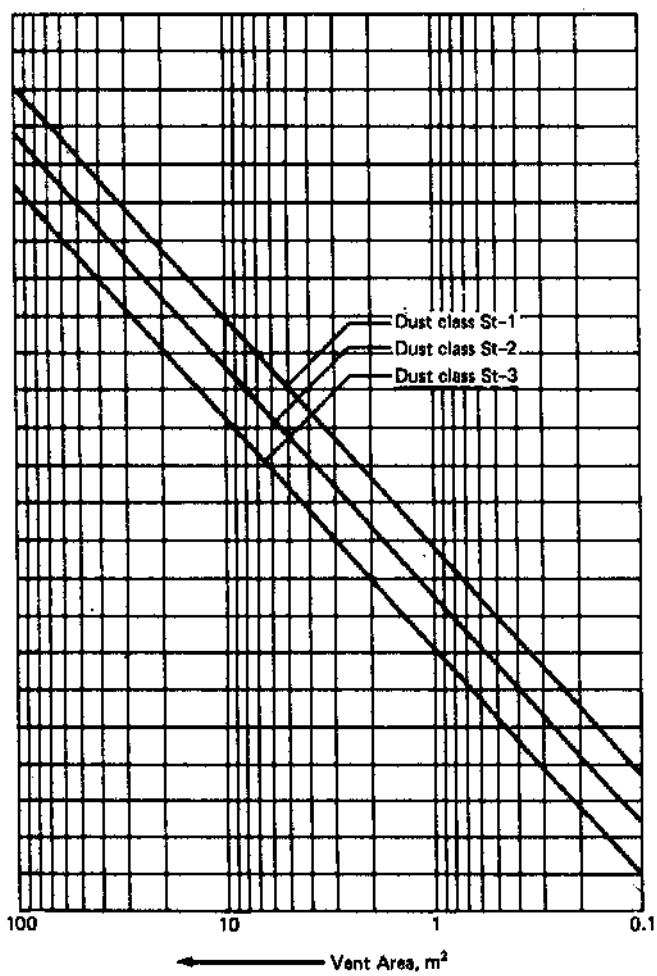
$$A_v = \text{vent area, } \text{m}^2$$

$$V = \text{vessel volume, } \text{m}^3$$

$$P_{\text{red}} = \text{reduced explosion pressure, barg}$$

$$P_{\text{stat}} = \text{vent closure release pressure} = 0.1 \text{ barg.}$$

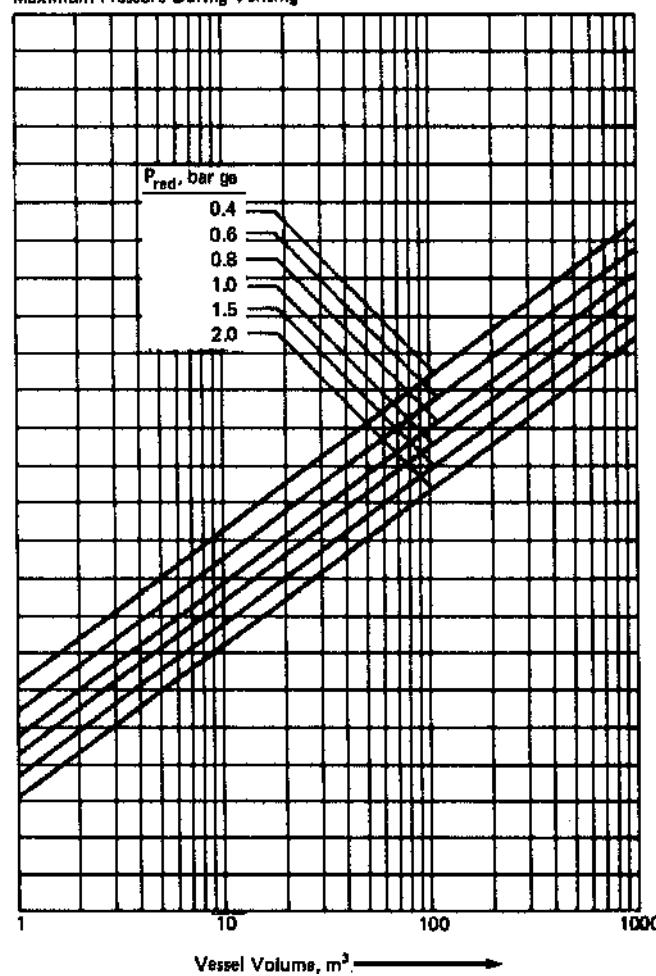
These equations are a satisfactory simulation of the Radand nomographs.



**Figure 9-71f** Venting nomograph for classes of dusts,  $P_{\text{stat}} = 0.2 \text{ barga}$ . (Reprinted with permission from NFPA 68-1988, *Deflagration Venting* (1988), National Fire Protection Association, Quincy, MA 02269). See note in Figure 9-63a.

Scholl [77] has described an empirical equation for the determination of vent areas:

$$A_v = [3.264 \times 10^{-5} P_{\max} K_{\text{st}} P_{\text{red}}^{-0.569} + 0.27 (P_{\text{stat}}^{-0.1}) P_{\text{red}}^{-0.5}] V^{0.753} \quad (9-79)$$



where the various pressures are in units of barg. It applies when  $P_{\max} \leq 9 \text{ barg}$  for St 1 and St 2 dusts and  $P_{\max} \leq 12 \text{ barg}$  for St 3 dusts; for vessel volumes between 0.1 and  $10,000 \text{ m}^3$ , for  $P_{\text{stat}}$  between 0.1 and 1 barg and  $P_{\text{red}}$  between 0.1 and 2 barg. The  $L/D$  ratio of the vessel should not exceed 2.

#### EXAMPLE 9-21 Use of the Dust Nomographs

A storage silo for lignite particles (not lumps) is 10 ft diameter by 12 ft tall. What protection is needed to guard against destruction from a dust explosion (deflagration)?

From Table 9-35c, lignite is dust hazard class 1, with  $K_{\text{st}} = 151 \text{ bar m/s}$ .

- Vessel volume  $= \pi(10)^2(12)/4 = 942.4 \text{ ft}^3$

$$= 942.4/35.3 = 26.7 \text{ m}^3$$

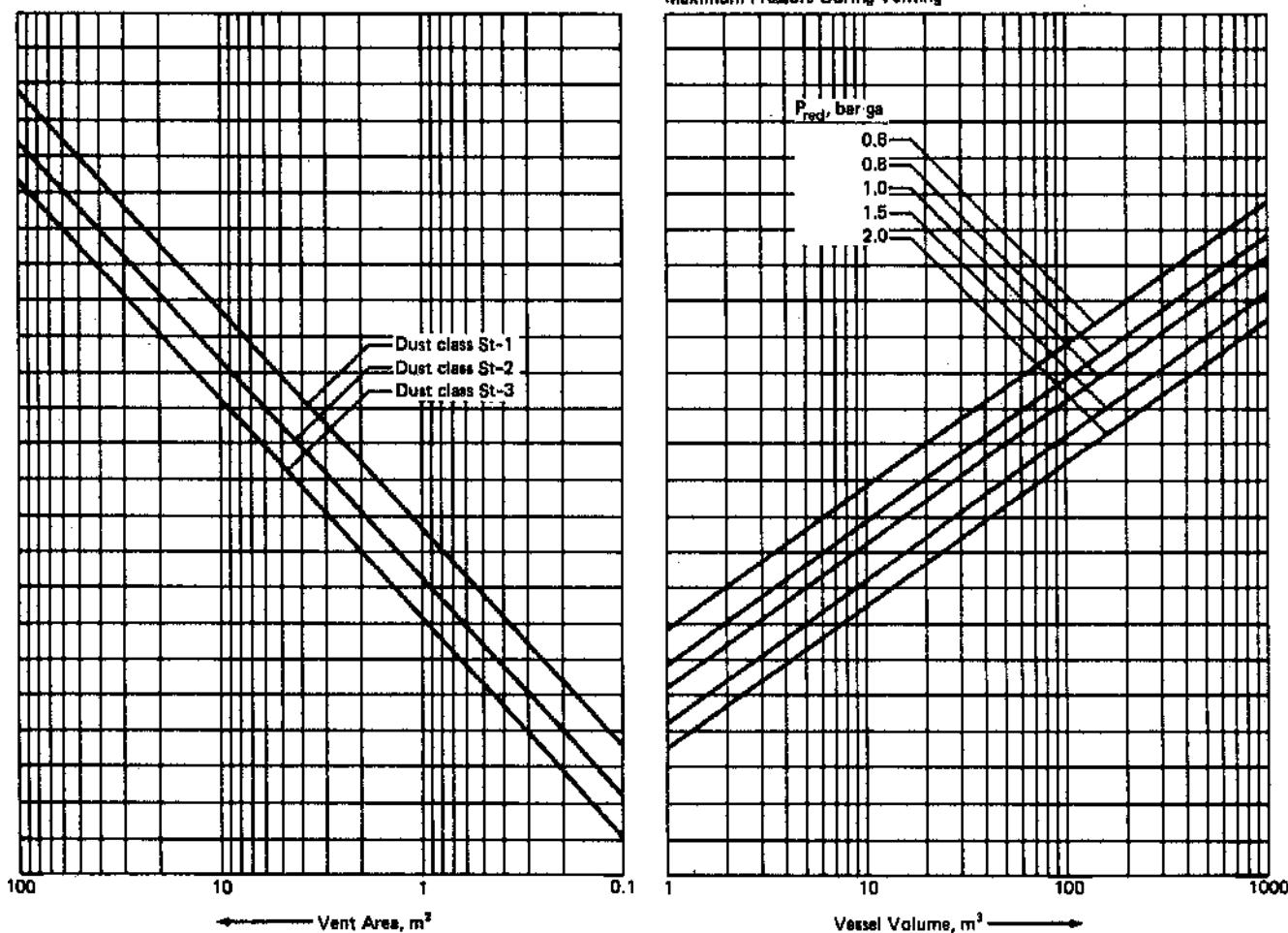
$$L/D = 12/10 = 1.2 \text{ length/diameter is ok.}$$

- $P_{\text{red}}$ , barga  $= 2.0 = (2)(14.5) = 29 \text{ psig}$  maximum allowable vessel internal pressure. Note: 1 bar  $= 14.5 \text{ psi}$

- Reading vent area (Figure 9-71b)  $= 0.50 \text{ m}^2$

$$\begin{aligned} &= 0.50 \times 10.8 \text{ ft}^2/\text{m}^2 \\ &= 5.4 \text{ ft}^2 \end{aligned}$$

select,  $P_{\text{stat}} = 0.1 \text{ barga}$ , vent device release pressure  $= (0.1)(14.5) = 1.45 \text{ psig}$ .



**Figure 9-71g** Venting nomograph for classes of dusts,  $P_{\text{stat}} = 0.5 \text{ barga}$ . (Reprinted with permission from NFPA 68-1988, *Deflagration Venting* (1988), National Fire Protection Association, Quincy, MA 02269). See note in Figure 9-63a.

#### An Alternative Method: Using Equations to Represent the Nomographs

Using the Eq. (9-76) for dust with Class St 1 and  $K_{\text{st}} = 151$

$$\log A_v = 0.77957 \log V - 0.42945 \log P_{\text{red}} - 1.24669$$

where  $V = 26.7 \text{ m}^3$

At  $P_{\text{red}} = 1.0 \text{ barg}$  (2.0 bara) and  $P_{\text{stat}} = 0.1 \text{ barg}$

The vent area  $A_v$  is

$$\log A_v = 0.77957 \log V - 0.42945 \log P_{\text{red}} - 1.24669$$

$$\log_{10} A_v = 0.77957 (\log 26.7) - 0.42945 (\log 1.0) - 1.24669$$

$$\log A_v = -0.13462$$

The vent area,  $A_v$  is

$$A_v = 0.733 \text{ m}^2$$

$$= 0.733 \times 10.8$$

$$= 7.29 \text{ ft}^2$$

*Note:* As  $P_{\text{red}}$  and  $P_{\text{stat}}$  change for a given class of dust, the vent area required also changes. Examine charts.

Fauske [78] proposed a convenient general equation for deflagration venting of both low and high pressure vessels. The following formula for estimating the vent area in case of deflagrations is

$$A_v = \frac{V^{2/3} v^{1/3} \dot{P}_{\text{MAX}}}{CP [2(P - P_o)/\rho_g]^{1/2}} \quad (9-80)$$

where

$A_v$  = vent area,  $\text{m}^2$

$V$  = vessel volume,  $\text{m}^3$

$v$  = test vessel volume,  $\text{m}^3$

$\dot{P}_{\text{MAX}}$  = measured peak rate of pressure rise in unvented volume,  $v$  ( $\text{Pa/s}$ )

$P$  = maximum venting pressure,  $\text{Pa}$

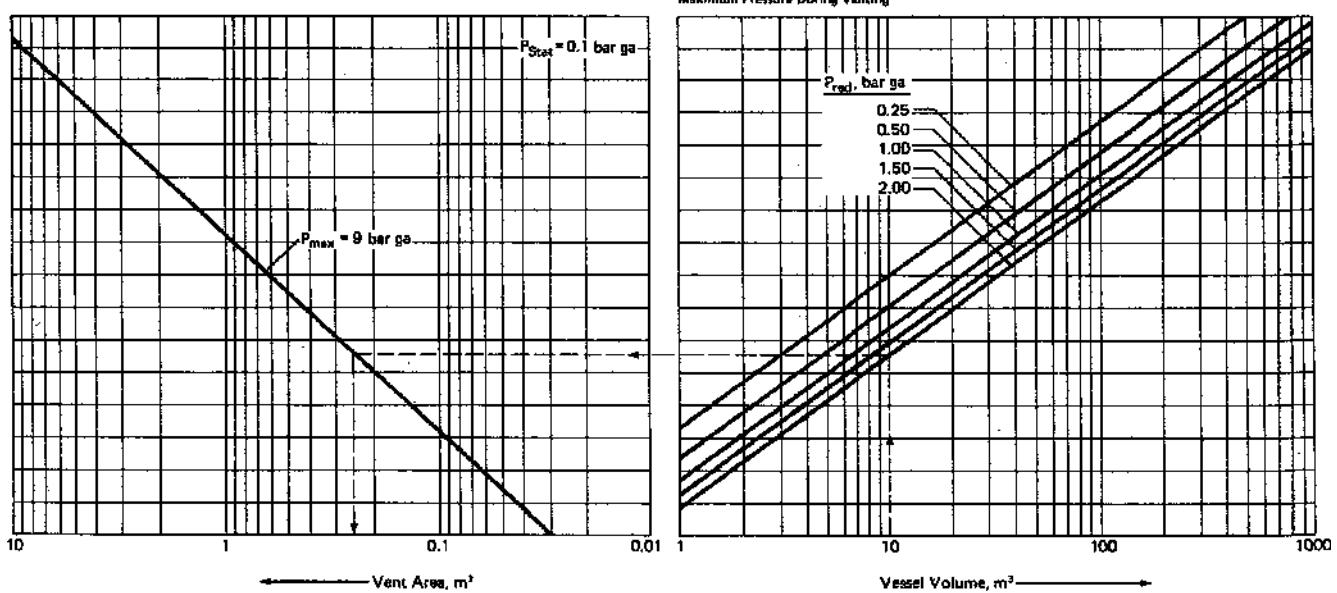
$P_o$  = ambient pressure,  $\text{Pa}$

$\rho_g$  = gas density,  $\text{kg/m}^3$ .

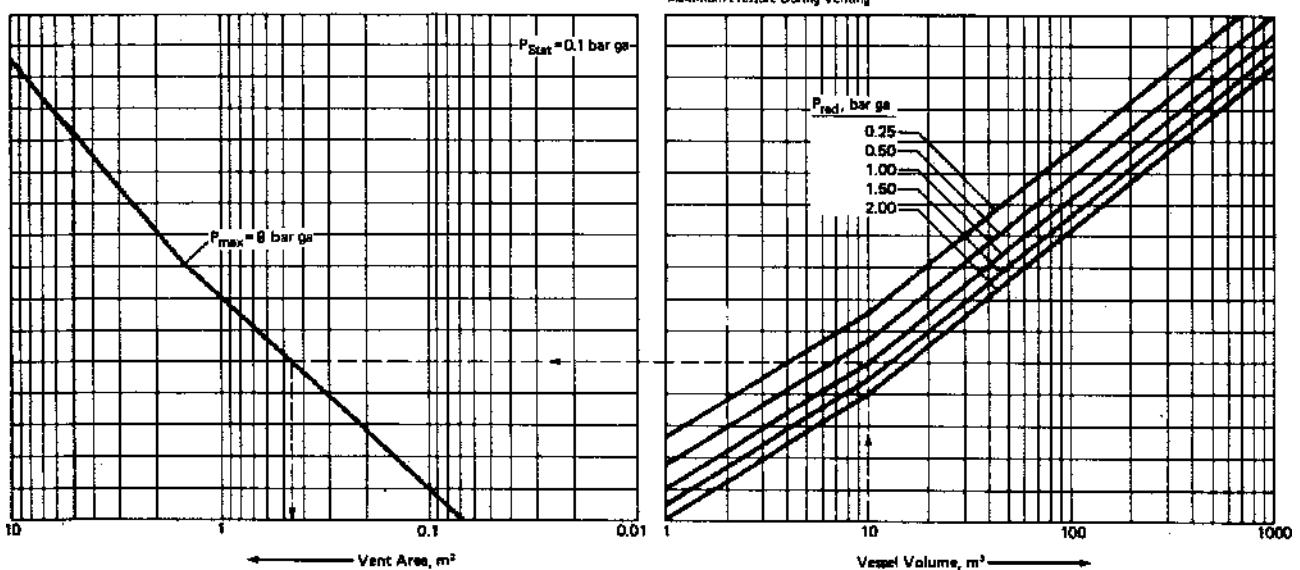
And the values of  $C$  indicate the degree of turbulence prior to ignition in the test vessel.

For fuel-oxidizer gas mixtures (vapor deflagration)  $\dot{P}_{\text{MAX}}$  is defined as

$$K_g (\text{Pa m/s}) = \dot{P}_{\text{MAX}} v^{1/3} \quad (9-81)$$



**Figure 9-71h** Venting nomograph for classes of dusts, St-1 whose maximum deflagration pressure does not exceed 9 barga. (Reprinted with permission from NFPA 68-1988, *Deflagration Venting* (1988), National Fire Protection Association, Quincy, MA 02269). See note in Figure 9-63a.



**Figure 9-71i** Venting nomograph for classes of dusts, St-2 whose maximum deflagration pressure does not exceed 9 barga. (Reprinted with permission from NFPA 68-1988, *Deflagration Venting* (1988), National Fire Protection Association, Quincy, MA 02269). See note in Figure 9-63a.

or

$$K = \dot{P}_{\text{max}} v^{1/3} \quad (9-81a)$$

where

$v$  = standard test volume ( $\text{m}^3$ )

$\dot{P}_{\text{max}}$  = maximum measured rate of pressure rise (bar/s).

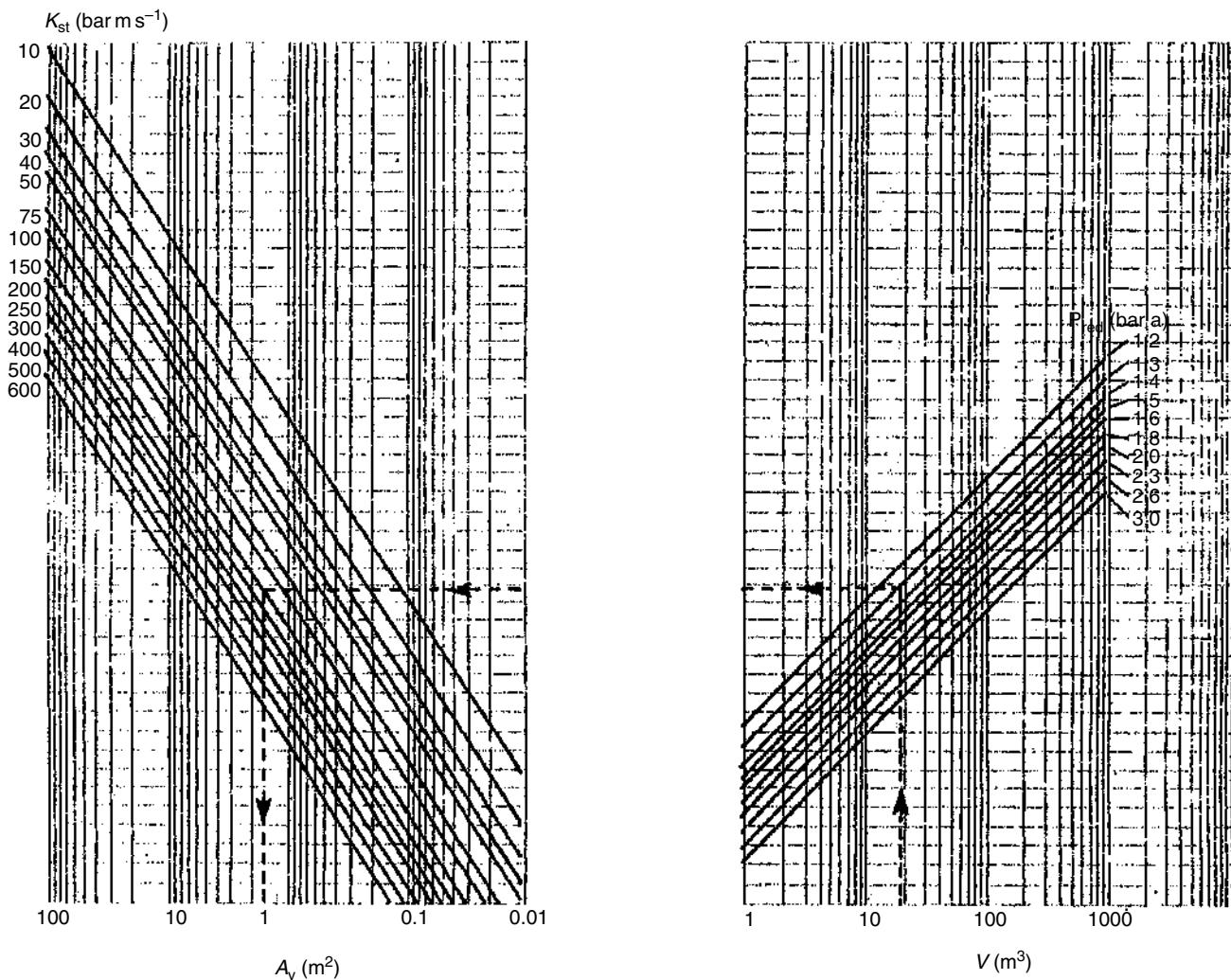
The measurement of  $\dot{P}_{\text{max}}$  is generally made on a quiescent or non-turbulent mixture and  $C = 1.0$  and Eq. (9-80) can be expressed by

$$A_v = \frac{V^{2/3} K_g}{P [2(P - P_o)/\rho_g]^{1/2}} \quad (9-82)$$

Equation (9-82) provides predictions consistent with large-scale hydrocarbon-air deflagration data.

$$\text{The measurement of } \dot{P}_{\text{MAX}} \text{ or } K_{\text{st}} (\text{Pa m/s}) = \dot{P}_{\text{MAX}} v^{1/3} \quad (9-83)$$

For dust, deflagration is made on initially highly turbulent conditions (in contrast to typical industrial conditions) where the degree



**Figure 9-71j** Extended nomograph:  $P_{stat} = 1.1$  bar(a). (From G. Lunn [67], by permission from the IChemE, all rights reserved.)

of turbulence is of the order of 6 relative to quiescent conditions. Setting  $C = 6$ , Eq. (9-80) can be expressed as:

$$A_v = \frac{V^{2/3} K_{st}}{6P [2(P - P_0)/\rho_g]^{1/2}} \quad (9-84)$$

Equation (9-84) provides predictions consistent with industrial dust conditions.

Fauske [79] further proposed an equation for a volumetric venting rate associated with a deflagration event as:

$$\dot{Q}_g = \frac{V^{2/3} K (P_{max} - P)}{P (P_{max} - P_i) - P_{max} (P_s - P_i)} \quad (9-85)$$

where

- $P$  = final venting pressure, bar(a)
- $V$  = vessel volume,  $m^3$
- $P_{max}$  = maximum unvented pressure, bar(a)
- $P_s$  = relief set pressure, bar(a)
- $P_i$  = initial pressure, bar(a)

$K$  = deflagration index measured in a standard non-vented spherical vessel (such as the 20L vessel) with quiescent initial conditions and central ignition and is given by Eq. (9-81a).

The required vent area  $A_v$  is then given by:

$$\frac{A_v}{V^{2/3}} = \left[ P \left( 1 + \frac{18.37}{P^{1.75}} \right)^{0.286} \right]^{-1} \left[ \frac{K}{0.61} \left( \frac{M_w}{RT} \right)^{1/2} \right] \left[ \frac{P (P_{max} - P)}{P (P_{max} - P_i) - P_{max} (P_s - P_i)} \right] \quad (9-86)$$

where

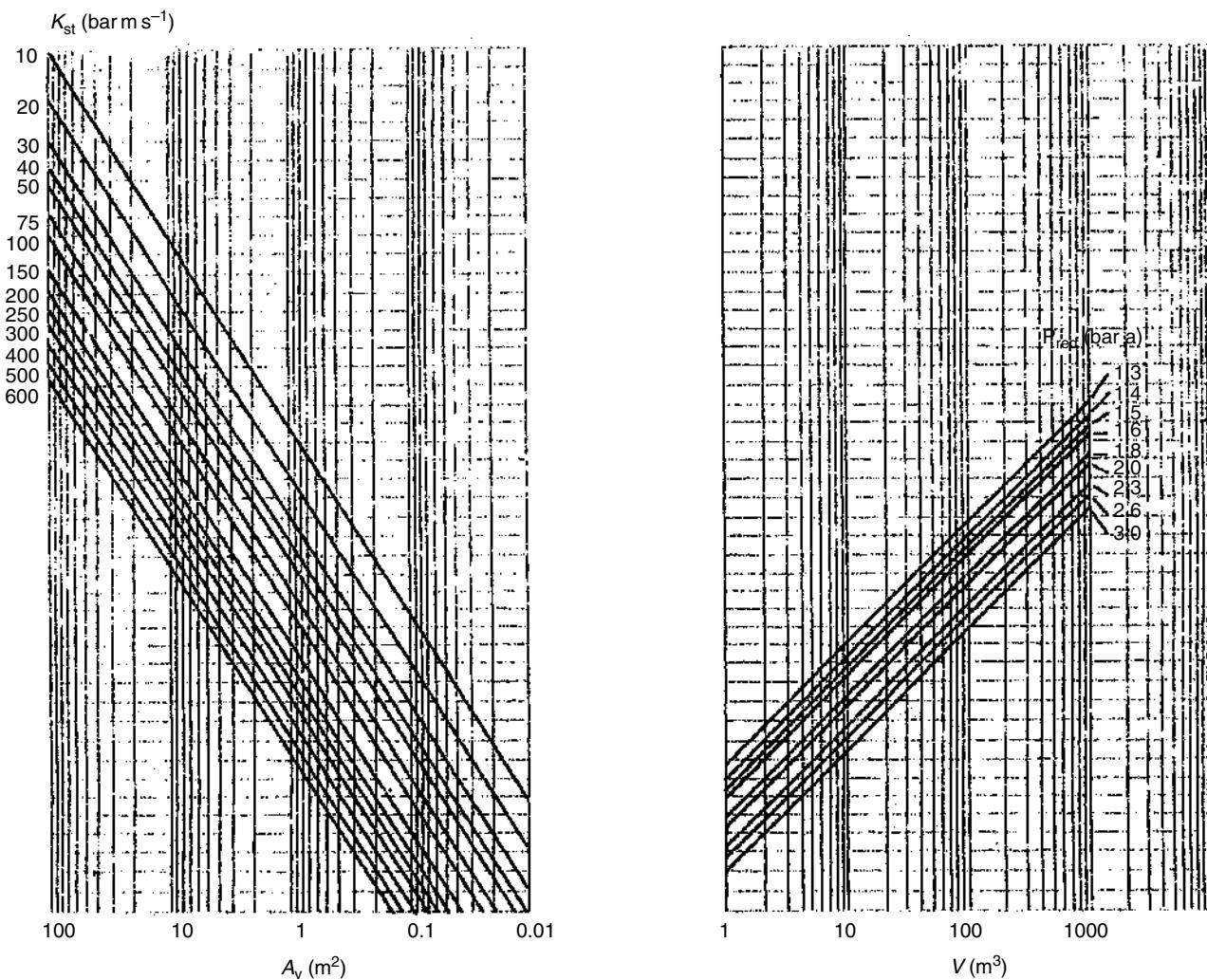
the unit of  $P$  in the first bracket on the right side of Eq. (9-86) is now barg,

$M_w$  = molecular weight

$R$  = gas constant (8314 Pa  $m^3/K kg mol$ )

$T$  = initial temperature (K).

For fuel oxidizer gas mixtures such as propane-air, the measurement of the deflagration index is generally made on an initially



**Figure 9-71k** Extended nomograph:  $P_{stat} = 1.2$  bara. (From G. Lunn [67], by permission from the IChemE, all rights reserved.)

quiescent or non-turbulent mixture and  $K$  in Eq. (9-86) is set equal to  $K_g$ , which provides predictions that are consistent with both subsonic and sonic deflagration experiments. In contrast to gas mixtures, the measurement of  $P_{max}$  or  $K_{st}$  for dust mixtures is made with initially highly turbulent conditions where the degree of turbulence is of the order of 6 relative to quiescent conditions. In order to represent similar initial conditions to that of gas mixtures,  $K$  is set equal to  $K_{st}/6$ , which provides predictions consistent with industrial experience [79].

#### Reference conversions

$$\begin{aligned} 1 \text{ bar m/s} &= 47.6 \text{ psi ft/sec} \\ 1 \text{ psi ft/sec} &= 0.021 \text{ bar m/s} \\ 1 \text{ atm} &= 1.01 \text{ bar} \\ 1 \text{ atm} &= 14.7 \text{ psi} \end{aligned}$$

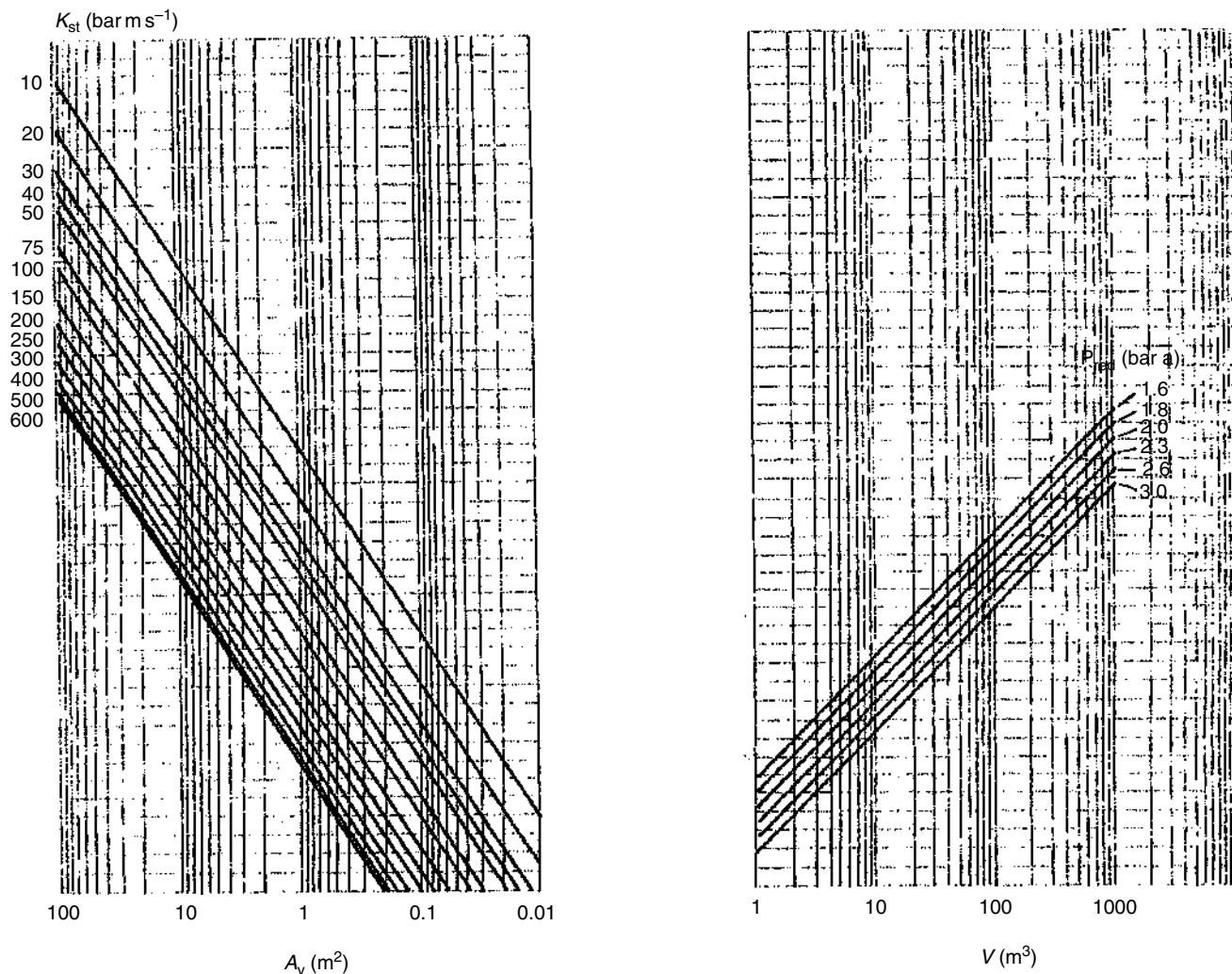
## 9.82 THE VENT RATIO METHOD

The vent ratio is defined as:

$$\text{Vent ratio} = \frac{\text{Area of vent}}{\text{Volume of the vessel}} \quad (9-87)$$

The USA and the UK have employed vent area method using a vent ratio which is based on the maximum rate of pressure rise measured in the Hartmann apparatus. Data from the 20 L sphere or 1-m<sup>3</sup> test apparatus must not be used. For vessels up to about 30 m<sup>3</sup>, the recommended vent ratios are given in Table 9-32. When the opening pressure and the inertia of the vent cover are low, and any vent duct less than 3 m long, the recommended vent ratios are designed to limit the reduced explosion pressure,  $P_{red}$ , to 1.14 bara.

Because of the dimensional nature of the vent ratio, m<sup>-1</sup>, the vent area becomes a greater proportion of the vessel surface as the volume increases. In large vessels, the vent ratio data in Table 9-32 produce large vent areas that they become increasingly difficult to accept. The vent ratio can be reduced as vessel volume increases above 30 m<sup>3</sup> according to the recommendations shown in Figure 9-75. The major criticism of this method is that it provides very large vent areas that can become costly if not impossible to fit. These areas arise because the method is based on rapid flame propagation throughout the whole of the vessel volume rather than on a spherical flame front expanding from a single ignition source. Gibson and Harris [80] suggest that in most dust handling situations such a high degree of turbulence and fragmentation of the flame front is unlikely to occur throughout the whole of the



**Figure 9-71** Extended nomograph:  $P_{stat} = 1.5$  bara. (From G. Lunn [67], by permission from the IChemE, all rights reserved.)

**TABLE 9-33 Hazard Class for Dusts**

Hazard Class	$K_{st}$ (bar m/s)	Maximum Rate Pressure Rise*	Characteristics	Examples†
St-0	0		Non-explosible	Sand, Sodium bicarbonate, Talcum powder
St-1	$0 < K_{st} < 200$	$< 7300$	Weak to moderately explosive	Coal, starch, maize, Polyethene, Sugar
St-2	$200 < K_{st} < 300$	$7300-22,000$	Strongly explosive	Dextrin, Organic pigment
St-3	$K_{st} > 300$	$> 22,000$	Very strongly explosive	Aluminum powder, magnesium powder

Note: The nomographs are limited to an upper  $K_{st}$  value of 600.  $K_{st}$  values determined in an approximate spherical test vessel of at least 20 L capacity. See Tables 9-34 and 9-35 for typical  $K_{st}$  values.

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\* Added by E.E. Ludwig [34].

† Added by this author.

**TABLE 9-34**  $K_{st}$  Values of Technical Fine Dusts – High Ignition Energy

Type of Dust	$P_{max}$ (bar)	$K_{st}$ value (bar m s <sup>-1</sup> )
PVC	6.7–8.5	27–98
Milk powder	8.1–9.7	58–130
Polyethylene	7.4–8.8	54–131
Sugar	8.2–9.4	59–165
Resin dust	7.8–8.9	108–174
Brown coal	8.1–10.0	93–176
Wood dusts	7.7–10.5	83–211
Cellulose	8.0–9.8	56–229
Pigments	6.5–10.7	28–344
Aluminum	5.4–12.9	16–750

By permission from Bartknecht, W., *Explosions*, 2nd ed. (1980), Springer-Verlag.

vessel volume. Therefore, the vent ratio could be expected to specify over size vents.

The explosion characteristics of dusts of the specific type or identification vary with the size and shape of the dust particle [10]. These characteristics and a few other features [10] are summarized in Table 9-33. These data are

- a) Lower flammability limit or minimum explosive concentration of the dust with air, averaging about 30–100 g/m<sup>3</sup>.
- b) Explosion characteristics vary with the degree of dispersion of the dust. Most dusts exhibit an optimum concentration for maximum explosion pressure and pressure rise.
- c) Dust particles of <420 microns may deflagrate. The smaller particles yielding maximum rate of pressure rise and maximum pressure, both increase as particle size decreases, and are more sensitive in the 200–420  $\mu$  range.
- d) Minimum ignition energy is very sensitive to particle size. The use of energy of ignition decreases with decreasing particle size. Decrease in particle size also increases the capacitance of a dust cloud; that is, the size of electrical charge on the cloud.
- e) Mixtures of dusts, combustible gas/vapors, and an oxidant such as air are termed “hybrid mixtures” and have unique character-

istics. Some of these lead to an explosion under conditions not necessarily conforming to the gas or dust; therefore, when the possibilities of such mixtures develop, special care and investigation should be made.

- f) Ignition energies for many combustible dusts have been established. Some of the values have been quite low, which allow various types of static charges to be eligible to ignite such systems.
- g) Other than general atmospheric humidity, the moisture that is absorbed on a dust particle surface will usually raise its ignition temperature. Once an ignition has commenced, the humidity of the air has no effect.

### 9.83 EXTRAPOLATION/INTERPOLATION OF DUST NOMOGRAPHS

Use the technique described earlier for gases.

### 9.84 VENTING OF BINS, SILOS, AND HOPPERS

For elongated vessels and pipelines, explosions create axial flow [30] resulting in definite differences compared to explosions in more cubical vessels.

Vessels are considered elongated when the height ( $H$ )/diameter ( $D$ ) ratio  $> 5:1$  [30].

In elongated confined vessels, with one end closed and the opposite end open or removable, when an explosion begins at or near the closed end, the rapid movement of the flame front caused by the high volume from combustion will cause displacement of the unburnt mixture ahead of it. Apparently, this characteristic is independent of the nature of the combustible material [30], and the velocity can reach 80–90% of the flame velocity, in part due to the high turbulence generated in the unburnt mixtures.

For these types of vessels or pipeline (see [30] for details), explosion venting should always include the entire cross-sectional area of the top or roof area of a silo (for example), regardless of the volume. If the venting relief is placed on the sides, the entire top could be torn off.

#### EXAMPLE 9-22

A dust collector of 10 m<sup>3</sup> capacity is used to handle a dust with  $K_{st}$  value of 150 bar m/s. The static opening pressure of the vent panel,  $P_{stat}$ , is set at 1.1 bara (0.1 barg). Estimate the vent area required to limit the reduced explosion pressure to 1.3 bara (0.3 barg) using the nomograph and the regression analysis.

*Solution*

The appropriate nomograph is (Figure 9-71i):

Following the arrows on the dashed line gives a vent area of 0.97 m<sup>2</sup>.

Applying Simpson's regression analysis of the nomograph [76]

$$A_v = (a) (V^{2/3}) (K_{st})^b (P_{red})^c$$

where

$$a = 0.000571 e^{(2)(P_{stat})}$$

$$b = 0.978 e^{(-0.105)(P_{stat})}$$

$$c = -0.687 e^{(0.226)(P_{stat})}$$

$$A_v = \text{vent area, m}^2$$

with the units of  $P_{stat}$  and  $P_{red}$  in barg.

$$a = 0.000571 e^{(2 \times 0.1)}$$

$$= 6.974 \times 10^{-4}$$

$$b = 0.978 e^{(-0.105 \times 0.1)}$$

$$= 0.9677$$

$$c = -0.687 e^{(0.226 \times 0.1)}$$

$$= -0.7027$$

Therefore

$$A_v = (10^{2/3}) (6.974 \times 10^{-4}) (150^{0.9677}) (0.3^{-0.7027})$$

$$= 0.96 \text{ m}^2$$

**TABLE 9-35 Hazard Classes and  $K_{st}$  Values for Selected Types of Dusts**

Type of Dust	Material	Median Particle Size ( $\mu\text{m}$ )	Minimum Explosive Concentration ( $\text{g}/\text{m}^3$ )	$P_{\max}$ (barg)	$(dP/dt)_{\max}$ (barg)	$K_{st}$ (bar/ms)	Dust Hazard Class
Agricultural Products	Cellulose	33	60	9.7	229	229	2
	Cellulose, pulp	42	30	9.9	62	62	1
	Cork	42	30	9.6	202	202	2
	Corn	28	60	9.4	75	75	1
	Egg white	17	125	8.3	38	38	1
	Milk, powdered	83	60	5.8	28	28	1
	Milk, non-fat, dry	60	—	8.8	125	125	1
	Soy flour	20	200	9.2	110	110	1
	Starch, corn	7	—	10.3	202	202	2
	Starch, rice	18	60	9.2	101	101	1
	Starch, wheat	22	30	9.9	115	115	1
	Sugar	30	200	8.5	138	138	1
Chemical Dusts	Sugar, milk	27	60	8.3	82	82	1
	Sugar, beet	29	60	8.2	59	59	1
	Tapioca	22	125	9.4	62	62	1
	Whey	41	125	9.8	140	140	1
	Wood flour	29	—	10.5	205	205	2
	Adipic acid	<10	60	8.0	97	97	1
	Anthra-quinone	<10	—	10.6	364	364	3
	Ascorbic acid	39	60	9.0	111	111	1
	Calcium acetate	92	500	5.2	9	9	1
	Calcium acetate	85	250	6.5	21	21	1
	Calcium stearate	12	30	9.1	132	132	1
	Carboxy-methyl-cellulose	24	125	9.2	136	136	1
Carbonaceous Dusts	Dextrin	41	60	8.8	106	106	1
	Lactose	23	60	7.7	81	81	1
	Lead stearate	12	30	9.2	152	152	1
	Methyl-cellulose	75	60	9.5	134	134	1
	Paraform-aldehyde	23	60	9.9	178	178	1
	Sodium ascorbate	23	60	8.4	119	119	1
	Sodium stearate	22	30	8.8	123	123	1
	Sulfur	20	30	6.8	151	151	1
	Charcoal, activated	28	60	7.7	44	44	1
	Charcoal, wood	14	60	9.0	10	10	1
	Coal, bituminous	24	60	9.2	129	129	1
	Coke, petroleum	15	125	7.6	47	47	1
Metals and Plastics	Lampblack	<10	60	8.4	121	121	1
	Lignite	32	60	10.0	151	151	1
	Peat, 15% $\text{H}_2\text{O}$		58	60	10.9	157	1
	Peat, 22% $\text{H}_2\text{O}$		46	125	8.4	69	1
	Soot, pine	<10	—	7.9	26	26	1
	Aluminum	29	30	12.4	415	415	3
	Bronze	18	750	4.1	31	31	1
	Iron, Carbonyl	<10	125	6.1	111	111	1
	Magnesium	28	30	17.5	508	508	3
	Zinc	10	250	6.7	125	125	1
	Zinc	<10	125	7.3	176	176	1
Plastics	(poly) Acrylamide	10	250	5.9	12	12	1
	(poly) Acrylonitrile	25	—	8.5	121	121	1
	(poly) Ethylene (Low Pressure Process)	<10	30	8.0	156	156	1
	Epoxy Resin	26	30	7.9	129	129	1
	Melamine, Resin	18	125	10.2	110	110	1
	Melamine molded (Wood flour and Mineral filled Phenol-Formaldehyde)	15	60	7.5	41	41	1
	Melamine molded (Phenol-Cellulose)	12	60	10.0	127	127	1

(continued)

**TABLE 9-35—(continued)**

Type of Dust	Material	Median Particle Size ( $\mu\text{m}$ )	Minimum Explosive Concentration ( $\text{g}/\text{m}^3$ )	$P_{\max}$ (barg)	$(dP/dt)_{\max}$ (barg)	$K_{st}$ (bar/ms)	Dust Hazard Class
Metal Dusters	(poly) Methyl Acrylate	21	30	9.4	269	269	2
	(poly) Methyl Acrylate, Emulsion Polymer	18	30	10.1	202	202	2
	Phenolic Resin	<10	15	9.3	129	129	1
	(poly) Propylene Terpene-Phenol Resin	25	30	8.4	101	101	1
	Urea Formaldehyde/Cellulose, Molded	10	15	8.7	143	143	1
	(poly) Vinyl Acetate/Ethylene Copolymer	13	60	10.2	136	136	1
	(poly) Vinyl Alcohol	32	30	8.6	119	119	1
	(poly) Vinyl Butyral	26	60	8.9	128	128	1
	(poly) Vinyl Chloride	65	30	8.9	147	147	1
	(poly) Vinyl Chloride/Vinyl Acetylene Emulsion Copolymer	107	200	7.6	46	46	1
	(poly) Vinyl chloride/Ethylene/Vinyl Acetylene Suspension Copolymer	35	60	8.2	95	95	1
	(poly) Vinyl chloride/Ethylene/Vinyl Acetylene Suspension Copolymer	60	60	8.3	98	98	1

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### 9.85 SIZING GUIDELINES (SEE [30] FOR DETAILS)

1. The relief area should never be less than that determined from the nomograms. A silo's cross-sectional area establishes the maximum that can be protected.
2. The nomograms are applicable only for vessels with volumes up to  $1000 \text{ m}^3$  ( $35,315 \text{ ft}^3$ ).
3. Using the limiting relief area as the cross-sectional area of an elongated vessel,  $A_2$ , the greatest volume,  $V_2$ , that can be protected by this relief area can be calculated by the cubic law [30]:

$$V_2 = V_1 \sqrt{(A_2/A_1)^3} \quad (9-88)$$

The required relief area,  $A_1$ , for a volume  $V_1 = 1 \text{ m}^3$  obtained from the nomograms. The same reduced explosion pressure,  $P_{\text{red}}$ , and same static activation pressure,  $P_{\text{stat}}$ , of the relief device are the same for both volumes  $V_1$  and  $V_2$ , and therefore constant. As the mechanical strength of the vessel,  $P_{\text{red}}$ , changes, the maximum volume and height will change with the hazard class, St-1, St-2, or St-3.

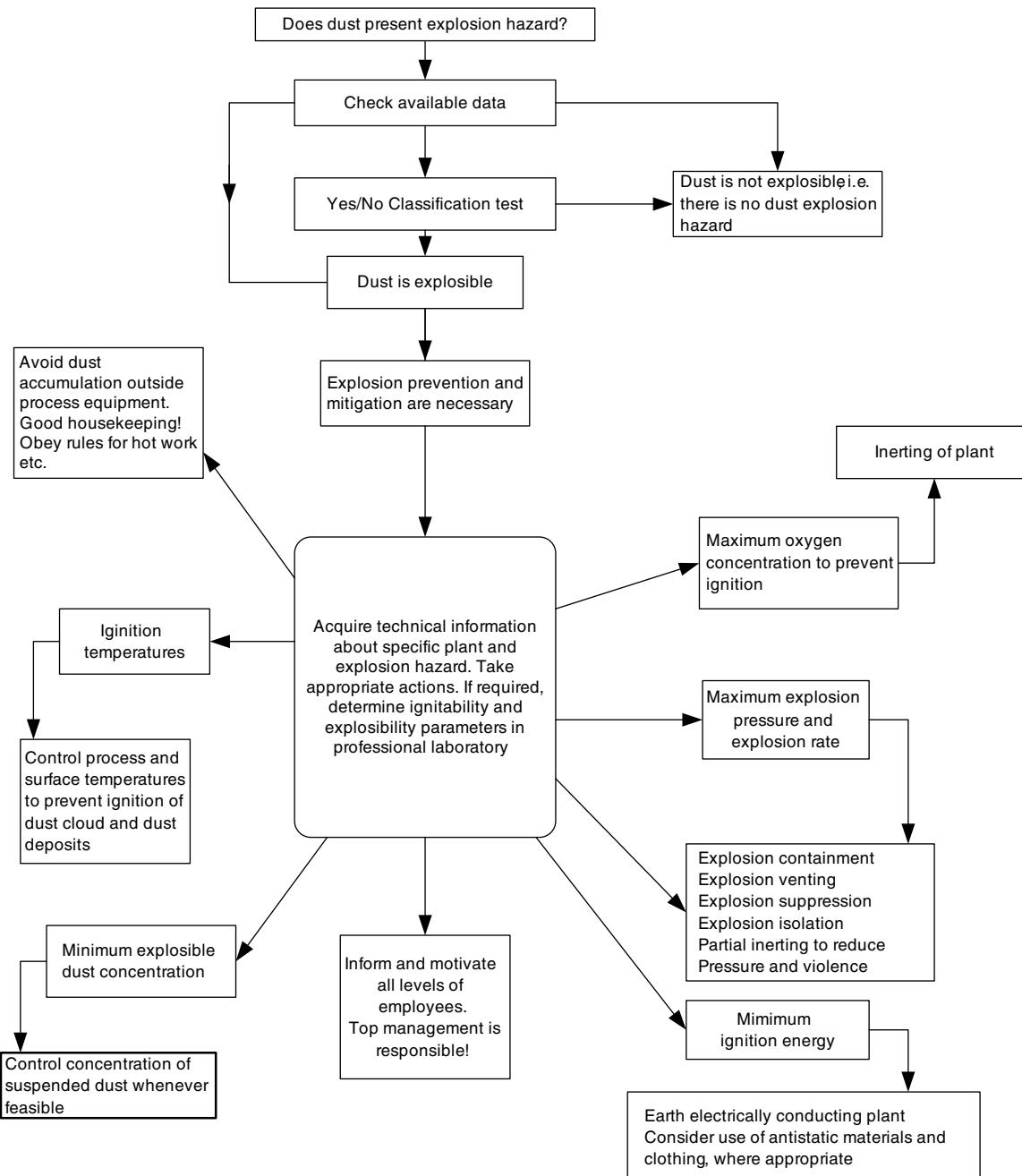
### 9.86 SECONDARY DUST EXPLOSIONS IN BUILDINGS

Secondary dust explosions occur when the primary explosion causes a breach in equipment or in a weak enclosure, and the associated blast wave creates a suspended dust cloud that is ignited by

the vented flame from the primary explosion or other local ignition source. The ruptured equipment or conveyor lifts the accumulated dust into suspension, and the flame from the primary explosion subsequently ignites the suspension dust cloud. This causes further devastation with the structural damage to the building, and intolerable casualties.

Zalosh et al. [69] have illustrated examples of secondary dust explosions, and Figure 9-76 shows an aerial view of the destruction in the dust explosion at West Pharmaceuticals Kinston, North Carolina plant in the US. The plant manufactured rubber components for drug vials and syringes. Polyethylene powder mixed with water was used to reduce the stickiness of the rubber. The US Chemical Safety and Hazard Investigation Board (CSB) determined that the fuel for the explosion was accumulated polyethylene dust above a suspended ceiling which was dispersed either by some undefined primary explosion or by some unidentified disturbance. The polyethylene dust cloud produced a deflagration in the space between the ceiling tiles and the concrete floor above.

Implementation and maintenance practices of an appropriate good housekeeping should prevent or minimize emissions and accumulation of particulate solids that can lead to secondary explosions where processing equipment is located in a building or other equipment. All equipment and piping that leak should be tightened up. Capture and removal equipment that transport leaking dust to safe location should be provided at dust escape points. FM Data sheet 7-76 [69] and NFPA 654 [81] provide additional housekeeping and maintenance recommendations.



**Figure 9-72** A general approach to practical dust explosion protection. (From R.K. Eckhoff [68].)

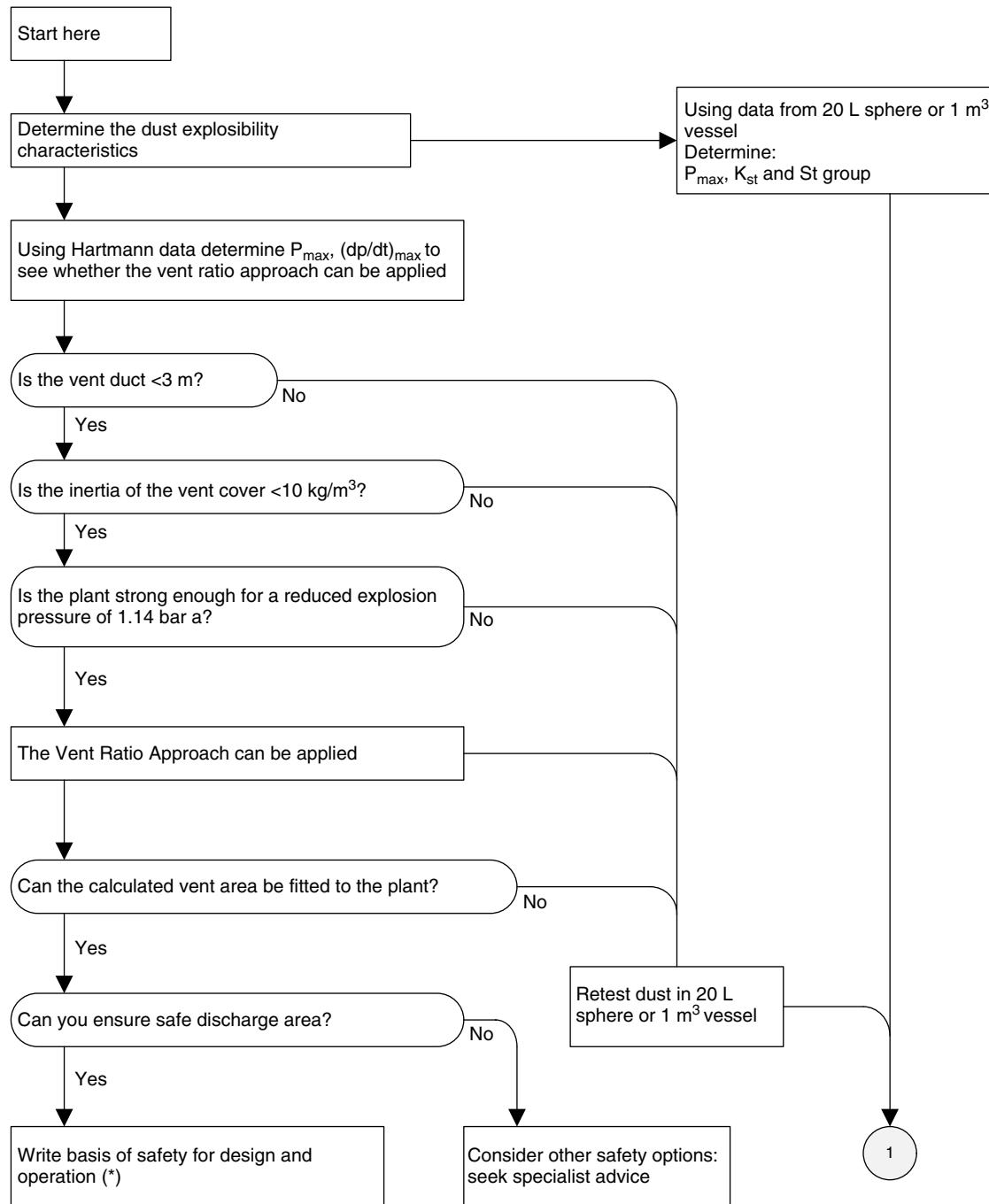
### 9.87 DUST CLOUDS

These can be readily ignited by flames, sparks, static electrical discharges (often the most likely), hot surfaces, and many other sources. Table 9-36 lists dust cloud ignition temperatures ranging from 572 to 1112°F, and can be contrasted to flammable vapor-air ignition temperatures from 428 to 1170°F. Generally, ignition temperatures and energies required for a dust explosion are lower than many common sources of ignition. Hence, caution must be used in handling dusting materials [6] (also see [82]).

### 9.88 DUST EXPLOSION SEVERITY

Table 9-36 lists the explosibility index that is a relative measure of the potential damage from a dust explosion. A rating of 2 to 4 requires large vent areas. Above 4, for most cases, the explosion cannot be controlled by venting design and therefore requires the use of protection such as inert gas or explosive suppression systems, some of which are commercially available.

Unfortunately, the rate of pressure rise and maximum explosion pressure listed in Table 9-36 are subject to uniqueness of the



**Figure 9-73a** Logic diagram for venting decisions. (Lunn, G., *Dust Explosion Prevention and Protection*, Part 1-Venting, 2nd ed., IChemE, 1992, Courtesy of the IChemE, UK, All rights reserved.)

\* Formal definition of basis of safety – All practicable steps should be taken to avoid hazards. It is therefore strongly recommended that the basis of safety for the design should be formally recorded.

test conditions and are the function of particle size, dust concentration and uniformity, available ignition energy, and moisture content. Therefore, the data presented are useful for reference, but cannot be counted as absolute. For serious design, actual tests should be performed on the dust by qualified laboratories using standardized test equipment. See illustration in [6], pp. 4:94–95.

## 9.89 PREVENTING, MITIGATING, AND PROTECTION AGAINST DUST EXPLOSIONS

Various studies have been conducted in the explosion of dusts, and means of preventing, mitigating, and protection against dust explosions have been well documented [68, 81, and 83–90]. Table 9-37 gives an overview of the various means that are

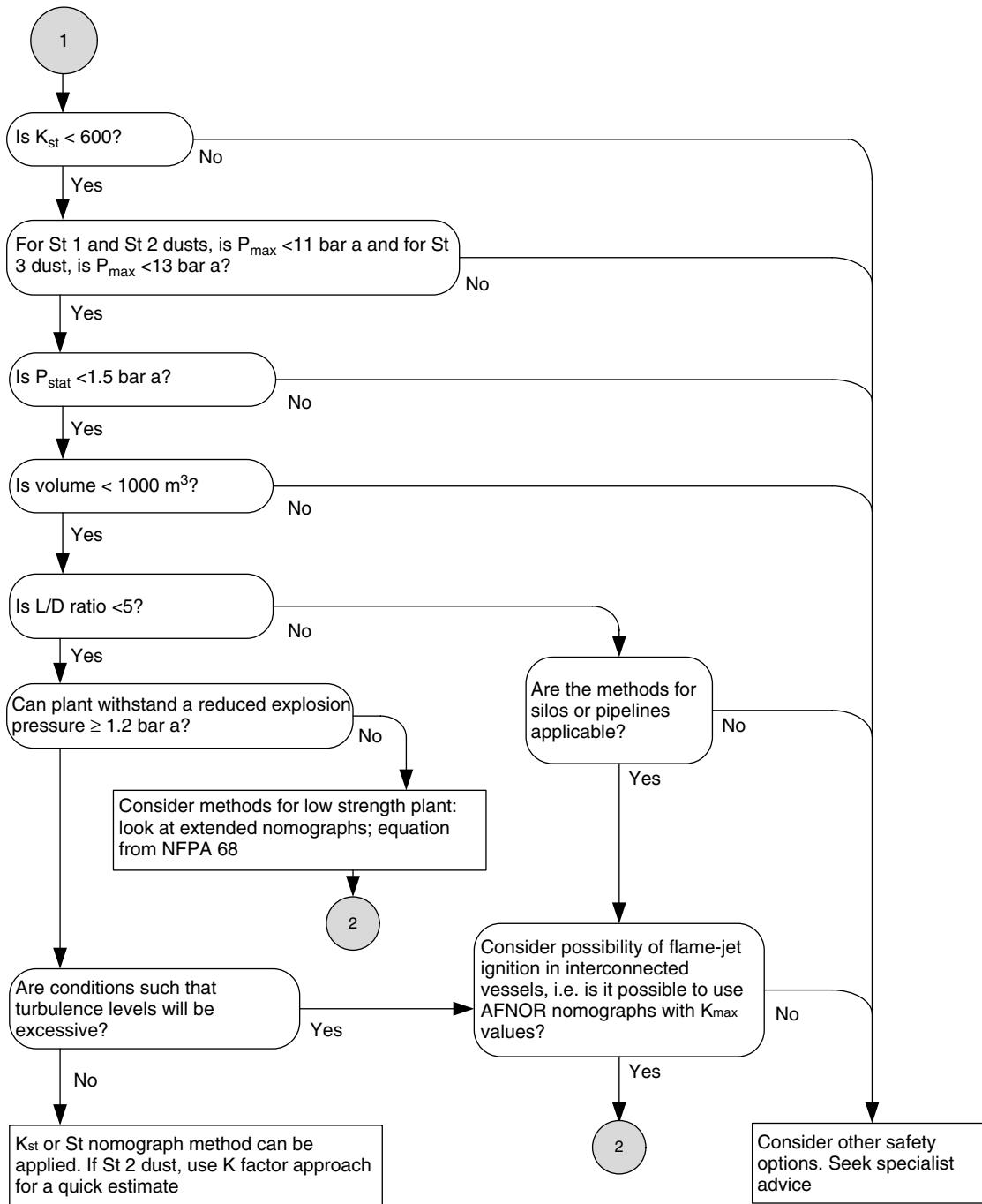


Figure 9-73b—(continued)

presently in use. Eckhoff [90] has shown how research on relevant fundamental phenomena can promote further development of practical means for preventing and mitigating dust explosions in industry.

Explosion protection incorporates the measures being implemented against explosion hazards in the handling of combustible substances and the assessment of the effectiveness of protective measures of either avoiding or reducing these hazards. The explosion protection concept is valid for all mixtures of combustible substances and distinguishes between the following [68].

1. Measures which prevent or restrict formation of a hazardous, explosive atmosphere.
2. Measures which prevent the ignition of a hazardous, explosive atmosphere.
3. Constructional measures which limit the effects of an explosion to harmless level.

Priority must be given to the measures in (1) from safety viewpoint. Measures in (2) cannot be used as a sole protective measure for flammable gas or solvent vapors in industrial practice with

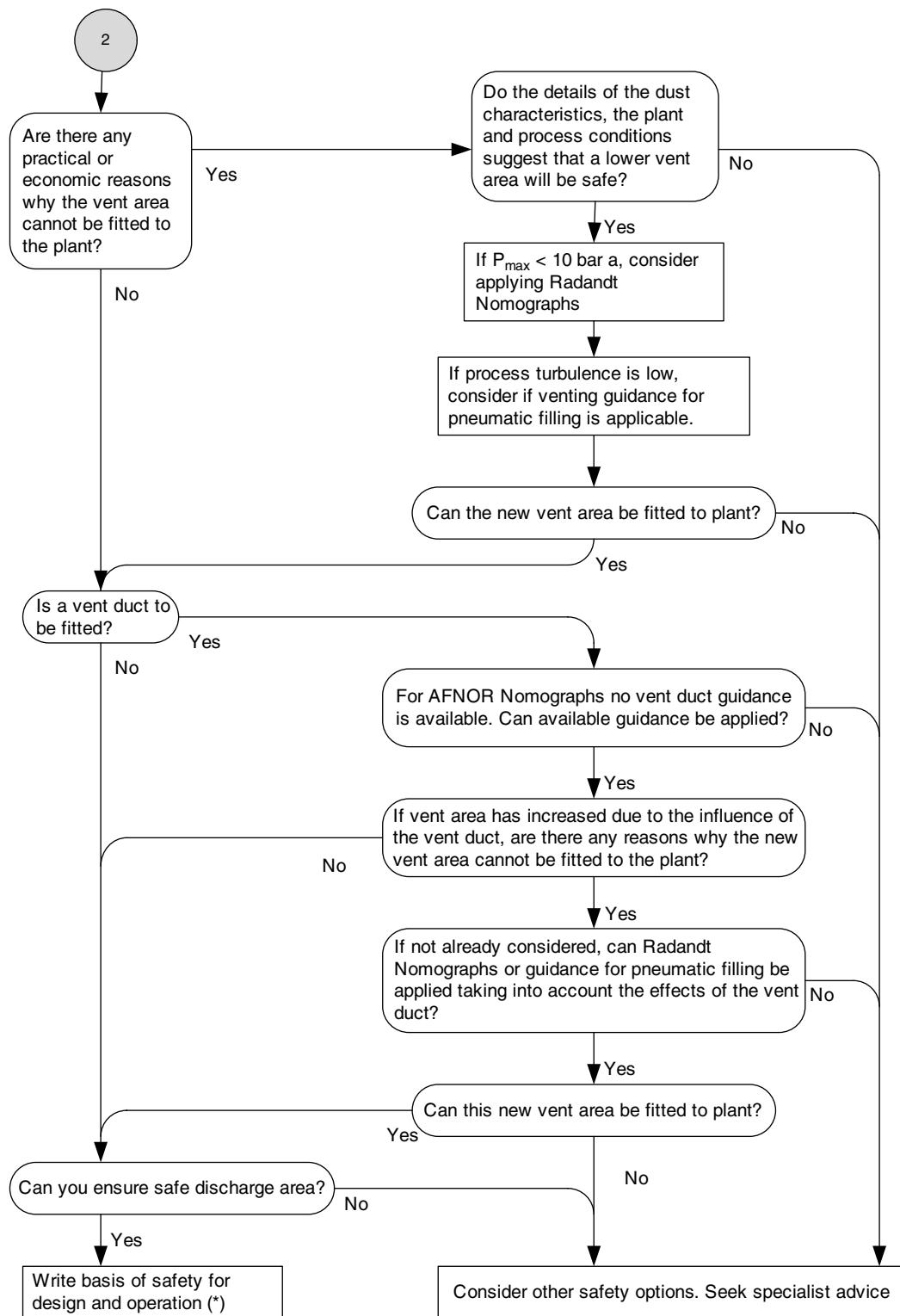
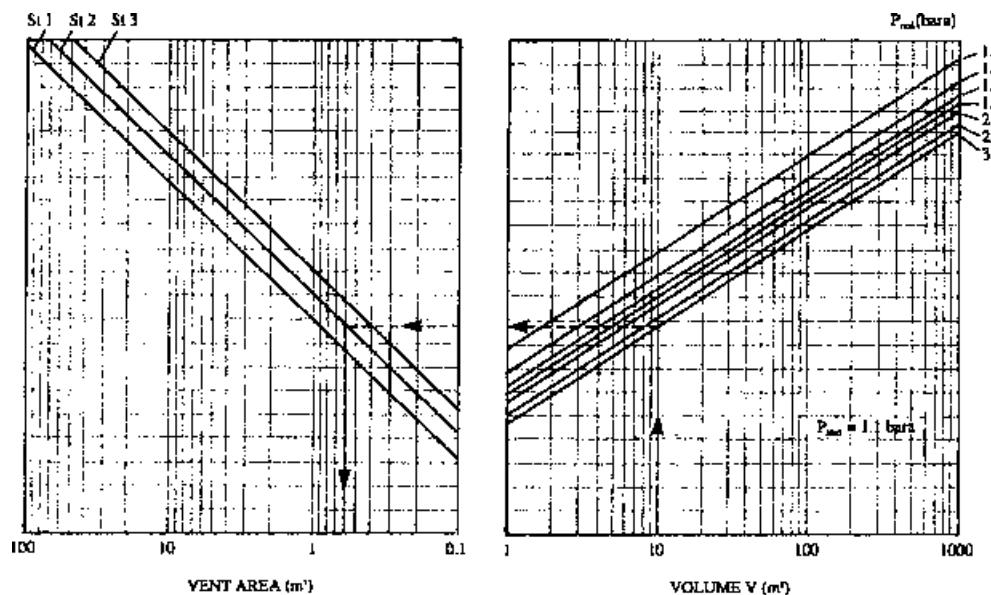
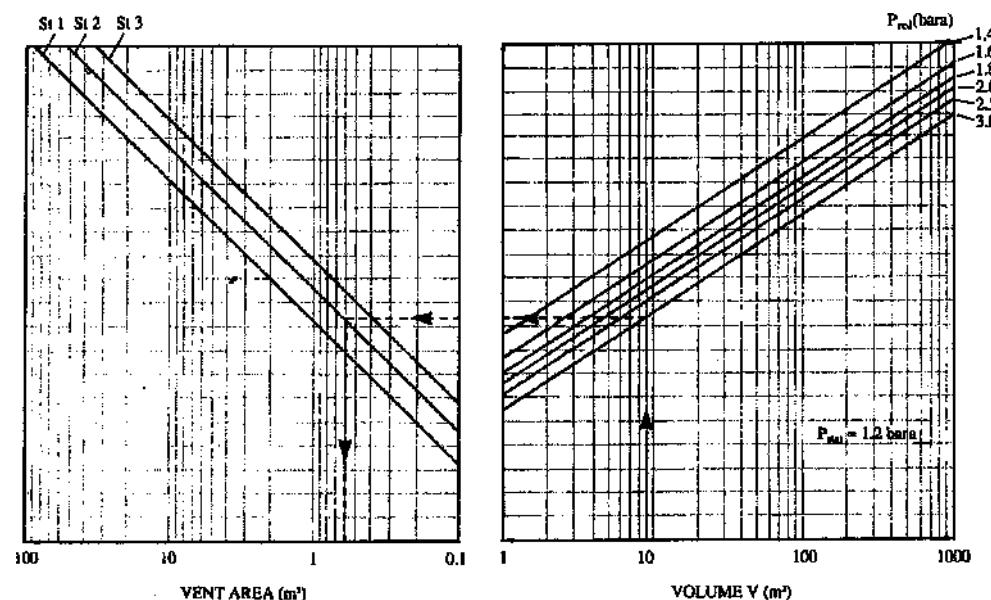


Figure 9-73c—(continued)



**Figure 9-74a** Explosion relief venting: dusts. (From Guideline VDI 3673 (Fig. 7a–c). Printed with permission from the Verien Deutscher Ingenieure, Dusseldorf, Germany.)



**Figure 9-74b** Explosion relief venting: dusts. (From Guideline VDI 3673 (Fig. 7a–c). Printed with permission from the Verien Deutscher Ingenieure, Dusseldorf, Germany.)

sufficient reliability. However, it can be applied as the sole protective measure when only combustible dusts are present if the minimum ignition energy of the dusts is high ( $>10\text{mJ}$ ) and the opening areas concerned can easily be monitored. If the measures under (1) and (2), which are referred to as preventive measures, cannot be applied with sufficient reliability, the constructional measures of (3) must be applied.

## 9.90 PREVENTIVE EXPLOSION PROTECTION

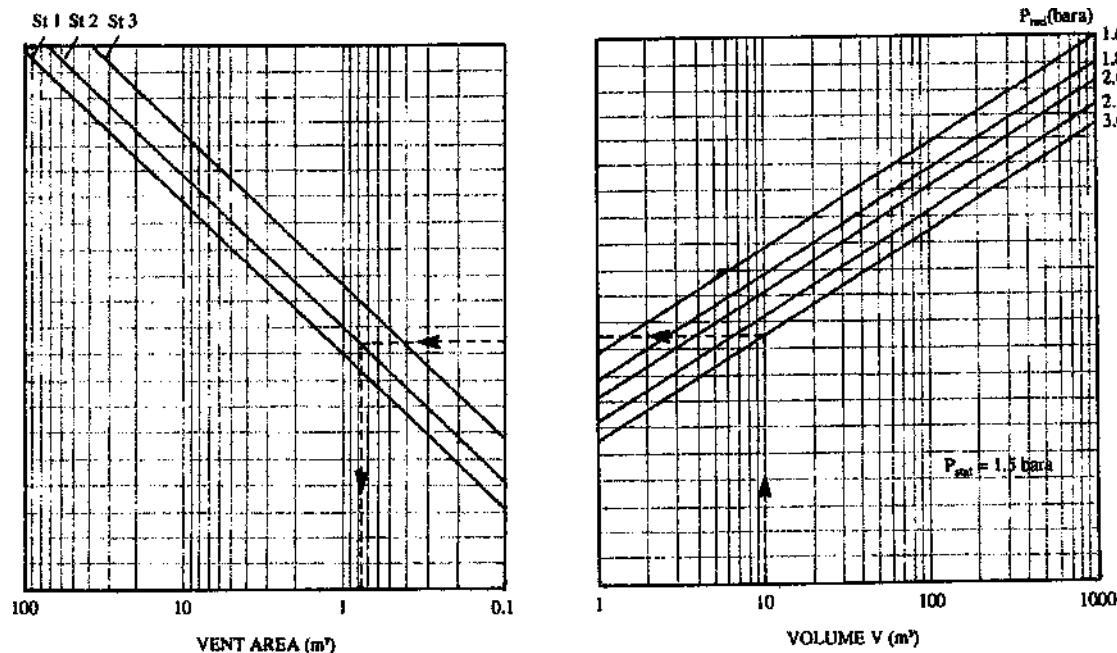
The principle of preventive explosion protection involves the exclusion of one of the requirements necessary for the development of

an explosion. If one of the three sources of a fire hazard is broken, an explosion can be excluded. This is achieved by

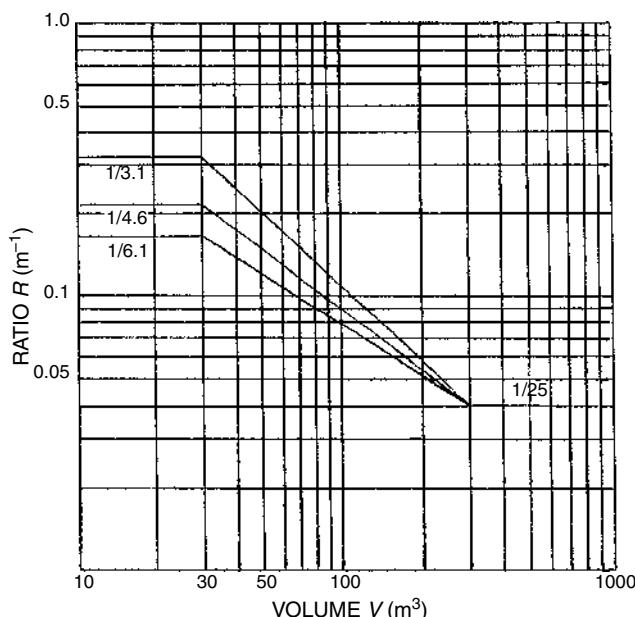
- avoiding the development of explosive mixtures
- replacing the atmospheric oxygen by inert gas, working in a vacuum or using inert dust
- preventing the occurrence of effective ignition source.

## 9.91 EXPLOSION SUPPRESSION

Protection to guard against explosion has been classed as venting, suppression, and isolation. Bartknecht [30] discusses the subject rather thoroughly and [6] discusses some of the methods used to



**Figure 9-74c** Explosion relief venting: dusts. (From Guideline VDI 3673 (Fig. 7a–c). Printed with permission from the Verien Deutscher Ingenieure, Dusseldorf, Germany.)



**Figure 9-75** Vent area approach. (From G. Lunn [67], by permission from the IChemE, all rights reserved.)

suppress explosions (see Figures 9-61, 9-62, 9-77, and 9-78). After an explosion has started special sensors are required, such as

1. thermoelectrical
2. optical
3. pressure.

These require special electronics to cause responses of the systems to actually inject the suppression medium, such as [30]

1. halons (halogenated hydrocarbons), see Figure 9-62
2. water, primarily for dust explosion
3. powders
4. ammonium phosphate for some organic peroxide.

Special explosion characteristics must be understood before selecting the type of suppression medium. Table 9-38 classifies the precautions against dust explosions into prevention and protection.

For pipelines, bursting disks have been proven practical, especially when equipped with a sensor to pick up the explosion and a detonator to rupture the disks in advance of the pressure wave. The installation of a moveable explosion door carefully designed and weighted can also prove useful for pipelines.

The use of properly designed relief panels or “free floating” vessel covers is usually effective for dust explosions in silos tanks, filter housings, and the like. The design basis has been previously discussed.

A very careful study is required to develop a confidence for deflagration suppression in any flammable or dust system. The potential damage can be enormous.

The Fauske and Associates’ (F & AI) *Reactive System Screening Tool* (RSST) was developed as a result of the DIERS studies and allows rapid evaluation of the potential for runaway reactions. It measures the rate of energy and gas release during the runaway and is valuable for screening various process systems before commercial designs are completed (see Figure 9-61). Fauske and Associates have recently improved on RSST with the development of an Advanced Reactive System Screening Tool (ARSST), which contains enhanced features (Figures 9-85a–c). A listed website at the end of the chapter gives further information on dust explosion hazards and two-phase reactive systems. A particular website is [www.fauske.com](http://www.fauske.com).



**Figure 9-76** Aerial view of the destruction and residual fire in dust explosion at West Pharmaceuticals. (From the US Chemical Safety Board report.)

### 9.92 UNCONFINED VAPOR CLOUD EXPLOSIONS (UVCE)

These explosions in air are usually the result of the release of flammable gas and/or mists by leaks, rupture of equipment, or rupture of safety relieving devices and release to the atmosphere, which become ignited by spark, static electricity, hot surfaces, and many other potential sources. They usually spread up into the air above the plant area and/or equipment. When the concentration is right, they ignite to form a deflagration and often a detonation that results in heavy damage. From some of these incidents, plant areas have been destroyed, glass broken up to or beyond 5 mi away, and people severely injured or killed.

These clouds can travel with the prevailing wind currents and thereby explode considerable distances from the initial discharge of vapors. Reference [51] presents a rather thorough review of the history and theoretical analysis of these types of explosions.

Atmospheric releases may form relatively still clouds, or they may plume and trail with the wind, or they may jet high into the atmosphere before forming a cloud, all depending to some extent on the unit. Wells [60] presents a thorough analysis of these phenomena.

### 9.93 EFFECTS OF VENTING DUCTS

Usually the relief of explosions cannot readily, safely, or conventionally be released right at the source, whether in a building or in a working plant area. Therefore, these reliefs are directed to some discharge point where the released material (dusts, gases, or mixtures), which may be very hot, can be safely released.

First and foremost, these venting ducts should be as straight as possible, with few, if any elbows, and even these should be sweeping bends. There should be no valve of any type to keep flow resistance as low as possible, as this creates friction that creates

back pressure on the relief device and raises burst conditions, which can be terribly dangerous. Figures 9-77 and 9-78 are used to assess the increased pressure due to ducts on relief discharge as affected by the duct length. (This data is limited. See [60, 91].)

### 9.94 MAXIMUM DISTANCE BETWEEN VENTS

Figure 9-79 indicates the maximum allowable distance between vents on a vessel or pipe related to vent diameter when multiple vents are required. When distances are greater than indicated, a detonation should be anticipated in the design of the equipment strength. Figure 9-79 is applicable for systems with operating pressures up to 0.2 barg and for systems such as elongated vessels, pipes, and ducts with dusts or gases that are vented at one end, and have a velocity of less than 2 m/s. Above 2 m/s, an alternative protection is recommended because a detonation is likely [10, 9]. See NFPA-68 [10] for details of application.

### 9.95 RUNAWAY REACTIONS: DIERS

One of the standardized methods for predicting or controlling runaway reaction that may lead to explosions (deflagration or detonations) is the Fauske approach (Figure 9-61). Others are presented elsewhere [92].

Accordingly, to emphasize the safety problems affecting all industrial process plants and laboratories, the AIChE established the industry-supported DIERS. The purposes of the Institute are as follows [22].

- Reduce the frequency, severity, and consequences of pressure producing accidents.
- Promote the development of new techniques that will improve the design of emergency relief systems.

**TABLE 9-36** Explosion Characteristics of Various Dusts (Partial Listing)

(Compiled from the following reports of the U.S. Department of Interior, Bureau of Mines: RI 5753, The Explosibility of Agricultural Dusts; RI 6516, Explosibility of Metal Powders; RI 5971, Explosibility of Dusts Used in the Plastic Industry; RI 6597, Explosibility of Carbonaceous Dusts; RI 7132, Dust Explosibility of Chemicals, Drugs, Dyes and Pesticides; and RI 7208, Explosibility of Miscellaneous Dusts).

Type of Dust	Explosibility Index	Ignition Sensitivity	Explosion Severity	Maximum Explosion Pressure (psig)	Max Rate of Pressure Rise (psi/s)	Ignition Temperature		Min Cloud Ignition Energy (J)	Min Explosion Concentration (oz/ft <sup>3</sup> )	Limiting Oxygen Percentage* (Spark Ignition)
						Cloud (°C)	Layer (°C)			
<i>Agricultural Dusts</i>										
Alfalfa meal	0.1	0.1	1.2	66	1,100	530	—	0.32	0.105	—
Almond shell	0.3	0.9	0.3	101	1,400	450	210	0.08	0.065	—
Apricot pit	1.9	1.6	1.2	109	4,000	440	230	0.08	0.035	—
Cellulose	2.8	1.0	2.8	130	4,500	480	270	0.080	0.055	C13
Cellulose, alpha	>10	2.7	4.0	117	8,000	410	300	0.040	0.045	—
Cellulose, flock, fine cut	8.7	2.3	3.8	112	7,000	460	260	0.035	0.055	C13
Cereal grass	<0.1	<0.1	0.1	65	400	620	230	0.80	0.20	—
Cherry pit	4.4	2.0	2.2	113	4,400	430	220	0.08	0.03	—
Cinnamon	5.8	2.5	2.3	121	3,900	440	230	0.03	0.06	—
Citrus peel	0.6	0.7	0.9	51	1,200	500	330	0.10	0.06	—
Coca bean shell	13.7	3.6	3.8	77	3,300	470	370	0.03	0.04	—
Cocoa, natural 19% fat	0.6	0.5	1.1	68	1,200	510	240	0.10	0.075	—
Coconut shell	4.2	2.0	2.1	115	4,200	470	220	0.06	0.035	—
Coffee, raw bean	<0.1	0.1	0.1	33	150	650	280	0.32	0.15	C17
Coffee, fully roasted	<0.1	0.2	0.1	38	150	720	270	0.16	0.085	C17
Coffee, instant spray dried	<0.1	0.1	0.1	68	500	410	350	†	0.28	—
Corn	6.9	2.3	3.0	113	6,000	400	250	0.04	0.055	—
Corncob grit	5.5	2.5	2.2	127	3,700	450	240	0.045	0.045	—
Corn dextrin, pure	12.1	3.1	3.9	124	5,500	410	390‡	0.04	0.04	—
Cornstarch commercial product	9.5	2.8	3.4	106	7,500	400	—	0.04	0.045	—
Cornstarch (thru No. 325 Sieve)	23.2	4.3	5.4	145	9,500	390	350	0.03	0.04	C11
Cork dust	>10	3.6	3.3	96	7,500	460	210	0.035	0.035	—
Cotton linter, raw	<0.1	<0.1	<0.1	73	400	520	—	1.92	0.50	C21

(continued)

TABLE 9-36—(continued)

Type of Dust	Explosibility Index	Ignition Sensitivity	Explosion Severity	Maximum Explosion Pressure (psig)	Max Rate of Pressure Rise (psi/s)	Ignition Temperature		Min Cloud Ignition Energy (J)	Min Explosion Concentration (oz/ft <sup>3</sup> )	Limiting Oxygen Percentage* (Spark Ignition)
						Cloud (°C)	Layer (°C)			
Cottonseed meal	1.1	0.9	1.2	104	2,200	540	—	0.08	0.055	—
Cube root, South American	6.5	2.7	2.4	69	2,100	470	230	0.04	0.04	—
Egg white	<0.1	<0.1	0.2	58	500	610	—	0.64	0.14	—
Flax shive	0.2	0.7	0.3	108	1,500	430	230	0.08	0.08	—
Garlic, dehydrated	0.2	0.2	1.2	57	1,300	360	—	0.24	0.10	—
Grain dust, winter wheat corn oats	9.2	2.8	3.3	131	7,000	430	230	0.03	0.055	—
Grass seed, blue	<0.1	0.1	0.1	51	400	490	180	0.26	0.29	—
Guar seed	2.4	1.7	1.4	70	1,200	500	—	0.06	0.04	—
Gum, Arabic	1.1	0.7	1.6	84	1,500	500	260	0.10	0.06	—
Gum, karaya	0.3	0.2	1.5	83	1,100	520	240	0.18	0.10	—
Gum, Manila (copal)	18.0	6.2	2.9	63	2,800	360	390‡	0.03	0.03	—
Gum, tragacanth	8.1	2.6	3.1	88	2,400	490	260	0.045	0.04	—
Hemp hurd	20.5	3.8	5.4	121	10,000	440	220	0.035	0.04	—
Lycopodium	16.4	4.2	3.9	75	3,100	480	310	0.04	0.025	C13
Malt barley	5.5	2.6	2.1	95	4,400	400	250	0.035	0.055	—
Milk, Skimmed	1.4	1.6	0.9	95	2,300	490	200	0.05	0.05	N15
Moss, Irish	<0.1	<0.1	<0.1	35	400	480	230	†	§	—
Onion, dehydrated	<0.1	<0.1	<0.1	35	500	410	—	†	0.13	—
Pea flour	4.0	1.8	2.2	68	1,900	560	260	0.04	0.05	—
Peach pit shell	7.1	3.1	2.3	115	4,700	440	210	0.05	0.03	—
Peanut hull	4.0	2.0	2.0	116	8,000	460	210	0.05	0.045	—
Peat, sphagnum, sun dried	2.0	2.0	1.0	104	2,200	460	240	0.05	0.045	—
Pecan nut shell	7.4	3.1	2.4	112	4,400	440	210	0.05	0.03	—
Pectin (from ground dried apple pulp)	10.3	2.2	4.7	132	8,000	410	200	0.035	0.075	—
Potato starch, dextrinized	20.9	5.1	4.1	120	8,000	410	—	0.025	0.045	—

TABLE 9-36—(continued)

Type of Dust	Explosibility Index	Ignition Sensitivity	Explosion Severity	Maximum Explosion Pressure (psig)	Max Rate of Pressure Rise (psi/s)	Ignition Temperature		Min Cloud Ignition Energy (J)	Min Explosion Concentration (oz/ft <sup>3</sup> )	Limiting Oxygen Percentage* (Spark Ignition)
						Cloud (°C)	Layer (°C)			
Pyrethrum, ground flower leaves	0.4	0.6	0.6	95	1,500	460	210	0.08	0.10	—
Rauwolfia vomitoria root	9.2	2.2	4.2	106	7,500	420	230	0.045	0.055	—
Rice	0.3	0.5	0.5	47	700	510	450	0.10	0.085	—
Rice bran	1.4	1.1	1.3	61	1,300	490	—	0.08	0.045	—
Rice hull	2.7	1.6	1.7	109	4,000	450	220	0.05	0.055	—
Safflower meal	5.2	4.0	1.3	90	2,400	460	210	0.025	0.055	—
Soy flour	0.7	0.6	1.1	94	800	550	340	0.10	0.06	C15
Soy protein	4.0	1.2	3.3	98	6,500	540	—	0.06	0.05	C15
Sucrose, chemically pure	3.3	1.1	3.0	76	2,500	420	470†	0.10	0.045	—
Sucrose	4.8	2.7	1.8	86	5,500	370	400†	0.03	0.045	—
Sugar, powdered	9.6	4.0	2.4	109	5,000	370	400†	0.03	0.045	—
1 Naphthyl-N-methylcarbamate ("Sevin") 15% (85% Inert)	>10	18.0	1.6	90	5,000	560	140	0.010	0.020	—
3,4,5,6-tetrahydro-3,5-dimethyl-2H-1,3,5-thiadeazine 2 thione(Crag" No. 974) 5% (95% Inert)	>10	8.7	2.0	97	6,000	310	330	0.030	0.025	—
a, a' Trithiobis (N,N-dimethylthioformamide)	8.9	3.4	2.6	96	7,000	280	230	0.035	0.060	—

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(continued)

TABLE 9-36—(continued)

Type of Dust	Explosibility Index	Ignition Sensitivity	Explosion Severity	Maximum Explosion Pressure (psig)	Max Rate of Pressure Rise (psi/s)	Ignition Temperature		Min Cloud Ignition Energy (J)	Min Explosion Concentration (oz/ft <sup>3</sup> )	Limiting Oxygen Percentage* (Spark Ignition)								
						Cloud (°C)	Layer (°C)											
<i>Thermoplastic Resins and Molding Compounds</i>																		
Group I. Acetal Resins																		
Acetal, linear (Poly-formaldehyde)	>10	6.5	1.9	113	4,100	440	—	0.020	0.035	C11								
Group II, Acrylic Resins																		
Methyl methacrylate polymer	6.3	7.0	0.9	84	2,000	480	—	0.020	0.030	C11								
Methyl methacrylate-ethyl acrylate copolymer	>10	14.0	2.7	85	6,000	480	—	0.010	0.030	C11								
Methyl methacrylate-ethyl acrylate styrene copolymer	>10	9.2	1.7	90	4,400	440	—	0.02	0.025	C11								
Methyl methacrylate-styrene-butadiene-acrylonitrile copolymer	>10	8.4	1.4	87	4,700	480	—	0.020	0.025	C11								
Methacrylic acid polymer, modified	0.6	1.0	0.6	97	1,800	450	290	0.100	0.045	—								
Acrylamide polymer	>2.5	4.1	0.6	85	2,500	410	240	0.030	0.040	—								
Acrylonitrile polymer	>10	8.1	2.3	89	11,000	500	460	0.020	0.025	C13								
Acrylonitrile-vinyl pyridine copolymer	>10	7.9	2.4	85	6,000	510	240	0.025	0.020	—								

(continued)

TABLE 9-36—(continued)

Type of Dust	Explosibility Index	Ignition Sensitivity	Explosion Severity	Maximum Explosion Pressure (psig)	Max Rate of Pressure Rise (psi/s)	Ignition Temperature		Min Cloud Ignition Energy (J)	Min Explosion Concentration (oz/ft <sup>3</sup> )	Limiting Oxygen Percentage* (Spark Ignition)
						Cloud (°C)	Layer (°C)			
Acrylonitrile-vinyl chloride-vinylidene chloride copolymer (70-20-10)	>10	5.9	3.0	87	15,000	650	210	0.015	0.035	—
<b>Group III, Cellulosic Resins</b>										
Cellulose acetate	>10	8.0	1.6	85	3,600	420	—	0.015	0.040	C14
Cellulose triacetate	7.4	3.9	1.9	107	4,300	430	—	0.030	0.040	C12
Cellulose acetate butyrate	5.6	4.7	1.2	85	2,700	410	—	0.030	0.035	C14
Cellulose propionate, 0.3% free hydroxyl	7.5	2.9	2.6	107	4,700	460	—	0.060	0.025	—
Ethyl cellulose 5-10 micron dust	>10	21.8	3.4	120	6,500	370	350 <sup>††</sup>	0.010	0.025	C12
Methyl cellulose	>10	9.3	3.1	133	6,000	360	340	0.020	0.030	C13
Carboxy methyl cellulose, low viscosity, 0.3 to 0.4% substitution, acid product	1.4	0.5	2.7	130	5,000	460	310	0.140	0.060	—
Hydroxyethyl cellulose-mono sodium phosphate sizing compound	1.7	2.1	0.8	110	4,000	390	340	0.035	0.070	—

(continued)

TABLE 9-36—(continued)

Type of Dust	Explosibility Index	Ignition Sensitivity	Explosion Severity	Maximum Explosion Pressure (psig)	Max Rate of Pressure Rise (psi/s)	Ignition Temperature		Min Cloud Ignition Energy (J)	Min Explosion Concentration (oz/ft <sup>3</sup> )	Limiting Oxygen Percentage* (Spark Ignition)
						Cloud (°C)	Layer (°C)			
Group IV, Chlorinated Polyether Resins										
Chlorinated polyether alcohol	0.2	0.6	0.3	88	1,900	460	—	0.160	0.045	—
Group V, Fluoro-carbon Resins										
Tetrafluoro-ethylene polymer (micronized)	0.1**	0.1**	—	††	—	670	570 <sup>‡</sup>	‡	§	—
Mono-chlortrifluoroethylene polymer	0.1**	0.1**	—	††	—	600	720 <sup>‡</sup>	‡	§	—
Group VI, Nylon (Polyamide) Resins										
Nylon (polyhexamethylene adipamide) polymer	>10	6.7	1.8	95	4,000	500	430	0.020	0.030	C13
Group VII, Polycarbonate Resin										
Polycarbonate	8.6	4.5	1.9	96	4,700	710	—	0.025	0.025	C15

Reprinted by permission from *Fire Protection Handbook*, 17th ed. (1991), pp. 4–85, National Fire Protection Association [6].

\* Numbers in this column indicate oxygen percentage while letter prefix indicates the diluent gas. For example, the entry "C13" means dilution to oxygen content of 13% with carbon dioxide as the diluent gas. The letter prefixes are: C = Carbon dioxide; N = Nitrogen; A = Argon; and H = Helium.

† No ignition to 8.32 Joules, the highest tried.

‡ Ignition denoted by flame, all others not so marked (‡) denoted by a glow.

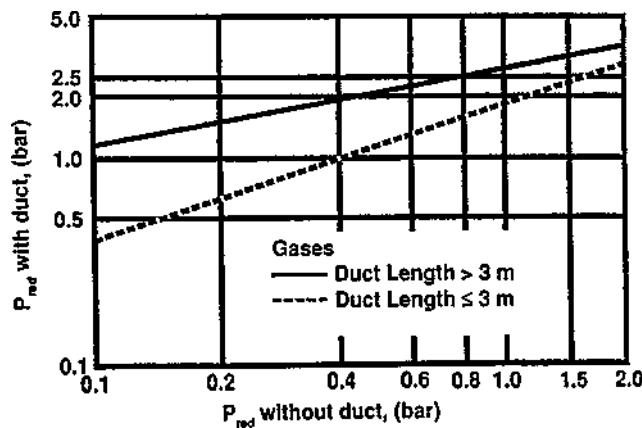
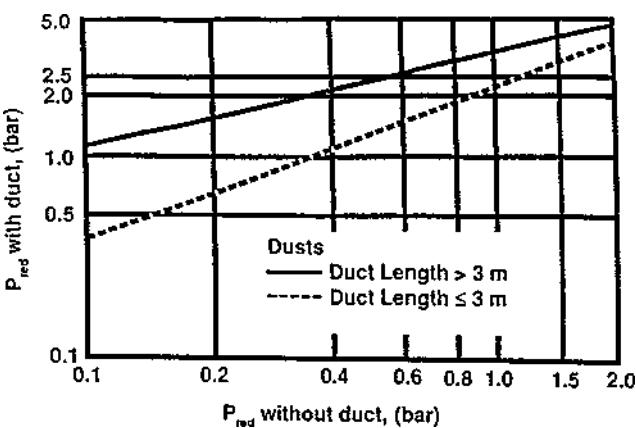
§ No ignition to 2 oz. per cu ft., the highest tried

\*\*0.1 designates materials presenting a fire ignition of the dust cloud is not obtained by the spark of flame source but only by the intense heated surface.

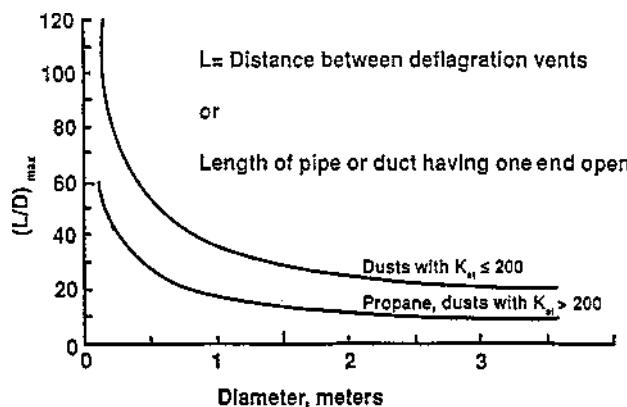
†† No ignition to 8.32 J, the highest tried

**TABLE 9-37 Means of Preventing and Mitigating Dust Explosions: A Schematic Overview**

Prevention		
Preventing Ignition Sources	Preventing Explosive Dust Clouds	Mitigation
Smoldering combustion in dust, dust fires	Inerting by $N_2$ , $CO_2$ , and rare gases	Partial inerting by inert gas
Other types of open flames (e.g., hot work)	Intrinsic inerting of dust cloud by combustion gases	Isolation (sectioning)
Hot surfaces (electrically, thermically or mechanically heated)	Inerting of dust cloud by adding inert dust	Venting
Electrical sparks and arcs. Electrostatic discharges	Dust concentration outside explosive range	Pressure-resistant construction
Heat from mechanical impact (metal sparks and hot-spots)		Automatic suppression
		Good housekeeping (dust removal, cleaning)

(Source: Eckhoff, R.K., *Dust Explosions in the Process Industries*, 3rd ed., Gulf Professional Publishing, 2003 [68].)**Figure 9-77** Increase in the reduced deflagration pressure for gases due to the effect of a vent duct. (By permission from AIChE, Meeting Mar 6, 1988 by I. Swift (deceased) [91], with data from [30].)**Figure 9-78** Increase in the reduced deflagration pressure for dusts due to the effect of a vent duct. (By permission from AIChE, Meeting Mar 6, 1988 by I. Swift (deceased) [91], with data from [30].)**TABLE 9-38 Common Methods Used in Controlling Dust Explosions**

Precautionary Measure	Comments	Parameter
Inerting (exclusion of oxygen using $N_2$ , $CO_2$ , or other suitable gas).	Reduces oxygen content below minimum necessary to support combustion (typically <5–15%). Requires monitoring of oxygen content. Usually requires closed system to conserve inert gas.	Limiting oxygen concentration.
Avoidance of ignition sources.	All practical measures must be taken to exclude ignition sources. Because sources are often unknown it is difficult to guarantee their exclusion, so other precautions are usually taken.	Layer ignition temperature. Cloud ignition temperature. Decomposition temperature. Spontaneous ignition temperature. Minimum ignition energy. Electrostatic behavior.
Detection of smoldering particles (wood flour industry, metal working).	Detection of smoldering particles in pneumatic transport or dust extraction lines by infrared sensors and quenching by a triggered water spray to reduce the probability of ignition.	
Concentration limitation (exclusion of dust cloud).	Material can be rendered less dusty and the handling system designed to minimize dust, or altered to a wet process.	Lower explosibility limit.
Replacement of combustible materials (diluent dust addition – to reduce explosibility of dust).	Non-combustible diluent, well mixed with dust, acts as heat sink thus reducing explosibility of dust. Limited application because of contamination.	Combustibility, heat of combustion, explosibility.



**Figure 9-79** Maximum allowable length of a vessel vented at one end, or maximum distance vents as a function of the vent diameter for the safe venting of deflagrations of dusts and gases. (By permission from AIChE, Meeting Mar 6, 1988 by I. Swift (deceased) [91], with data from [30].)

- Understand runaway reactions.
- Study the impact of two-phase flow on pressure-relieving device systems.

In the CPI, chemical manufacture especially in the fine, pharmaceutical and specialty chemical industries involves the processing of reactive chemicals, toxic or flammable liquids, vapors, gases, and powders. The safety records of these industries have improved in recent years; however, fires, explosions, and incidents involving hazardous chemical reactions do still occur. A basis for good engineering practice in assessing chemical reaction hazards is essential, with the aim to help designers, engineers, and scientists responsible for testing and operating chemical plants to meet the statutory duties of safety imposed by governmental organizations (e.g., U.S. Environmental Protection Agency (EPA), OSHA, USA, Health & Safety Executive, UK).

The control of chemical reactions (e.g., esterification, sulfonation, nitration, alkylation, polymerization, oxidation, reduction, and halogenation) and associated hazards are an essential aspect of chemical manufacture in the CPI. The industries manufacture nearly all their products such as inorganic, organic, agricultural, polymers, and pharmaceuticals through the control of reactive chemicals. The reactions that occur are generally without incident. Barton and Nolan [93] examined exothermic runaway incidents and found that the principal causes were as follows.

- inadequate temperature control
- inadequate agitation
- inadequate maintenance
- raw material quality
- little or no study of the reaction chemistry and thermochemistry
- human factors.

Other factors that are responsible for exothermic incidents are as follows.

- poor understanding of the reaction chemistry resulting in badly designed plant
- under-rated control and safety back-up systems
- inadequate procedures and training.

The research and technical evaluations have provided industry with extremely valuable information and design procedures, including, but not limited to, two-phase flow phenomena and runaway reactions during safety/overpressure relief.

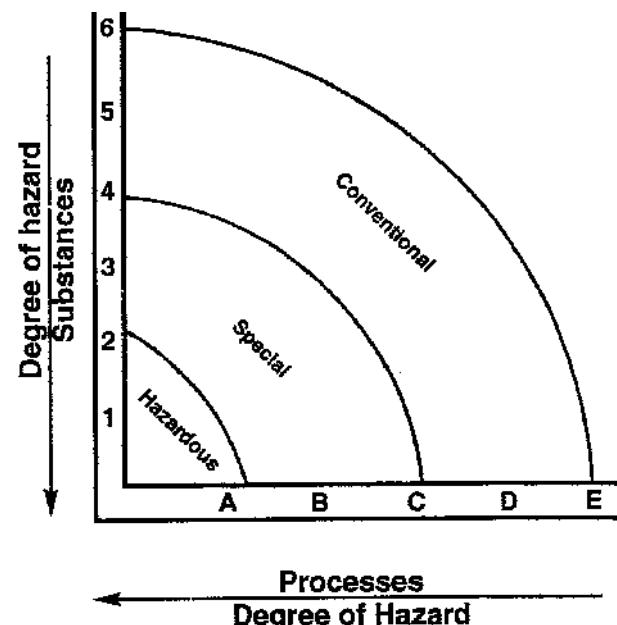
## 9.96 HAZARD EVALUATION IN THE CHEMICAL PROCESS INDUSTRIES

The safe design and operation of chemical processing equipment require detailed attention to the hazards inherent in certain chemicals and processes. Chemical plant hazards can occur from many sources; the principal ones arise from:

- fire and explosion hazards
- thermal instability of reactants, reactant mixtures, isolated intermediates, and products
- rapid gas evolution which can pressurize and possibly rupture the plant
- rapid exothermic reaction, which can raise the temperature or cause violent boiling of the reactants and also lead to pressurization.

Earlier reviews have been concerned with energy relationships for a particular chemical process, which are based upon two general classifications of chemical processes – conventional and hazardous [94]. The former is used to describe a non-explosive, non-flammable reaction and the latter to describe an explosive and flammable type reaction. The division of reactions does not account for the varying degree of safety or hazards of a particular reaction, which may lie between these extremes, and consequently becomes too limiting in the design. Safety is most likely to be neglected with conventional reactions while overdesign may be the expected practice with hazardous reactions. These limitations can be avoided by introducing a third class of reactions, "special" that covers the intermediate area between conventional and hazardous where reactions are relatively safe. Shabica [94] has proposed some guidelines for this third classification and Figure 9-80 shows the increasing degree of hazard of a particular process.

Chemical reaction hazards must be considered in assessing whether a process can be operated safely on the manufacturing scale. Furthermore, the effect of scale-up is particularly important. A reaction, which is innocuous on the laboratory or pilot plant scale, can be disastrous on a full-scale manufacturing plant. Therefore, the heat release from a highly exothermic process, for example the



**Figure 9-80** The increasing degree of hazard of matter plotted against the increasing hazard of a process. (Source: Shabica [66].)

reduction of an aromatic nitro compound, can be controlled easily in laboratory glassware. However, if the same reaction is carried out in a full-scale plant reactor with a smaller surface area to volume ratio, efficient cooling is essential, otherwise a thermal runaway and violent decomposition may occur. Similarly, a large quantity of gas produced by the sudden decomposition of a diazonium compound can be easily vented on the laboratory scale, but the same decomposition on the large scale could pressurize and rupture a full-scale plant.

In addition, consequences of possible process maloperations, such as incorrect charging sequence, contamination of reactants, agitation failure and poor temperature control, addition of reactants too quickly, omitting one of the reactants, and incorrect reactant concentration (recycling), must be considered. A number of parameters govern the reaction hazards associated with a process. These include the following.

- chemical components
- process temperature
- process pressure
- thermochemical characteristics of the reaction
- reaction rate
- reaction ratios
- solvent effects
- batch size
- operational procedure.

The assessment of the hazards of a particular process requires investigations of the effects of these parameters by experimental work, the interpretation of the results in relation to the plant situation, and the definition of a suitable basis for safe operation.

### 9.97 HAZARD ASSESSMENT PROCEDURES

Hazard assessments are essential, and should be performed on all chemical processes. The reactors such as batch, semi-batch and continuous can be employed for carrying out various operations. Many industrial reactions are exothermic (i.e., they are accompanied by the evolution of heat) and therefore overheating can occur. In batch operations, all the reactants are added to the reactor at the start of the reaction. In semi-batch operations, one or more of the reactants are charged to the reactor at the start, and others are then metered in, giving some control over the rate of reaction, and thus the rate of heat production. Overheating often results in thermal runaway, which is characterized by progressive increases in the rate of heat generation, and hence temperature and pressure.

Thermal runaway is a particular problem in unsteady state batch reactions, where the rate of reaction and therefore the rate of heat production vary with time. The consequences of thermal runaway can sometimes be very severe as in the incidents at Seveso [95]. In this case, a bursting disk ruptured on a reactor. The reactor was used to manufacture trichlorophenol at a temperature of 170 to 185°C and it was heated by steam at 190°C. The batch had been dehydrated and left at a temperature of 158°C over the weekend. Ethylene glycol and caustic soda give exothermic secondary reactions producing sodium polyglycolates, sodium oxalate, sodium carbonate, and hydrogen thus exhibiting an autocatalytic behavior. These reactions caused a temperature rise allowing the production of tetrachlorodioxin and the subsequent vessel pressurization. The Bhopal fatal incident [96] (is not a thermal runaway) involved a toxic release of an intermediate, methyl isocyanate (MIC), which resulted in the fatality of over 2000 people. A runaway polymerization of highly toxic MIC occurred in a storage tank. The runaway polymerization caused an uncontrolled pressure rise, which caused a relief valve to lift and discharge a plume of toxic vapor over the city.

The task of specifying the design, operation, and control of a reactor with stirrer, heating, or cooling coils, reflux facilities and emergency relief venting can pose a problem if all the time-dependent parameters are not considered. The use of batch processing techniques in the fine chemical industry is often characterized by

- multi-product plant (must have adaptable safety system)
- complex developing chemistry
- high frequency of change
- process control is simpler than continuous processes.

These factors are attributed to batch and semi-batch processes than continuous processes. However, the use of continuous processes on fine chemical manufacturing sites is limited. It is often preferable to use semi-batch mode as opposed to batch processes.

### 9.98 EXOTHERMS

Temperature-induced runaways have many causes including the following.

- loss of cooling
- loss of agitation
- excessive heating.

During a temperature-induced upset, the reactor temperature rises above the normal operating target. When any of the temperature elements senses a high reactor temperature, the programmable logic controller (PLC) software shutdown system automatically puts the reactor on idle (isolates). This action doubles the cooling water flow to the reactor by opening a bypass valve. If these reactions are unsuccessful in terminating the temperature rise, the shutdown system opens the quench valves, dumping the reaction mass into the water-filled quench tank. If the dump valves fail to operate, the reactor contents soon reach a temperature where a violent self-accelerating decomposition occurs. The uncontrolled exotherm generates a large volume of gas and ejects the process material out of the reactor into the containment pot.

### 9.99 ACCUMULATION

It is important to know how much heat of reaction can accumulate when assessing the hazards relating with an exothermic reaction. Accumulation in a batch or semi-batch process can be the result of:

- adding a reactant too quickly
- loss of agitation
- carrying out the reaction at too low a temperature
- inhibition of the reaction
- delayed initiation of the desired reaction.

Reactants can accumulate when the chosen reaction temperature is too low, and as such the reaction continues even after the end of the addition. In such a case, a hazardous situation could occur if cooling were lost as exemplified by [97].

Impurities or the delayed addition of a catalyst causes inhibition or delayed initiation resulting in accumulation in the reactors. The major hazard from accumulation of the reactants is due to a potentially rapid reaction and consequent high heat output that occurs when the reaction finally starts. If the heat output is greater than the cooling capacity of the plant, then it becomes a runaway reaction. The reaction might commence if an agitator is restarted after it has stopped, a catalyst is added suddenly, or because the desired reaction is slow to start.

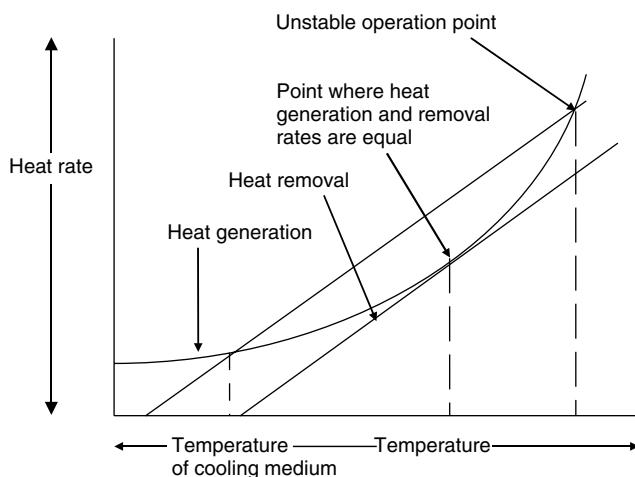


Figure 9-81 A typical curve of heat rate versus temperature.

### 9.100 THERMAL RUNAWAY CHEMICAL REACTION HAZARDS

Thermal runaway reactions are the results of chemical reactions in batch or semi-batch reactors. A thermal runaway commences when the heat generated by a chemical reaction exceeds the heat which can be removed to the surroundings as shown in Figure 9-81. The surplus heat increases the temperature of the reaction mass, which causes the reaction rate to increase, and subsequently accelerates the rate of heat production. Thermal runaway occurs as follows: as the temperature rises, the rate of heat loss to the surroundings increases approximately linearly with temperature. However, the rate of reaction, and thus the rate of heat generation, increases exponentially. If the energy release is large enough, high vapor pressures may result or gaseous products may be formed which

can finally lead to over pressurization and possible mechanical destruction of the reactor or vessel (thermal explosion).

The energy released might result from the wanted reaction or from the reaction mass if the materials involved are thermodynamically unstable. The accumulation of the starting materials or intermediate products is an initial stage of a runaway reaction. Figure 9-82 illustrates the common causes of reactant accumulation. The energy release with the reactant accumulation can cause the batch temperature to rise to a critical level thereby triggering the secondary (unwanted) reactions. Thermal runaway starts slowly and then accelerates until finally it may lead to an explosion.

#### 9.101 HEAT CONSUMED HEATING THE VESSEL: THE $\phi$ -FACTOR

The fraction of heat required to heat up the vessel, rather than its contents, depends on the heat capacity of the vessel (i.e., how much energy is required to raise the temperature of the vessel with respect to the reaction mass). The heat loss to the vessel is known as the  $\phi$ -factor [98]. The  $\phi$ -factor is the ratio of the total heat capacity of sample and vessel to that of the sample alone, and is defined by

$$\phi = \frac{M_s C_{ps}(\text{sample}) + M_v C_{pv}(\text{vessel})}{M_s C_{ps}(\text{sample})} \quad (9-89)$$

where

$M_s$  = mass of the sample (kg)

$C_{ps}$  = specific heat capacity of sample (J/kg K)

$M_v$  = mass of the vessel (kg)

$C_{pv}$  = specific heat capacity of vessel (J/kg K).

The  $\phi$ -factor does not account for the heat loss to the environment. It is used to adjust the self-heating rates as well as the observed adiabatic temperature rise.

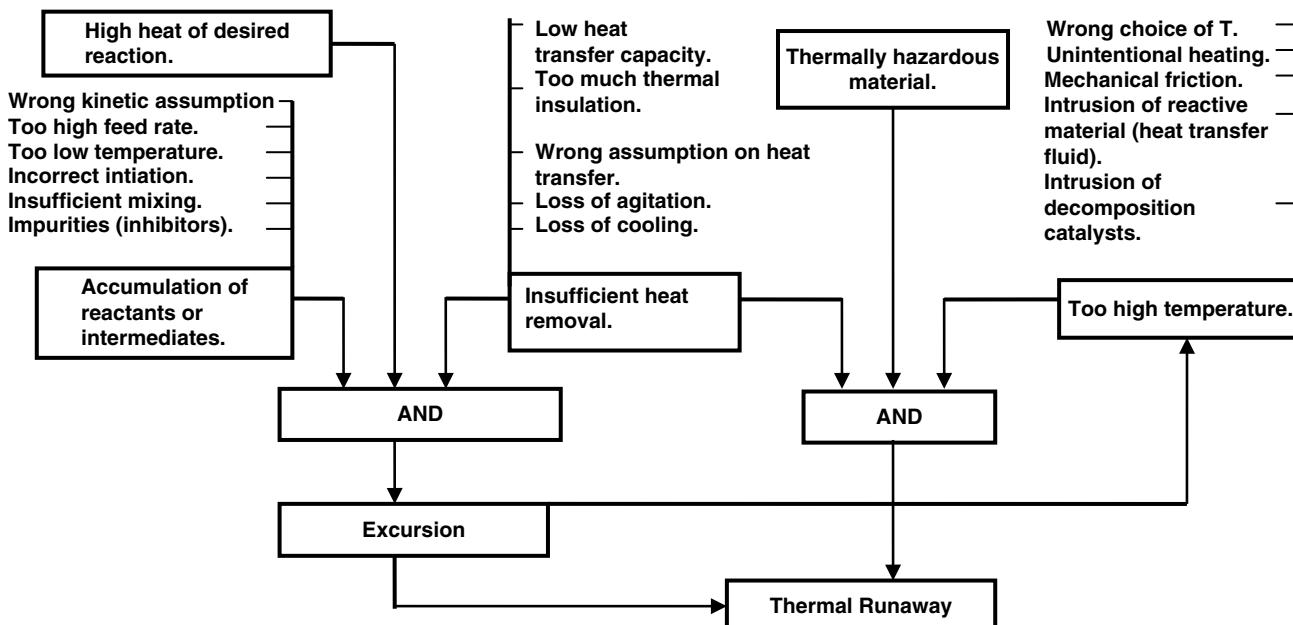


Figure 9-82 Causes of runaways in industrial reactors. (Source: W. Regenass, *Safe Operation of Exothermic Reactions*, Internal Ciba-Geigy Publication 1984.)

As the scale of operation increases, the effect of the heat consumption by the plant typically reduces, and thus the extent to which the kinetics of the runaway reaction is influenced by the plant is reduced. For plant scale vessels, the  $\phi$ -factor is usually low (i.e., 1.0–1.2) depending on the heat capacity of the sample and the vessel fill ratio. Laboratory testing for vent sizing must simulate these low  $\phi$ -factors. If the laboratory  $\phi$ -factor is high, several anomalies will occur.

- The rate of reaction will be reduced (i.e., giving incorrect reaction rate data for vent sizing).
- The magnitude of the exotherm will be smaller by a factor of  $\phi$ .
- Measured pressure effects will be smaller than those that would occur on the plant.
- The induction period of a thermal runaway reaction will be increased compared to the plant.

The consequences of these erroneous anomalies are as follows.

- inadequate safety design system
- undersized emergency relief system
- unknown decompositions may occur at elevated temperatures, which may not be realized in the laboratory.

## 9.102 ONSET TEMPERATURE

The reaction onset temperature is that temperature where it is assumed that significant fuel consumption begins. The onset temperature is expressed by [99]

$$T_{\text{onset}} = \frac{B}{\ln \left[ \frac{x_0 \Delta H_{\text{Rx}} A}{C \phi T_{\text{ex}}^*} \right]} \quad (9-90)$$

where

$$B = \frac{E_A}{R} \quad (9-91)$$

- A = Rate constant (pre-exponential factor from Arrhenius equation  $k = Ae(-E_A/RT)$ , s<sup>-1</sup> (i.e., for a first-order reaction))  
 B = Reduced activation energy, K  
 C = Liquid heat capacity of the product (J/kg K)  
 E<sub>A</sub> = Activation energy (J/mol)  
 R = Gas constant (8.314 J/mol K)  
 T<sub>onset</sub> = Onset temperature, K  
 x<sub>0</sub> = Initial mass fraction  
 ΔH<sub>Rx</sub> = Heat of decomposition, J/kg  
 φ = Dimensionless thermal inertial factor for the sample holder or product container (PHI-factor)  
 T<sub>ex</sub><sup>\*</sup> = Bulk heat-up rate driven by an external heat source (°C/s).

## 9.103 TIME-TO-MAXIMUM RATE

We may select an onset temperature based upon an arbitrary time-to-maximum rate from the relation

$$t_{\text{mr}} = \frac{C \phi T^2 e(B/T_{\text{onset}})}{x_0 \Delta H_{\text{Rx}} AB} \quad (9-92)$$

Re-arranging Eq. (9-92) yields

$$T_{\text{onset}} = \frac{B}{\ln \left( \frac{t_{\text{mr}} x_0 \Delta H_{\text{Rx}} AB}{C \phi T^2} \right)} \quad (9-93)$$

Equation 9-93 gives an onset temperature  $T_{\text{onset}}$  that corresponds to a time-to-maximum rate  $t_{\text{mr}}$  (min) using a successive substitution solution procedure. An initial guess of  $T = 350\text{ K}$  in the RHS of Eq. (9-93) will give a solution value of  $T_{\text{onset}}$  on the LHS of Eq. (9-93) within 1% or on an absolute basis ±3°C. Convergence is reached within several successive substitution iterations.

## 9.104 MAXIMUM REACTION TEMPERATURE

Once we have determined the onset temperature, we can then obtain the maximum reaction temperature by the adiabatic temperature rise and any contribution due to external heat input. The theoretical adiabatic temperature increase is

$$\Delta T_{\text{adia}} = \frac{x_0 \Delta H_{\text{Rx}}}{C \phi} \quad (9-94)$$

The contribution to the overall temperature increase from external heat input is defined by

$$\Delta T_{\text{ex}} = \frac{T_{\text{ex}}^* t_{\text{mr}}}{2} \quad (9-95)$$

where

- ΔT<sub>ex</sub> = The temperature increase attributed to external heating effects, °C  
 T<sub>ex</sub><sup>\*</sup> = Bulk heat-up rate due to external heating alone, °C/min  
 t<sub>mr</sub> = Time-to-maximum rate as determined by Eq. (9-93) with temperature  $T$  set equal to  $T_{\text{onset}}$ , min.

From these expressions, the maximum reaction temperature  $T_{\text{max}}$  is defined by

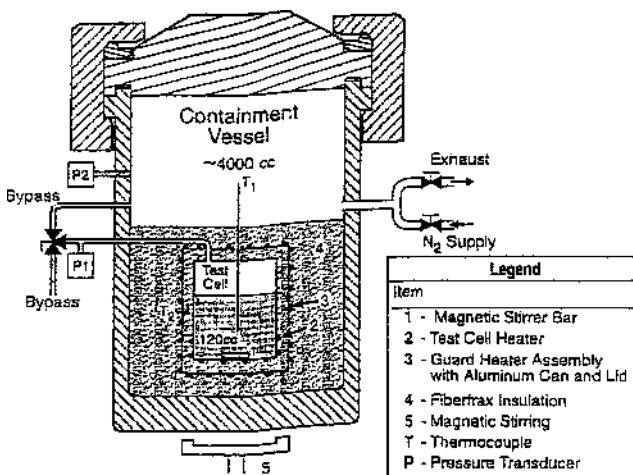
$$T_{\text{max}} = T_{\text{onset}} + \Delta T_{\text{adia}} + \Delta T_{\text{ex}} \quad (9-96)$$

To obtain updated listing of the published information on this research, contact the AIChE office in New York. The work is original and conducted by thoroughly qualified researchers/engineers. The work on runaway reactions is the first systematized examination of the subject and is really the only design approach available, which requires careful study.

Two-phase flow is an important aspect of venting relief as well as of runaway reactions, and is a complicated topic when related to liquid flashing in a vessel as it discharges on pressure relief. It cannot be adequately covered by conventional fluid two-phase flow. The following briefly reviews different calorimetric methods employed for screening and testing two-phase relief system. Other techniques are illustrated elsewhere [92].

## 9.105 VENT SIZING PACKAGE

The vent sizing package (VSP) was developed by Fauske & Associates Inc. The VSP and its latest version VSP2 employ the low thermal mass test cell stainless steel 304 and Hastelloy test cell with a volume of 120 ml contained in a 4L high pressure vessel as shown in Figure 9-83. The typical  $\phi$ -factor is 1.05–1.08 for a test cell wall thickness of 0.127–0.178 mm. Measurements consist of sample temperature  $T_1$  and pressure  $P_1$ , external guard temperature  $T_2$  and containment pressure  $P_2$ . During the runaway or the self-heating period, the guard heater assembly serves to provide an adiabatic environment for the test sample by regulating  $T_2$  close to  $T_1$ . For closed (non-vented) test cells, the containment vessel serves to prevent bursting of the test cell by regulating its own pressure  $P_2$  to follow the test cell sample pressure  $P_1$ . This pressure-tracking feature makes possible the utilization of the thin wall



**Figure 9-83** Vent sizing package (VSP) apparatus. (By permission from Fauske and Associates, Inc.)

(low  $\phi$ -factor) test cell design. Vented or open tests, where the vapor or gas generated is vented either into the containment vessel or to an external container, are unique capabilities of the VSP instruments. The typical onset sensitivity is  $0.1^{\circ}\text{C}/\text{min}$  for the VSP and  $0.05^{\circ}\text{C}/\text{min}$  for the VSP2.

The VSP experiments allow the comparison of various process versions, the direct determination of the wanted reaction adiabatic temperature rise, and the check of the possible initiations of secondary reactions. If no secondary reaction is initiated at the wanted reaction adiabatic final temperature, a further temperature scan allows the determination of the temperature difference between the wanted reaction adiabatic final temperature and the subsequent decomposition reaction onset temperature.

The VSP experiments are not suitable to measure the controlling reactant accumulation under normal process conditions. This is obtained using reaction calorimeters. The VSP experiments are suitable to assess the consequences of runaway decomposition reactions after their identification using other screening methods (Differential Thermal Analysis (DTA), Differential Scanning Calorimetry (DSC)

and Accelerating Rate Calorimeter (ARC)). The decomposition is then initiated by a temperature scan or an isothermal exposure, depending on the known kinetic behavior of the reaction. In some instances where a reliable baseline cannot be determined on DTA thermograms, a VSP experiment enables a better and safer determination of the decomposition exotherm.

The VSP experiments provide both thermal information on runaway reactions and information on pressure effects. The type of pressurization, following the DIERS methodology (i.e., vapor pressure, production of non-condensable gases, or both) can be determined from VSP experiments. The following experimental conditions are readily achievable using the VSP [100, 101].

- Temperature up to  $350\text{--}400^{\circ}\text{C}$  under temperature scan conditions.
- Pressure up to 200 bar.
- Closed or open test cells. Open test cells are connected to a second containment vessel. Test cells are of various materials including stainless steel, Hastelloy C276 and glass. Glass test cells of various sizes and shape are suitable for testing fine chemical and pharmaceutical products/reaction conditions, when the process is in glass vessels only or when small samples only are available, or if the samples are very expensive.
- Mechanical stirring is recommended for testing polymerization reactions or viscous reaction mixtures. This requires the use of taller containment vessels than the original one to install the electric motor for the agitator.

## 9.106 VENT SIZING PACKAGE 2™ (VSP2™)

Figure 9-84 shows the VSP2 system. It uses a patented low thermal mass temperature and pressure equalized 120-ml test cell configuration. The equipment is versatile and enables users to obtain accurate adiabatic temperature and pressure rate data for the fastest runaway reactions. The test data can directly be applied to process scale without performing tedious computations. The VSP2 tests are employed to model upset conditions as:

- loss of cooling
- loss of stirring
- mischarge of reagents



**Figure 9-84** The Vent Sizing package 2™ (VSP 2™) apparatus. (By permission from Fauske and Associates, Inc.)

- batch contamination
- fire exposure heating.

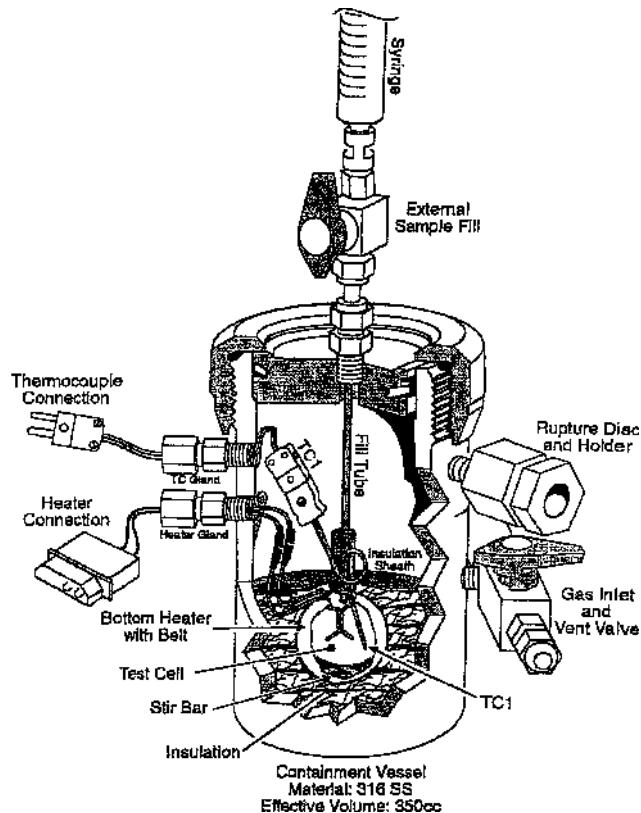
The VSP2 provides vent sizing and thermal data under adiabatic runaway reaction conditions, which can be directly applied to process scale.

### 9.107 ADVANCED REACTIVE SYSTEM SCREENING TOOL

Fauske & Associates Inc. developed the ARSST, as an easy and inexpensive screening tool for characterizing chemical systems and acquiring relief system design data. It can safely identify potential chemical hazards in the process industry. The ARSST (Figures 9-85a, 9-85b and c) consists of a spherical glass test cell, and immersion heater. It has surrounding jacket heater and insulation, thermocouples and a pressure transducer, a stainless steel containment vessel that serves as both a pressure simulator and safety vessel. The ARSST uses a small magnetic stir bar which is placed in the test cell and driven by an external magnetic stirrer.

The sample cell volume is 10 ml and the containment volume is 350 ml. The apparatus has a low effective heat capacity relative to that of the sample whose value, expressed as the capacity ratio, is approximately 1.04 (i.e., quite adiabatic). This key feature allows the measured data to be directly applied to process scale.

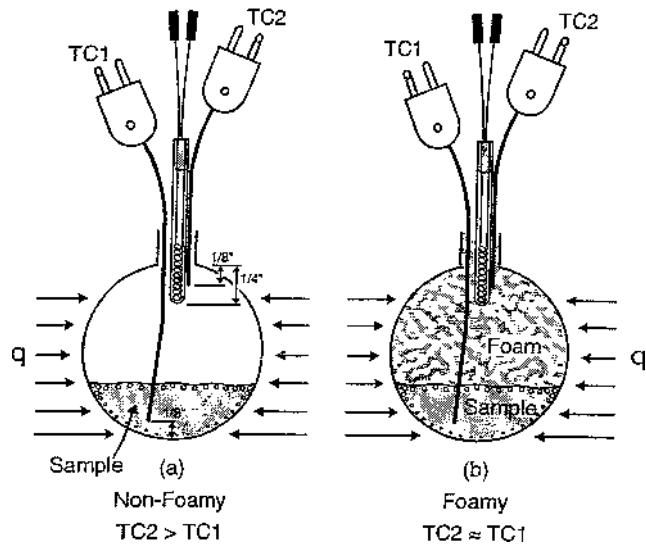
The ARSST features a heat-wait-search (HWS) mode of operation that provides onset detection sensitivity as low as  $0.1^\circ\text{C}/\text{min}$ , and isothermal operation at elevated temperature. It can readily cope with endothermic behavior (phase change) and an optical flow regime detector enables the ARSST operator to distinguish between "foamy" and "non-foamy" runaway reactions (Figure 9-85c). The ARSST is computer controlled, which records time, temperatures and pressure, and heater power during a test. Figures 9-86–9-90 show typical plots generated by the equipment [102].



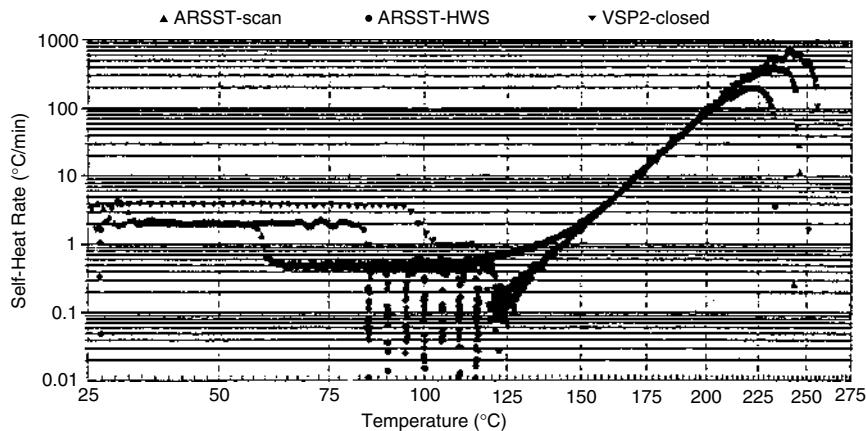
**Figure 9-85b** Schematic of the Advanced Reactive System Screening Tool™ (ARSST™) containment vessel and internals. (By permission from Fauske and Associates, Inc.)



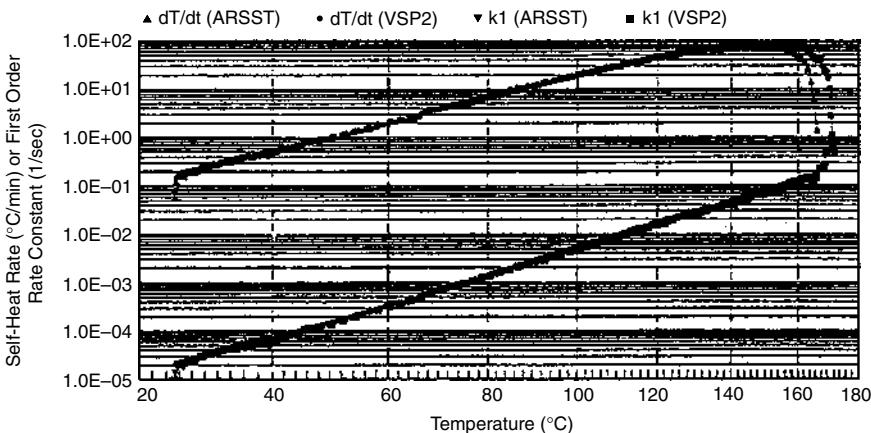
**Figure 9-85a** The Advanced Reactive System Screening Tool™ (ARSST™). (By permission from Fauske and Associates, Inc.)



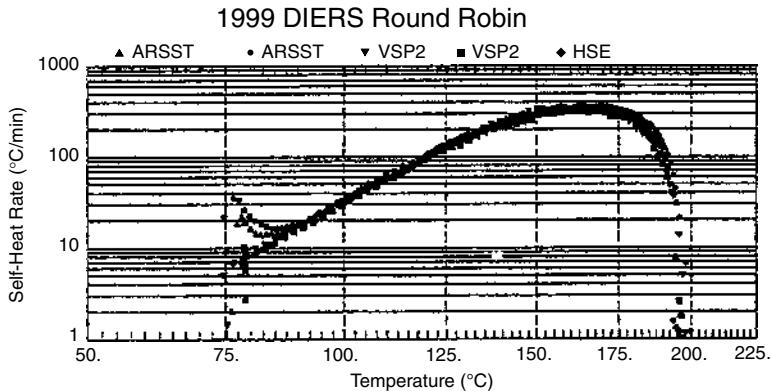
**Figure 9-85c** The advanced Reactive System Screening Tool™ (ARSST™) flow regime detector. (By permission from Fauske and Associates, Inc.)



**Figure 9-86** Self-heat rate data for 25% DTBP in Toluene. (By permission from Fauske and Associates, Inc.)



**Figure 9-87** Self-heat rate data and first-order rate constant for Methanol/Acetic Anhydride. (By permission from Fauske and Associates, Inc.)



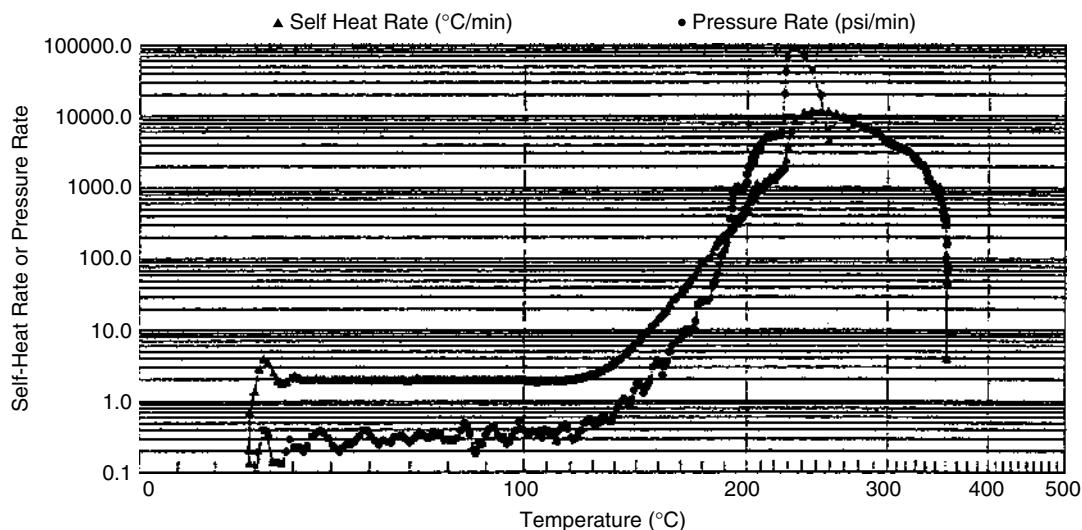
**Figure 9-88** Composite self-heat plot of selected Round Robin data. (By permission from Fauske and Associates, Inc.)

### 9.108 TWO-PHASE FLOW RELIEF SIZING FOR RUNAWAY REACTION

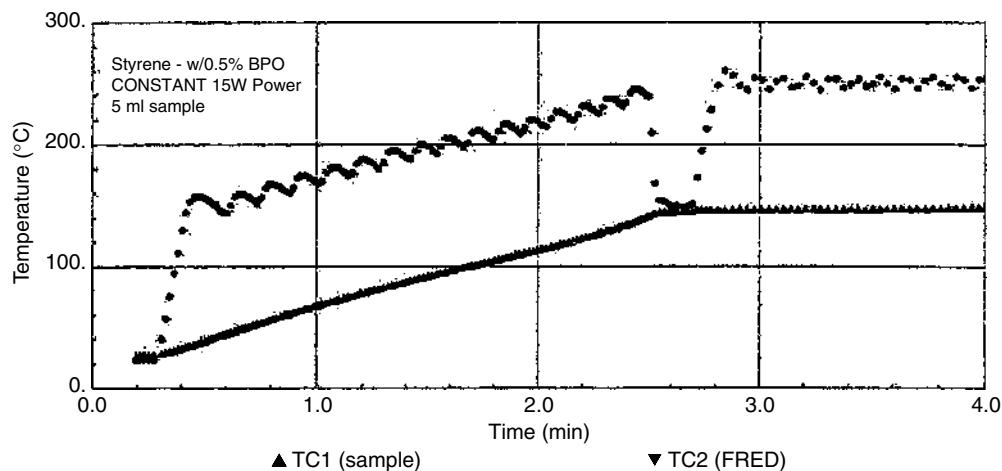
Many methods have been used to size relief systems: area/volume scaling, mathematical modeling using reaction parameters and flow theory, and empirical methods by the Factory Insurance Association (FIA). The DIERS of the AIChE has carried out studies of sizing reactors undergoing runaway reactions. Intricate laboratory instruments as described earlier have resulted in better vent sizes.

A selection of relief venting as the basis of safe operation is based upon the following considerations [103].

- compatibility of relief venting with the design and operation of the plant/process
- identifying the worst scenarios
- type of reaction
- means of measuring the reaction parameters during the runaway reaction



**Figure 9-89** Self-heat rate and pressure rate data for neat DTBP at 300 psig in the ARSST. (By permission from Fauske and Associates, Inc.)



**Figure 9-90** Example of flow regime detection with the ARSST. (By permission from Fauske and Associates, Inc.)

- relief sizing procedure
- design of the relief system including discharge ducting and safe discharge area.

### 9.109 RUNAWAY REACTIONS

A runaway reaction occurs when an exothermic system becomes uncontrollable. The reaction leads to a rapid increase in the temperature and pressure which if not relieved can rupture the containing vessel. A runaway reaction happens because the rate of reaction, and therefore the rate of heat generation, increases exponentially with temperature. In contrast, the rate of cooling increases only linearly with temperature. Once the rate of heat generation exceeds available cooling, the rate of increase in temperature becomes progressively faster. Runaway reactions nearly always result in two-phase flow reliefs. In reactor venting, reactions essentially fall into three classifications.

1. Vapor-pressure systems
2. Hybrid (gas + vapor) systems
3. Gassy systems.

The significance of these categories in terms of relief is that, once the vent has opened, both vapor pressure and hybrid reactions temper by losing enough heat through vaporization to maintain temperature and pressure at an acceptable level. In a gas generating system, there is negligible or sometimes no control of temperature during venting, such that relief sizing is based upon the peak gas generation rate. Experimental studies conducted by the testing methods must not only be able to differentiate between the reaction types, but must also simulate large-scale process conditions. The results of the experimental studies should greatly enhance the design of the relief system. We shall review the reactor venting categories in the following sections.

### 9.110 VAPOR-PRESSURE SYSTEMS

In this type of reaction, no permanent gas is generated. The pressure generated by the reaction is due to the increasing vapor pressure of the reactants, products, and/or inert solvent as the temperature rises.

It is the rate of temperature increase (i.e., power output) between the set pressure and the maximum allowable pressure,

which determines the vent size and not the peak rate. Boiling is attained before potential gaseous decomposition (i.e., the heat of reaction is removed by the latent heat of vaporization). The reaction is tempered, and the total pressure in the reactor is equal to the vapor pressure. The principal parameter determining the vent size is the rate of the temperature rise at the relief set pressure.

Systems that behave in this manner obey the Antoine relationship between pressure  $P$  and temperature  $T$  as represented by

$$\ln P = A + B/T \quad (9-97)$$

where A and B are constants and T is the absolute temperature. An example is the methanol and acetic anhydride reaction.

### 9.111 GASSY SYSTEMS

Here, the system pressure is due entirely to the pressure of non-condensable gas rather than the vapor pressure of the liquid. The gas is the result of decomposition. The exothermic heat release is largely retained in the reaction mass since the cooling potential of volatile materials is not available. As such, both the maximum temperature and the maximum gas generation rate can be attained during venting. Gaseous decomposition reactions occur without tempering. The total pressure in the reactor is equal to the gas pressure. The principal parameter determining the vent size is the maximum rate of pressure rise. Unlike the vapor-pressure systems, the pressure is controlled (and reduced) without cooling the reaction.

A survey within the Fine Chemical Manufacturing Organization of Imperial Chemical Industries (ICI) has shown that gassy reaction systems predominate due to established processes such as nitrations, diazotizations, sulphonations, and many other types of reactions [104]. Very few vapor-pressure systems have been identified which also generated permanent gas, that is hybrid type.

### 9.112 HYBRID SYSTEMS

These are systems that have a significant vapor pressure and at the same time produce non-condensable gases. Gaseous decomposition reaction occurs before boiling: the reaction is still tempered by vapor stripping. The total pressure in the reactor is the summation of the gas partial pressure and the vapor pressure. The principal parameters determining the vent size are the rates of temperature and pressure rise corresponding to the tempering condition. A tempered reactor contains a volatile fluid that vaporizes or flashes during the relieving process. This vaporization removes energy via the heat of vaporization and tempers the rate of temperature rise due to the exothermic reaction.

In some hybrid systems, the vapor generation in a vented reaction is high enough to remove sufficient latent heat to moderate or “temper” the runaway, that is to maintain constant temperature. This subsequently gives a smaller vent size.

Richter and Turner [105] have provided systematic schemes for sizing batch reactor relief systems. They employed logic diagrams that outlined the various decisions to produce a model of the system. Figure 9-91 reviews the reaction kinetics and thermodynamics required to create a reactor model. Figure 9-92 shows a sequence of steps used to model flow from the reactor, assuming relief is occurring as a homogeneous vapor–liquid mixture. The DIERS program has supported the use of a homogeneous vapor–liquid mixture (froth) model, which relies on the assumption that the vapor phase is in equilibrium with the batch liquid phase [106]. Figure 9-93 shows the steps used to model compressible vapor venting from the reactor. Table 9-39 lists formulae for computing the area of the three systems.

### 9.113 SIMPLIFIED NOMOGRAPH METHOD

Boyle [107] and Huff [108] first accounted for two-phase flow with relief system design for runaway chemical reactions. Computer simulation approaches to vent sizing involve extensive thermokinetic and thermophysical characterization of the reaction system. Fisher [109] has provided an excellent review of emergency relief system design involving runaway reactions in reactors and vessels. The mass flux through the relief device representing choked two-phase flow through a hole is expressed as

$$G = \frac{Q_m}{A} = \frac{\Delta H_V}{v_{fg}} \left( \frac{g_c}{C_p T_s} \right)^{0.5} \quad (9-98)$$

For two-phase flow through pipes, we apply an overall dimensionless discharge coefficient  $\psi$ . Equation (9-98) is referred to as the equilibrium rate model (ERM) for low quality choked flow. Leung [110] indicated that Eq. (9-98) be multiplied by a factor of 0.9 to bring the value in line with the classic homogeneous equilibrium model (HEM). Equation (9-98) becomes

$$G = \frac{Q_m}{A} = 0.9\psi \frac{\Delta H_V}{v_{fg}} \left( \frac{g_c}{C_p T_s} \right)^{0.5} \quad (9-99)$$

where

$A$  = area of the hole,  $m^2$

$g_c$  = correction factor  $1.0 \frac{kg\ m}{s^2}/N$

$Q_m$  = mass flow through the relief,  $kg/s$

$\Delta H_V$  = the heat of vaporization of the fluid,  $J/kg$

$v_{fg}$  = the change of specific volume of the flashing liquid,  $m^3/kg$

$C_p$  = the heat capacity of the fluid,  $J/kg\ K$

$T_s$  = the absolute saturation temperature of the fluid at the set pressure,  $K$ .

The result is applicable for homogeneous venting of a reactor (low quality, not restricted just to liquid inlet condition). Figure 9-94 gives the value of  $\psi$  for L/D ratio. For a pipe length zero,  $\psi = 1$  as the pipe length increases, the value of  $\psi$  decreases. Equation (9-99) can be further re-arranged in terms of a more convenient expression as follows:

$$\frac{\Delta H_V}{v_{fg}} = T_s \frac{dP}{dT} \quad (9-100)$$

Substituting Eq. (9-100) into Eq. (9-99) gives

$$G = 0.9\psi \frac{dP}{dT} \left( \frac{g_c T_s}{C_p} \right)^{0.5} \quad (9-101)$$

The exact derivative is approximated by a finite difference derivative to yield

$$G \cong 0.9\psi \frac{\Delta P}{\Delta T} \left( \frac{g_c T_s}{C_p} \right)^{0.5} \quad (9-102)$$

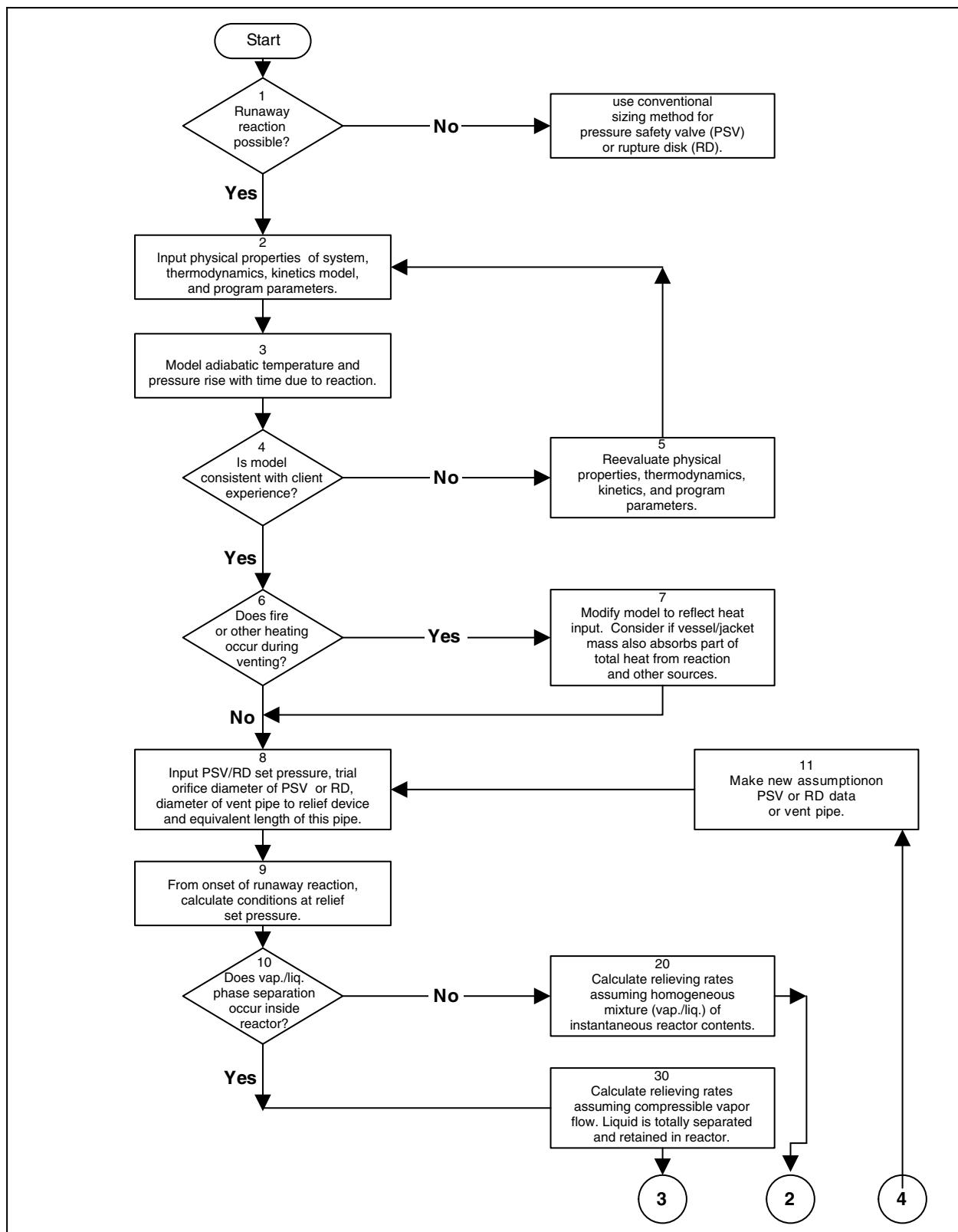
where

$\Delta P$  = the overpressure

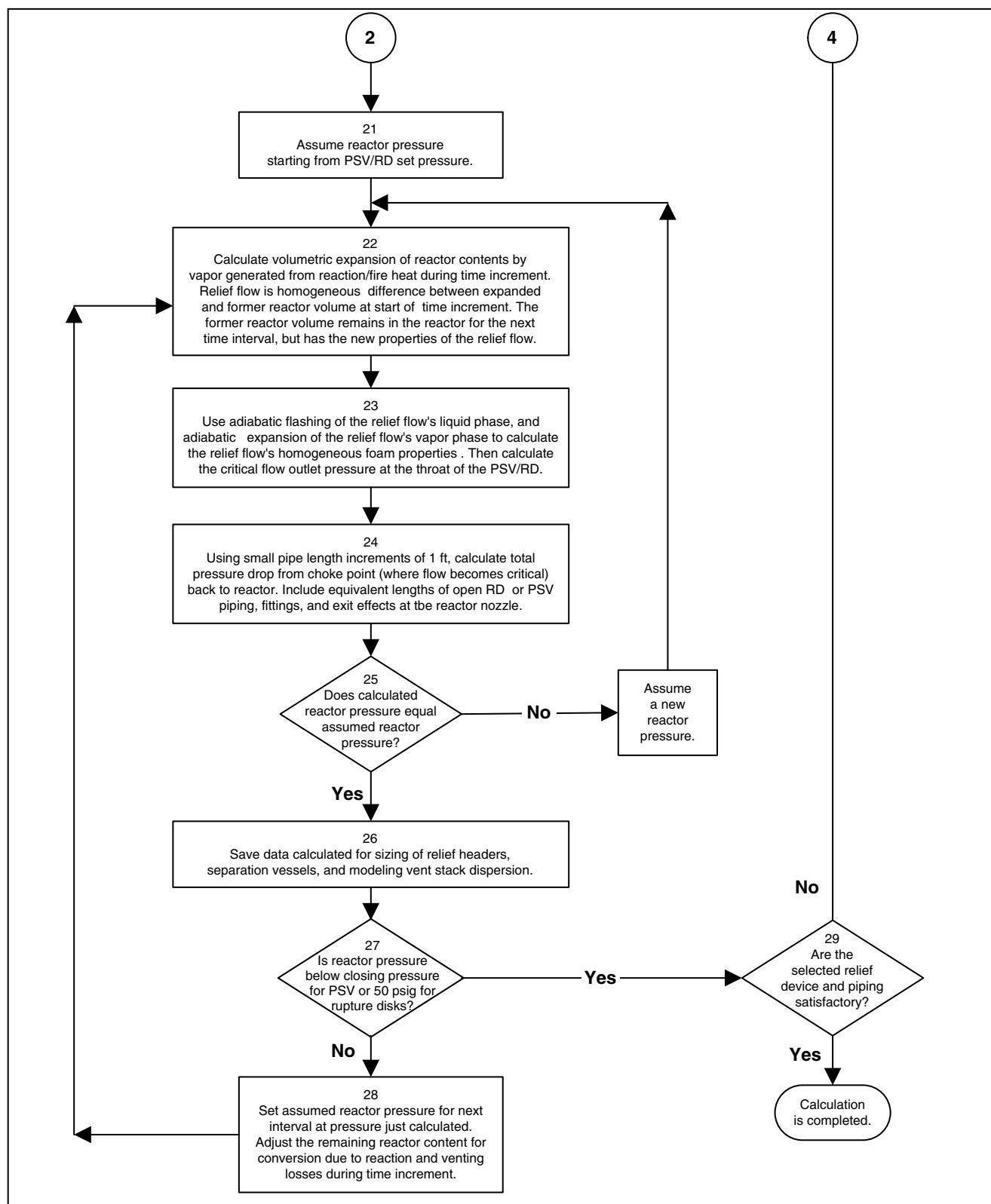
$\Delta T$  = the temperature rise corresponding to the overpressure.

Fauske [111] has developed a simplified chart for the two-phase calculation. He expressed the relief area as:

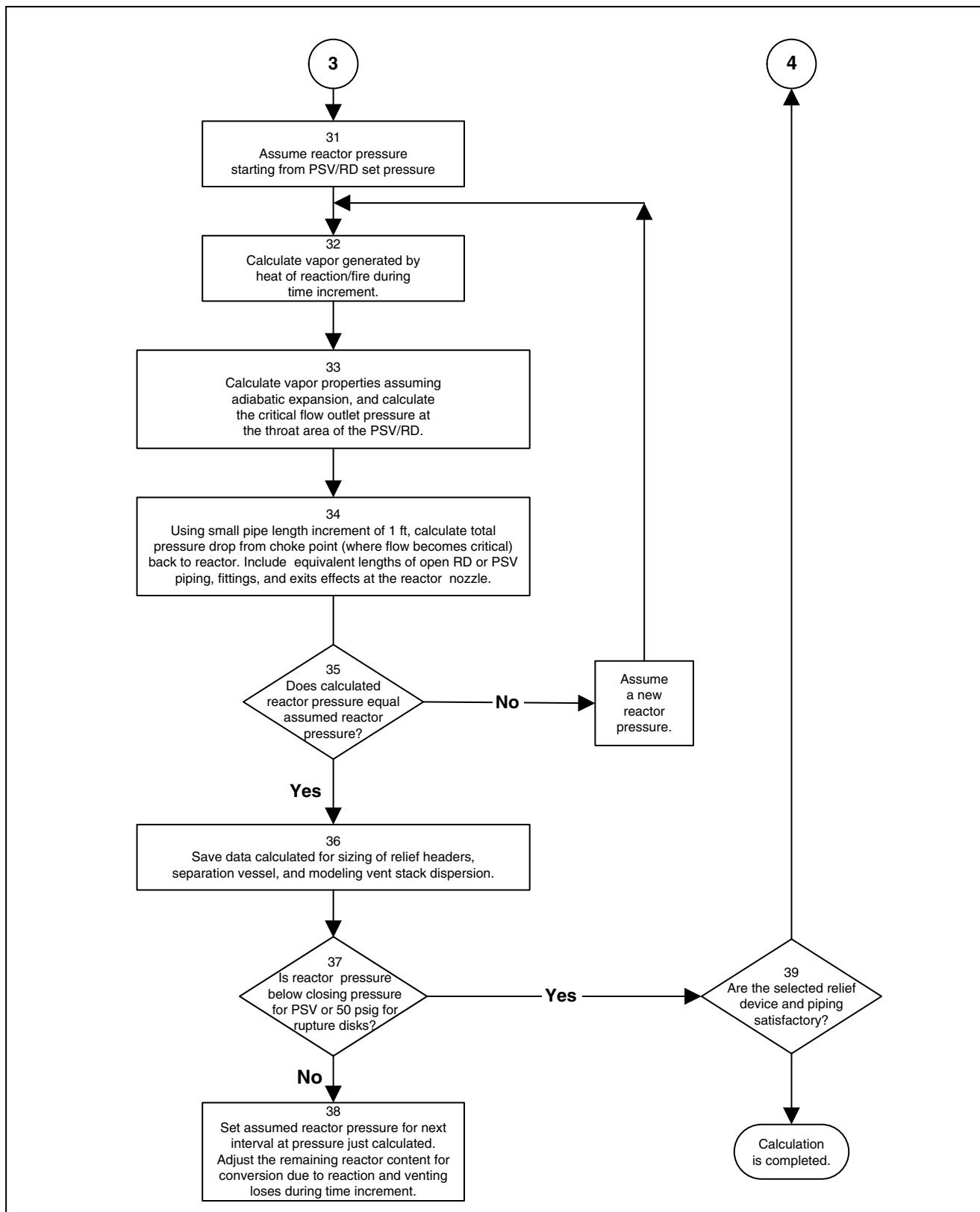
$$A = \frac{V\rho}{G\Delta t_v} \quad (9-103)$$



**Figure 9-91** Block flow diagram showing the reaction kinetics and thermodynamics needed to create a reactor model. (Reproduced with the permission from the AIChE copyright © 1996 AIChE, All rights reserved.)



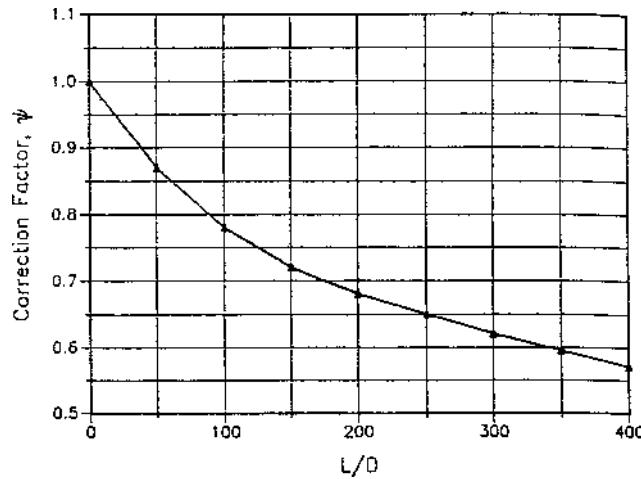
**Figure 9-92** Steps used to model flow from the reactor, assuming relief is occurring as a homogeneous vapor-Liquid mixture. (Reproduced with permission from the AIChE. Copyright © 1996 AIChE. All rights reserved.)



**Figure 9-93** Steps used to model compressible vapor venting from the reactor. (Reproduced with permission from the AIChE. Copyright © 1996 AIChE. All rights reserved.)

**TABLE 9-39 Vent Areas and Diameters of the Three Systems**

	<b>Vapor System</b>	<b>Gassy System</b>	<b>Hybrid System</b>
Area	$A = 1.5 \times 10^{-5} \left( \frac{m_o \frac{dT}{dt}}{F P_s} \right)$	$A = 3 \times 10^{-6} \left( \frac{1}{F} \right) \left( \frac{m_o}{m_t} \right) \left( \frac{\frac{dP}{dt}}{P_{MAP}^{1.5}} \right)$	$A = 5.6 \times 10^{-6} \left( \frac{1}{F} \right) \left( \frac{m_o}{m_t} \right) \left( \frac{\frac{dP}{dt}}{P_s^{1.5}} \right)$
Diameter	$d = \left( \frac{4A}{\pi} \right)^{0.5}$	$d = \left( \frac{4A}{\pi} \right)^{0.5}$	$d = \left( \frac{4A}{\pi} \right)^{0.5}$
L/D = 0, F = 1.00		L/D = 0, F = 1.00	L/D = 0, F = 1.00
L/D = 50, F = 0.85		L/D = 50, F = 0.70	L/D = 50, F = 0.70
L/D = 100, F = 0.75		L/D = 100, F = 0.60	L/D = 100, F = 0.60
L/D = 200, F = 0.65		L/D = 200, F = 0.45	L/D = 200, F = 0.45
L/D = 400, F = 0.50		L/D = 400, F = 0.33	L/D = 400, F = 0.33



**Figure 9-94** Correction factor versus L/D for two-phase flashing flow through pipes. (Source: J.C. Leung and M.A. Grolmes, "The Discharge of Two-Phase Flashing Flow in a Horizontal Duct", *AIChE J.*, Vol. 33, No. 3, p. 524, 1987.)

where

$A$  = the relief vent area,  $\text{m}^2$

$V$  = reactor volume,  $\text{m}^3$

$\rho$  = density of the reactants,  $\text{kg}/\text{m}^3$

$G$  = mass flux through the relief,  $\text{kg}/\text{s m}^2$

$\Delta t_V$  = venting time, s.

Boyle [107] developed Eq. (9-103) by defining the required area as that size which would empty the reactor before the pressure could rise above some allowable pressure for a given vessel. The mass flux  $G$  is given by Eq. (9-99) or Eq. (9-102), and the venting time is given by

$$\Delta t_V = \frac{(\Delta T)(C_p)}{q_s} \quad (9-104)$$

where

$\Delta T$  = the temperature rise corresponding to the overpressure  $\Delta P$

$T$  = the temperature

$C_p$  = the heat capacity

$q_s$  = the energy release rate per unit mass at the set pressure of the relief system.

Combining Eqs (9-103), (9-104) and (9-98) yields

$$A = V\rho (g_c C_p T_s)^{-0.5} \frac{q_s}{\Delta P} \quad (9-105)$$

Equation (9-105) gives a conservative estimate of the vent area and the simple design method represents overpressure ( $\Delta P$ ) between 10 and 30%. For a 20% absolute overpressure, a liquid heat capacity of  $2510 \text{ J/kg K}$  for most organics, and considering that a saturated water relationship exists, the vent size area per 1000 kg of reactants is given by

$$A = \left( \frac{m^2}{1000 \text{ kg}} \right) = \frac{0.00208 \left( \frac{dT}{dt} \right) \text{ °C/min.}}{P_s \text{ bar}} \quad (9-106)$$

Figure 9-95 shows a nomograph for determining the vent size. The vent area is calculated from the heating rate, the set pressure, and the mass of reactants. The nomograph is used for obtaining quick vent sizes and checking the results of the more rigorous computation. Crowl and Louvar [41] have expressed that the nomograph data of Figure 9-95 applies for a discharge coefficient of  $\psi = 0.5$ , representing a discharge  $L/D$  of 400.0. However, use of the nomograph at other discharge pipe lengths and different  $\psi$  requires a suitable conversion.

### 9.114 VENT SIZING METHODS

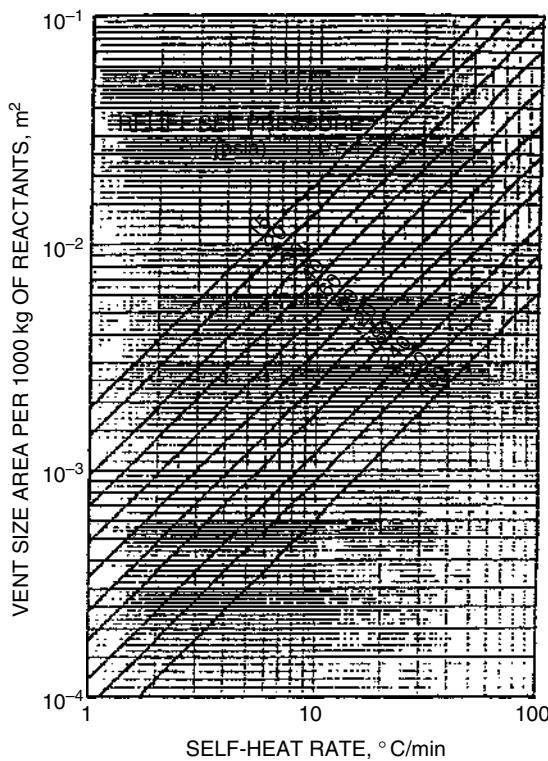
Vents are usually sized on the assumption that the vent flow is

- all vapor or gas
- all liquid, and
- a two-phase mixture of liquid and vapor or gas.

The first two cases represent the smallest and largest vent sizes required for a given rate at increased pressure. Between these cases, there is a two-phase mixture of vapor and liquid. It is assumed that the mixture is homogeneous, that is, that no slip occurs between the vapor and the liquid. Furthermore, the ratio of vapor to liquid determines whether the venting is closer to all vapor or all liquid case. As most relief situations involve a liquid fraction of over 80%, the idea of homogeneous venting is closer to all liquid than all vapor. Table 9-40 shows vent area for different flow regimes.

### 9.115 VAPOR-PRESSURE SYSTEMS

These systems are called "tempering" (i.e., to prevent temperature rise after venting) systems as there is sufficient latent heat available to remove the heat of reaction, and to temper the reaction at the



**Figure 9-95** A vent sizing nomograph for tempered (high vapor-pressure) runaway chemical reactions. (Source: H.K. Fauske, "Generalized Vent Sizing Monogram for Runaway Chemical Reactions", *Plant/Operations Prog.*, Vol. 3, No. 4, 1984. Reproduced with permission from the AIChE, Copyright ©1984, All rights reserved.)

**TABLE 9-40 Vent Areas for Different Flow Regimes**

Type of Flow	Required Vent Area as a Multiple of all Vapor Vent Area
All vapor	1
Two-phase: Churn turbulent	2–5
Bubbly	7
Homogeneous	8
All liquid	10

set pressure. The vent requirements for such systems are estimated from the Leung's Method [112, 113].

$$A = \frac{M_O q}{G \left[ \left( \frac{V}{M_O} T_S \frac{dP}{dT} \right)^{0.5} + (C_V \Delta T)^{0.5} \right]^2} \quad (9-107)$$

Alternatively, the vent area can be expressed as:

$$A = \frac{M_O q}{G \left[ \left( \frac{V}{M_O} \frac{\Delta H_V}{v_{fg}} \right)^{0.5} + (C_V \Delta T)^{0.5} \right]^2} \quad (9-108)$$

where

$M_O$  = the total mass contained within the reactor vessel prior to relief, kg

$q$  = the exothermic heat release rate per unit mass,  $\frac{J}{kg \cdot s}$ ,  $\frac{W}{kg}$

$V$  = the volume of the vessel,  $m^3$

$C_V$  = the liquid heat capacity at constant volume,  $J/kg \cdot K$

$\Delta H_V$  = heat of vaporization of the fluid,  $J/kg$

$v_{fg}$  = change of specific volume of the flashing liquid,  $(v_g - v_f)$ ,  $m^3/kg$ .

The heating rate  $q$  is defined by

$$q = \frac{1}{2} C_V \left[ \left( \frac{dT}{dt} \right)_s + \left( \frac{dT}{dt} \right)_m \right] \quad (9-109)$$

The first derivative Eq. (9-109), denoted by the subscript "s", corresponds to the heating rate at the set pressure and the second derivative, denoted by subscript "m", corresponds to the temperature rise at the maximum turnaround pressure.

The above equations assume the following.

- Uniform froth or homogeneous vessel venting occurs.
- The mass flux,  $G$ , varies little during the relief.
- The reaction energy per unit mass,  $q$ , is treated as constant.
- Constant physical properties  $C_V$ ,  $\Delta H_V$ , and  $v_{fg}$ .
- The system is a tempered reactor system. This applies to the majority of reaction systems.

We should take care of using consistent units in applying the above two-phase equations. The best procedure is to convert all energy units to their mechanical equivalents before solving for the relief area, especially when Imperial (English) units are used. To be consistent, use the SI unit.

The vapor-pressure systems obey the Antoine relationships given by Eq. (9-97),

$$\ln P = A + \frac{B}{T}$$

Differentiating Eq. (9-97), yields

$$\frac{1}{P} \frac{dP}{dT} = -\frac{B}{T^2} \quad (9-110)$$

$$\frac{dP}{dT} = -\frac{B}{T^2} P \quad (9-111)$$

An equation representing the relief behavior for a vent length  $L/D < 400$  is given by [113]

$$M_O = \frac{(D_p)^2 (\Delta P_s)}{2.769 \left( \frac{dT}{dt} \right)_s} \cdot \left( \frac{T_s}{C_p} \right)^{0.5} \quad (9-112)$$

where

$M_O$  = allowable mass of the reactor mixture charge (kg) to limit the venting overpressure to  $P_p$  (psig)

$D_p$  = rupture disk diameter, in.

$\Delta P_s$  = the allowable venting overpressure (psi), that is the maximum venting pressure minus the relief device set pressure

- $P_P$  = maximum venting pressure (psig)  
 $P_S$  = the relief device set pressure (psig) (Note that the relief device set pressure can range from the vessel's MAWP to significantly below the MAWP.)  
 $T_s$  = the equilibrium temperature corresponding to the vapor pressure where the vapor pressure is the relief device set pressure ( $K$ )  
 $(\frac{dT}{dt})_s$  = the reactor mixture self-heat rate ( $^{\circ}\text{C}/\text{min}$ ) at temperature  $T_s(K)$  as determined by a DIERS or equivalent test  
 $C_p$  = specific heat of the reactor mixture (cal/g K or Btu/lb° F).

Equation (9-112) is a dimensional equation; therefore the dimensions given in the parameters must be used.

### 9.116 FAUSKE'S METHOD

Fauske [113] represented a nomograph for tempered reactions as shown in Figure 9-95. This accounts for turbulent flashing flow and requires information about the rate of temperature rise at the relief set pressure. This approach also accounts for vapor disengagement and frictional effects including laminar and turbulent flow conditions. For turbulent flow, the vent area is

$$A = \frac{1}{2} \frac{M_O (\frac{dT}{dt})_s (\alpha_D - \alpha_O)}{F \left( \frac{T_s}{C_p} \right)^{0.5} \Delta P (1 - \alpha_O)} \quad \text{for } 0.1 P_S \leq \Delta P \leq 0.3 P_S \quad (9-113)$$

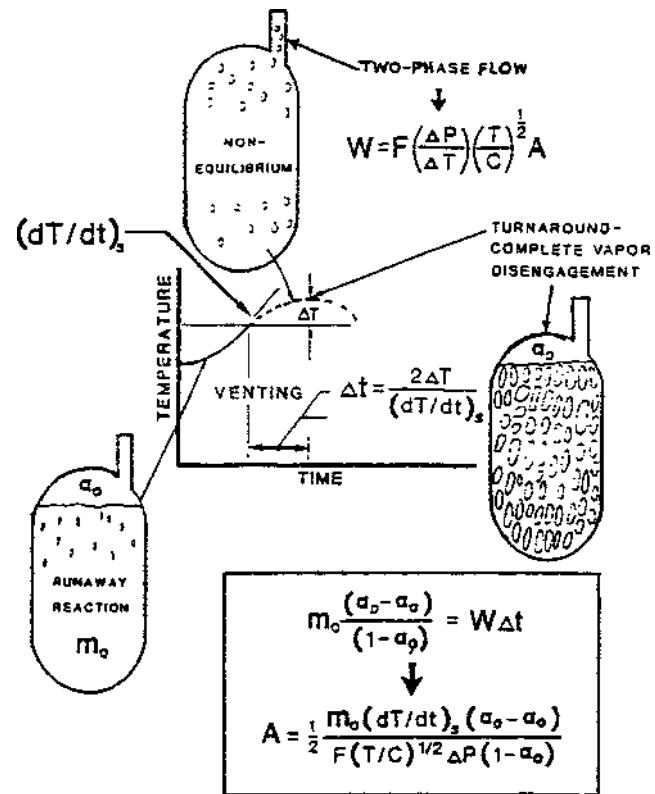
where

- $M_O$  = initial mass of reactants, kg  
 $(\frac{dT}{dt})_s$  = self-heat rate corresponding to the relief set pressure, (K/s)  
 $(\frac{dT}{dt})_m$  = self-heat rate at turnaround temperature, (K/s)  
 $P_S$  = relief set pressure (Pa)  
 $\alpha_D$  = vessel void fraction corresponding to complete vapor disengagement  
 $\alpha_O$  = initial void fraction in vessel  
 $T_s$  = temperature corresponding to relief actuation, K  
 $C_p$  = liquid specific heat capacity, J/kg K  
 $\Delta P$  = equilibrium overpressure corresponding to the actual temperature rise, Pa.  
 $\Delta T$  = temperature rise following relief actuation, K  
 $F$  = flow reduction correction factor for turbulent flow ( $L/D = 0, F \approx 1.0; L/D = 50, F \approx 0.85; L/D = 100, F \approx 0.75; L/D = 200, F \approx 0.65; L/D = 400, F \approx 0.55$ ). where  $L/D$  is the length-to-diameter ratio of the vent line.

Figure 9-96 shows a sketch of temperature profile for high vapor-pressure systems.

The VSP bench scale apparatus can be employed to determine the information about the self-heat rate and vapor disengagement when this is not readily available. In addition, the VSP equipment can be used for flashing flow characteristics using a special bottom-vented test cell. Here, the flow rate,  $G_O(\text{kg/sm}^2)$ , is measured in a simulated vent line (same  $L/D$  ratio) of diameter  $D_O$  using the VSP apparatus. The following is the recommended scale-up approach in calculating the vent size.

$$\text{If } G_O \left( \frac{D_T}{D_O} \right) \geq G_T \cong F \left( \frac{\Delta P}{\Delta T} \right) \left( \frac{T_s}{C_{p,S}} \right)^{0.5}$$



**Figure 9-96** Vent sizing model for high vapor-pressure systems; due to non-equilibrium effects turnaround in temperature is assumed to coincide with the onset of complete vapor disengagement.

$D_T$ , the vent diameter required for turbulent flow, is expressed by

$$D_T \cong \frac{3}{2} \left[ \frac{M_O \frac{dT}{dt} (\alpha_D - \alpha_O)}{F P_S (1 - \alpha_O)} \right]^{0.5} \left( \frac{C_{p,S}}{T_s} \right)^{0.25}$$

If  $G_O \left( \frac{D_T}{D_O} \right) \leq G_T \cong F \left( \frac{\Delta P}{\Delta T} \right) \left( \frac{T_s}{C_{p,S}} \right)^{0.5}$ , the required vent diameter for laminar flow  $D_L$  is given by

$$D_L = \left( D_T^2 D_O \frac{G_T}{G_O} \right)^{0.33}$$

### 9.117 GASSY SYSTEMS

The major method of vent sizing for gassy system is two-phase venting to keep the pressure constant. This method was employed before DIERS with an appropriate safety factor [114]. The vent area is expressed by

$$A = \frac{Q_g (1 - \alpha) \rho_f}{G_1} = \frac{Q_g M}{G V} \quad (9-114)$$

where

- $Q_g$  = volumetric gas generation rate at temperature and in reactor during relief,  $\text{m}^3/\text{s}$   
 $M$  = mass of liquid in vessel, kg  
 $\rho_f$  = liquid density,  $\text{kg/m}^3$   
 $G, G_1$  = mass vent capacity per unit area,  $\text{kg/s m}^2$   
 $\alpha$  = void fraction in vessel  
 $V$  = total vessel volume,  $\text{m}^3$ .

Unlike systems with vapor present, gassy systems do not have any latent heat to temper the reaction. The system pressure increases as the rate of gas generation with temperature increases, until it reaches the maximum value. The vent area could be underestimated, if sizing is dependent on the rate of gas generated at the set pressure. Therefore, it is more plausible to size the vent area on the maximum rate of gas generation. Homogeneous two-phase venting is assumed even if the discharge of liquid during venting could reduce the rate of gas generation even further. The vent area is defined by

$$A = 3.6 \times 10^{-3} Q_g \left( \frac{M_O}{VP_m} \right)^{0.5} \quad (9-115)$$

The maximum rate of gas generation during a runaway reaction is proportional to the maximum value of  $\frac{dP}{dt}$  and can be calculated from

$$Q_{g1} = \frac{M_O}{M_t} \left( \frac{V_t}{P_m} \frac{dP}{dt} \right) \quad (9-116)$$

An equation representing the relief behavior for a length  $L/D < 400$  as in the tempered system is given by [115]

$$M_O = \left[ V_p \Delta P_p \left\{ \frac{(D_p)^2 (M_s)(P_p)}{(2.07)(T_t)(dP/dt)} \right\}^2 \right]^{1/3} \quad (9-117)$$

where

$V_p$  = vessel total volume (gal)

$P_p$  = maximum allowable venting pressure (psia)

$M_s$  = sample mass used in a DIERS test or equivalent test, g

$\frac{dP}{dt}$  = pressure rise in test (psi/s)

$T_t$  = maximum temperature in test (K)

$P_{amb}$  = ambient pressure at the end of the vent line, psia

$\Delta P_p = P_p - P_{amb}$ , psi.

Equation (9-117) is a dimensional equation and the dimensions given in the parameters must be used.

### 9.118 HOMOGENEOUS TWO-PHASE VENTING UNTIL DISENGAGEMENT

The ICI [114] developed a method for sizing a relief system that accounts for vapor/liquid disengagement. They proposed that homogeneous two-phase venting occurs that increases to the point of disengagement. Furthermore, they based their derivations upon the following assumptions.

- Vapor phase sensible heat terms may be neglected.
- Vapor phase mass is negligible.
- Heat evolution rate per unit mass of reactants is constant (or average value can be used).
- Mass vent rate per unit area is approximately constant (or safe value can be used).
- Physical properties can be approximated by average values.

The vent area is given by

$$A = \frac{qV(\alpha - \alpha_o)}{Gv_f \left\{ \frac{h_{fg}v_f(\alpha - \alpha_o)}{v_{fg}(1 - \alpha_o)(1 - \alpha)} + C\Delta T \right\}} \quad (9-118)$$

where

$A$  = vent area, m<sup>2</sup>

$C$  = liquid specific heat, J/kg K

$G$  = mass vent capacity per unit area, kg/m<sup>2</sup> s

$h_{fg}$  = latent heat, J/kg

$q$  = self-heat rate, W/kg

$V$  = total vessel volume, m<sup>3</sup>

$\Delta T$  = temperature rise corresponding to the overpressure, K

$\alpha_o$  = initial void fraction

$\alpha$  = void fraction in vessel

$v_f$  = liquid specific volume, m<sup>3</sup>/kg

$v_{fg}$  = difference between vapor and liquid specific volumes, m<sup>3</sup>/kg.

It is valid if disengagement occurs before the pressure would have turned over during homogeneous venting, otherwise Eq. (9-118) gives an unsafe (too small) vent size. Therefore, it is necessary to verify that

$$q > \frac{GAh_{fg}v_f^2}{Vv_{fg}(1 - \alpha)^2} \quad (9-119)$$

is satisfied at the point of disengagement.

### 9.119 TWO-PHASE FLOW THROUGH AN ORIFICE

Sizing formulae for flashing two-phase flow through relief devices were obtained through DIERS. It is based upon Fauske's ERM, and assumes frozen flow (non-flashing) forms a stagnant vessel to the relief device throat. This is followed by flashing to equilibrium in the throat. The orifice area is expressed by

$$A = \frac{W}{C_D} \left\{ \left( \frac{xV_G}{kP_1} \right) + \left[ \frac{(V_G - V_L)^2 C_L T_1}{\lambda^2} \right] \right\}^{0.5} \quad (9-120)$$

where

$A$  = vent area, m<sup>2</sup>

$C_D$  = actual discharge coefficient

$C_L$  = average liquid specific heat, J/kg K

$k$  = isentropic coefficient

$P_1$  = pressure in the upstream vessel, N/m<sup>2</sup>

$T_1$  = temperature corresponding to relief actuation, K

$x$  = mass fraction of vapor at inlet

$V_G$  = specific volume of gas, m<sup>3</sup>/kg

$V_L$  = specific volume of liquid, m<sup>3</sup>/kg

$W$  = required relief rate, kg/s

$\lambda$  = latent heat, J/kg.

Theoretical rate is given by

$$W = A_f (2\Delta P \cdot \rho)^{0.5} \quad (9-121)$$

For a simple sharp-edged orifice, the value of  $C_D$  is well established (about 0.6). For safety relief valves, its value depends upon the shape of the nozzle and other design features. In addition, the value of  $C_D$  varies with the conditions at the orifice. For saturated liquid at the inlet, Eq. (9-120) simplifies to yield

$$A = \frac{W(V_G - V_L)(C_L T_1)^{0.5}}{C_D \lambda} = \frac{W}{C_D \left( \frac{dP}{dT} \right) \left( \frac{T_1}{C_L} \right)^{0.5}} \quad (9-122)$$

These equations are based upon the following assumptions.

- Vapor phase behaves as an ideal gas.
- Liquid phase is an incompressible fluid.
- Turbulent Newtonian flow.
- Critical flow. This is usually the case since critical pressure ratios of flashing liquid approach the value of 1.

It is recommended that a safety factor of at least 2.0 be used [114]. In certain cases, lower safety factors may be employed. The designer should consult the appropriate process safety section in the engineering department for advice.

### 9.120 CONDITIONS OF USE

- If Fauske's method yields a significantly different vent size, then the calculation should be reviewed.
- The answer obtained from the Leung's method should not be significantly smaller than that from Fauske's method.
- The ICI recommends a safety factor of 1–2 on flow or area. The safety factor associated with the inaccuracies of the flow calculation will depend on the method used, the phase nature of the flow, and the pipe friction. For two-phase flow, use a safety factor of 2 to account for friction or static head.
- Choose the smaller of the vent size from the two methods.

A systematic evaluation of venting requires information on

- the reaction – vapor, gassy, or hybrid
- flow regime – foamy or non-foamy
- vent flow – laminar or turbulent
- vent sizing parameters  $dT/dt$ ,  $\Delta P/\Delta T$ , and  $\Delta T$ .

It is important to ensure that all factors (e.g., long vent lines) are accounted for, independent of the methods used. Designers should ascertain that a valid method is chosen rather than the most convenient or the best one. For ease of use, the Leung's method for vapor-pressure systems is rapid and easy. Different methods should give vent sizes within a factor of 2. Nomographs give adequate vent sizes for long lines to  $L/D$  of 400, but sizes are divisible by 2 for nozzles. A computer software VENT has been developed to size two-phase relief for vapor, gassy, and hybrid systems.

### 9.121 DISCHARGE SYSTEM

#### DESIGN OF THE VENT PIPE

The nature of the discharging fluid is necessary in determining the relief areas. The DIERS and ICI techniques can analyze systems that exhibit "natural" surface active foaming and those that do not. The DIERS further found that small quantities of impurities can affect the flow regime in the reactor. In addition, a variation in impurity level could arise by changing the supplier of a particular raw material. Therefore, care is needed in sizing emergency relief on homogeneous vessel behavior, that is two-phase flow. In certain instances, pressure relief during a runaway reaction can result in three-phase discharge, if solids are suspended in the reaction liquors. Solids can also be entrained by turbulence caused by boiling/gassing in the bulk of the liquid. Caution is required in sizing this type of relief system; especially, where there is a significantly static head of fluid in the discharge pipe. Another aspect in the design of the relief system includes the possible blockage in the vent line. This could arise from the process material being solidified in cooler sections of the reactor. It is important to consider all discharge regimes when designing the discharge pipe work.

### 9.122 SAFE DISCHARGE

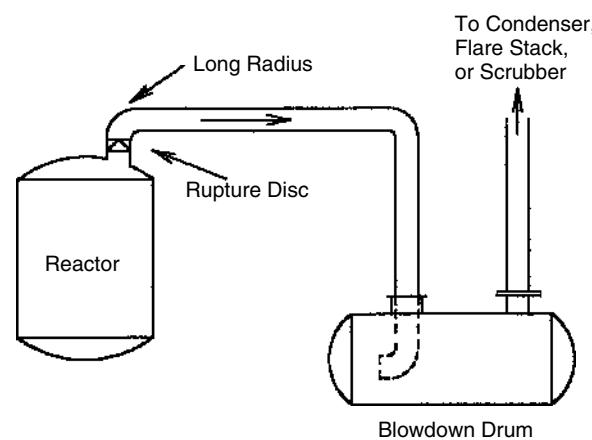
Reactors or storage vessels are fitted with overpressure protection vent directly to roof level. Such devices (e.g., relief valves) protect only against common process maloperations and not runaway reactions. The quantity of material ejected and the rate of discharge are low, resulting in good dispersion. The increased use of rupture (bursting) discs can result in large quantities (95% of the reactor contents) being discharged for foaming systems.

The discharge of copious quantities of chemicals directly to atmosphere can give rise to secondary hazards, especially if the materials are toxic, and can form a flammable atmosphere (e.g., vapor or mist) in air. In such cases, the provision of a knockout device (scrubber, dump tank) of adequate size to contain the aerated/foaming fluid will be required.

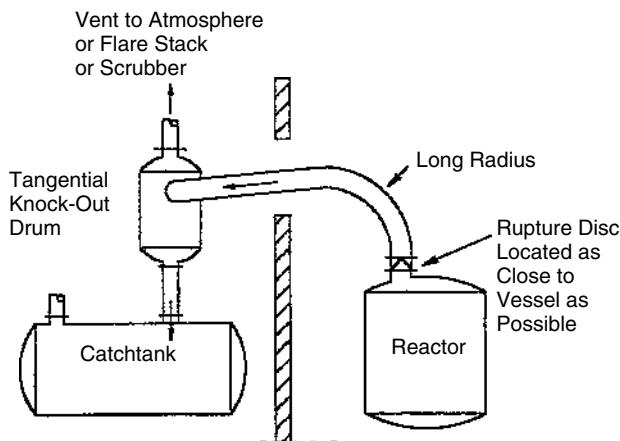
The regulatory authorities can impose restrictions on the discharge of the effluent from the standpoint of pollution. Therefore, reliefs are seldom vented to the atmosphere. In most cases, a relief is initially discharged to a knockout system to separate the liquid from the vapor. The liquid is collected and the vapor is then discharged to another treatment unit. The vapor treatment unit depends upon the hazards of the vapor, which may include a vent condenser, scrubber, incinerator, flare, or a combination of these units. This type of system is referred to as the total containment system as shown in Figure 9-97. The knockout drum is sometimes called a catch tank or blowdown drum. The horizontal knockout serves both as a vapor–liquid separator as well as a holdup vessel for the disengaged liquid. These types are commonly used where there is greater space as in the petroleum refineries and petrochemical plants. The two-phase mixture enters at one end, and the vapor leaves at the opposite end. Inlets may be provided at each end with a vapor outlet at the center of the drum to minimize vapor velocities involving two-phase streams with very high vapor flow rates. When there is limited space in the plant, a tangential knockout drum is employed as illustrated in Figure 9-98. Coker [117] has given detailed design procedures of separating gas–liquid separators.

### 9.123 DIRECT DISCHARGE TO THE ATMOSPHERE

Careful consideration and analysis by management are essential before flammable or hazardous vapors are discharged to the atmosphere. Consideration and analysis must ensure that the discharge



**Figure 9-97** Relief containment system with blowdown drum. The blowdown drum separates the vapor from the liquid. (Source: Grossel [116].)



**Figure 9-98** Tangential inlet knockout drum with separate liquid catch-tank. (Source: Grossel [116].)

can be carried out without creating a potential hazard or causing environmental problems. The possible factors are as follows.

- exposure of plant personnel and/or the surrounding population to toxic vapors or corrosive chemicals
- formation of flammable mixtures at ground level or elevated structures
- ignition of vapors at the point of emission (blowdown drum vent nozzle)
- air pollution.

These factors and methods for evaluating their effects are elaborated in API publications (API RP521) [5c and in 51,60].

We should take special care in the design of the vapor vent stack to ensure that the tip faces straight up (i.e., no goose-neck) in order to achieve good dispersion. The stack should not be located near a building so as to avoid vapor drifting into the building. However, if the drum is near a building, the stack should extend at least 12 ft above the building floor. Grossel [118] has provided various descriptions of alternative methods of disposal.

### EXAMPLE 9-23 Tempered Reaction

An 800-gal reactor containing a styrene mixture with a specific heat of 0.6 cal/gm°C has a 10-inch rupture disk and a vent line with equivalent length of 400. The vessel MAWP is 100 psig and the rupture disk set pressure is 20 psig. The styrene mixture had a self-heat rate of 60°C/min at 170°C as it is tempered in a DIERs venting test. What is the allowable reactor mixture charge to limit the overpressure to 10% over the set pressure?

*Solution*

Using Eq. (9-112), we have

$$M_O = \frac{(D_p)^2 (\Delta P_S)}{2.769 \left( \frac{dT}{dt} \right)_S} \cdot \left( \frac{T_S}{C_p} \right)^{0.5}$$

where

$M_O$  = allowable mass of the reactor mixture charge (kg) to limit the venting overpressure to  $P_p$  (psig).

$D_p$  = 10 in.

$P_S$  = the relief device set pressure = 20 psig

$P_p$  = maximum venting pressure =  $1.10(20) = 22.0$  psig

$\Delta P_S$  = the allowable venting overpressure (psi), that is the maximum venting pressure minus the relief device set pressure.  $(22.0 - 20) = 2$  psi

$T_S$  = the equilibrium temperature corresponding to the vapor pressure where the vapor pressure is the relief device set pressure (K) = 170°C  $(170 + 273.15 = 443.15$  K)

$\left( \frac{dT}{dt} \right)_S$  = the reactor mixture self-heat rate ( $^{\circ}\text{C}/\text{min}$ ) at temperature  $T_S$  (K) = 60°C/min

$C_p$  = specific heat of the reactor mixture (cal/g-K) = 0.6 cal/gm K.

$$M_O = \frac{(10^2)(2.0)}{(2.769)(60)} \left( \frac{443.15}{0.6} \right)^{0.5}$$

$$M_O = 32.72 \text{ kg}$$

The density of styrene is 0.9 g/cm³. Therefore, the quantity charged in the reactor is

$$\begin{aligned} &= \frac{(32.72) \text{ kg} (10^3) \text{ g}}{1 \text{ kg}} \times \frac{1 \text{ cm}^3}{0.9 \text{ g}} \times \frac{11}{10^3 \text{ cm}^3} \times \frac{1 \text{ gal}}{3.7851} \\ &= 9.6 \text{ gal} (\approx 10 \text{ gal}). \end{aligned}$$

The reactor charge is quite small for an 800-gal reactor. If the pressure is allowed to rise to 10% above MAWP, then

$$\begin{aligned} \Delta P_S &= 1.1(100) - 20 \\ &= 90 \text{ psi}. \end{aligned}$$

The amount charged is given by

$$\begin{aligned} M_O &= \frac{(10^2)(90)}{(2.769)(60)} \left( \frac{443.15}{0.6} \right)^{0.5} \\ M_O &= 1472.2 \text{ kg} \\ &= \frac{1472.2}{(0.9)(3.785)} \text{ gal} \\ &= 432 \text{ gal} \end{aligned}$$

This shows that a much larger initial charge will be required. An Excel spreadsheet program (EXAMPLE 9-23.xls) has been developed for this example.

**EXAMPLE 9-24**

A 700-gal reactor with a net volume of 850 gal containing an organic mixture has a 10-in. rupture disk and a vent line with an equivalent length  $L/D = 400$ . The vessel MAWP = 100 psig and the rupture disk set pressure is 20 psig. A venting test that was carried out showed that the reaction was "gassy." The test mass was 30 g, the peak rate of pressure rise was 550 psi/min, and the maximum test temperature was 300°C. Determine the allowable reactor mixture charge to limit the overpressure to 10% of the MAWP.

*Solution*

Using Eq. (9-117) for "gassy" reaction, the amount charge is

$$M_O = \left[ V_p \Delta P_p \left\{ \frac{(D_p)^2 (M_s) (P_p)}{(2.07) (T_r) (dP/dt)} \right\}^2 \right]^{1/3}$$

where

$$V_p = \text{vessel total volume (gal)} = 850 \text{ gal}$$

$$P_p = \text{maximum allowable venting pressure (psia)} \\ 1.10(100) + 14.7 = 124.7 \text{ psia}$$

$$M_s = \text{sample mass used in a Diers test or equivalent test, g} = 30 \text{ g}$$

$$\frac{dP}{dt} = \text{pressure rise in test (psi/sec)} = 550 \text{ psi/min} = 9.17 \text{ psi/s}$$

$$T_r = \text{maximum temperature in test (K)} (300^\circ \text{C} = 573.15 \text{ K})$$

$$\Delta P_p = P_p - P_{\text{amb}} = 124.7 - 14.7 = 110 \text{ psi}$$

$$P_{\text{amb}} = \text{ambient pressure at the end of the vent line} = 14.7 \text{ psia}$$

$$M_O = \left\{ (850)(110) \left[ \frac{(10)^2 (30) (124.7)}{(2.07)(573.15)(9.17)} \right]^2 \right\}^{1/3}$$

$$M_O = 479.9 \text{ kg.}$$

The amount charge to the reactor is 480 kg. A Microsoft Excel spreadsheet (Example 9-24.xls) has been developed for this example.

## 9.124 DIERS FINAL REPORTS

Refer to the bibliography at the end of this chapter for a listing of the final reports of this program [13].

## 9.125 FLARES/FLARE STACKS

Flares are useful for the proper disposal of waste or emergency released gas/vapors and liquids. The effects on the environment and the thermal radiation from the flare must be recognized and designed for. Flares may be "ground" flares or they may be mounted on a tall stack to move the venting away from immediate plant areas. Figure 9-99 illustrates a plant flare stack system. The flow noted "from processes" could also include pressure-relief valve discharges when properly designed for back pressure. This requires proper manifold design and, for safety, requires that the "worst case" volume condition be used particularly assuming that all relief devices discharge at the same time and any other process vents are also flowing. The piping systems sequence of entrance of the flare is important to back pressure determination for all respective relief devices.

The *knockout drums or separator tanks/pots* can be designed using the techniques offered in Chapter 6, and will not be repeated here. The API-RP 521 [9] specifies 20–30-min holdup liquid capacity from relief devices plus a vapor space for dropout and a drain volume.

The unit should have backup instrumentation to ensure liquid level control to dispose of the waste recovered liquid.

The *seal tank/pot* is not a separator but a physical liquid seal (Figure 9-100) to prevent the possibilities of backflash from the flare from backing into the process manifolds. It is essential for every stack design.

The back pressure created by this drum is an additive to the pipe manifold pressure drops and the pressure loss through the separator. Therefore, it cannot be independently designed and not "integrated" into the back pressure system. The flow capacity of the relief valve(s) must not be reduced due to back pressure on the valves' discharge side (outlet). The total back pressure of the

system must be limited to 10% of the set pressure of each pressure-relief valve that may be relieving concurrently [5]. When balanced relief valves are used, the manifold back pressure can be higher, less than 30% of the valve's set pressure, psia [5].

The key detail of a seal drum is the liquid seal:

$$h_1 = 144P''/\rho, \text{ see Figure 9-100} \quad (9-123)$$

where

$$h_1 = \text{seal, submerged, ft}$$

$$P'' = \text{maximum header exit pressure into seal, psig}$$

$$\rho = \text{density of seal liquid, lb/ft}^3$$

Calculate the cross section of the drum volume for vapor above the liquid level, establishing the level referenced to  $h_1$ , plus clearance to drum bottom,  $h_1$ , normally 12 in. to 18 in. This would be a segment (horizontal vessel) of a circle (see Appendix Table D-24). Reference [5] recommends that the cross section area of the vapor space above the liquid be at least equivalent to that of a circle diameter [(2) (inlet pipe)]. Thus, the cross-section area of vapor space  $A_s$  with equivalent diameter  $S$  should be at least [(2) ( $a_p$ )] where  $a_p$  is the cross-sectional area of the inlet pipe [5]. To avoid bubble burst slugging, Ludwig [34] suggests that the cross-sectional area of  $S$  be calculated to have a vapor velocity of less than the entrainment velocity for a mist-sized liquid particle, or that the vapor area be approximately one-third the cross-section area of the diameter of the horizontal drum. For a vertical drum, Reference [5] recommends that the vapor disengaging height be 3 ft.

When vacuum can form in the system due to condensing/cooling hot vapor entering, the seal drum liquid volume and possibly the seal drum diameter/length must be adjusted to maintain a seal when/if the seal fluid is drawn up into the inlet piping. A vacuum seal leg should be provided on the inlet 1.2 times the expected equivalent vacuum height in order to maintain a seal.

The following design points should be considered:

- Provide liquid low level alarms to prevent loss of liquid by evaporation, entrainment, leaks, or failure of the makeup liquid system.

**EXAMPLE 9-25**

Determine the vent size for a vapor-pressure system using the following data and physical properties.

Reactor parameters:

Volume,  $V = 10 \text{ m}^3$

Mass,  $M = 8000 \text{ kg}$

Vent opening pressure = 15 bara (217.5 psia)

Temperature at set pressure  $T_s = 170^\circ \text{C}$  (443.15 K)

Overpressure allowed above operating pressure = 10%  
 $= 1 \text{ bar } (10^5 \text{ Pa})$ .

Material properties:

Specific heat,  $C_p = 3000 \text{ J/kg K}$

Slope of vapor pressure and temperature curve  $dP/dT = 20,000 \text{ Pa/K}$

Rate of reaction  $\Delta T = dP/20,000 = 10^5/20,000 = 5 \text{ K}$

Rate of set pressure  $(dT/dt)_s = 6.0 \text{ K/min}$

Rate of maximum pressure = 6.6 K/min

Average rate = 6.3 K/min

$$= 0.105 \text{ K/s.}$$

*Solution*

Using Fauske's nomograph of Figure 9-95 at a self-heat rate of 6.3 K/min and a set pressure of 217.5 psia, the corresponding vent area per 1000 kg of reactants = 0.0008 m<sup>2</sup>.

The vent area of 8000 kg reactants = 0.0064 m<sup>2</sup>.

The vent size

$$d = \left( \frac{4\text{Area}}{\pi} \right)^{0.5} = 90.27 \text{ mm (3.6 in.)}$$

If Figure 9-95 is applicable for  $F = 0.5$ , then for  $F = 1.0$ , the area is given by

$$A = (0.0064 \text{ m}^2) \left( \frac{0.5}{1.0} \right) = 0.0032 \text{ m}^2$$

The area assumes a 20% absolute overpressure.

The result can be adjusted for other overpressures by multiplying the area by a ratio of 20/(new absolute percent overpressure).

Using the Leung's method:

Assuming  $L/D = 0$ ,  $F = 1.0$ , Two-phase mass flux from Eq. (9-102) gives

$$G = (0.9)(1.0)(20,000) \left( \frac{1.0 \times 443.15}{3000} \right)^{0.5}$$

$$= 6918.1 \frac{\text{kg}}{\text{m s}^2}$$

Rate of heat generation:  $q$

$$q = C_p \frac{dT}{dt} \left( \frac{\text{JK}}{\text{kg K s}} \right) = (3000)(0.105) \frac{\text{W}}{\text{kg}} = 315 \frac{\text{W}}{\text{kg}}$$

From Eq. (9-107), the vent area  $A$  is given by

$$A = \frac{8000 \times 315}{6918.1 \left\{ \left( \frac{10}{8000} \times 443.15 \times 20,000 \right)^{0.5} + (3000 \times 5.0)^{0.5} \right\}^2}$$

$$A = 7.024 \times 10^{-3} \text{ m}^2$$

$$d = 94.6 \text{ mm (3.72 in.)}$$

The vent size is about 4.0 in.

Using the Fauske's method:

Assuming  $\alpha = 1.0$ ,  $\alpha_O = 0.0$ ,  $F = 1.0$ ,

$$\Delta P = 1 \text{ bar} = 10^5 \text{ N/m}^2.$$

From Eq. (9-113), the vent area is

$$A = \frac{1}{2} \left[ \frac{(8000)(0.105)(1-0)}{(1.0) \left( \frac{443.15}{3000} \right)^{0.5} (10^5)(1-0)} \right]$$

$$A = 10.927 \times 10^{-3} \text{ m}^2$$

$$d = 117.95 \text{ mm (4.64 in.)}$$

- Use a sealing liquid that has a relatively low vapor pressure, and is not readily combustible, and will not readily freeze. Quite often glycol or mixtures are used. In freezing conditions the unit should receive personal inspection for condition of liquid.
- Provide overflow anti-siphon seal drains.
- Provide inlet vacuum seal legs.
- Some hydrocarbons may form gel clusters or layers with some sealant fluids; therefore, providing for cold weather heating and/or cleaning of the unit is necessary.
- Reference [5] suggests minimum design pressure for such a seal vessel of 50 psig, ASME code stamped (Ludwig [34]). Most flare seal drums operate at 0–5 psig pressure.
- Be extremely cautious and do not install light weight gauge glass liquid level columns. Rather use the heavier shatterproof style.
- Provide reliable seal liquid makeup, using liquid level gauging and monitoring with recording to ensure good records of perfor-

mance. The liquid level must be maintained; otherwise, the hazards of a bleed through or backflow can become serious.

**9.126 FLARES**

Flares are an attempt to deliberately burn the flammable safety relief and/or process vents from a plant. The height of the stack is important to the safety of the surroundings and personnel, and the diameter is important to provide sufficient flow velocity to allow the vapors/gases to leave the top of the stack at sufficient velocities to provide good mixing and dilution after ignition at the flare tip by pilot flames.

The API [5] discusses factors influencing flare design, including the importance of proper stack velocity to allow jet mixing. Stack gases must not be diluted below the flammable limit. The exit velocity must not be too low to allow flammable

**EXAMPLE 9-26**

A 3500-gal reactor with styrene monomer undergoes adiabatic polymerization after being heated inadvertently to 70°C. The MAWP of the reactor is 5 bara. Determine the relief vent diameter required. Assume a set pressure of 4.5 bara and a maximum pressure of 5.4 bara. Other data and physical properties are given as follows:

Data

$$\text{Volume } (V) = 13.16 \text{ m}^3 (3500 \text{ gal})$$

$$\text{Reaction mass } (m_o), \text{ kg} = 9500$$

$$\text{Set temperature } (T_s) = 209.4^\circ \text{C} = 482.5 \text{ K}$$

Data from VSP:

Maximum temperature ( $T_m$ )	$219.5^\circ \text{C} = 492.7 \text{ K}$
$\left(\frac{dT}{dt}\right)_s$	$29.6^\circ \text{C/min} = 0.493 \text{ K/s}$ (sealed system)
$\left(\frac{dT}{dt}\right)_m$	$39.7^\circ \text{C/min} = 0.662 \text{ K/s}$

Physical property data:

	4.5 bar set	5.4 bar set
$v_f, \text{m}^3/\text{kg}$	0.001388	0.001414
$v_g, \text{m}^3/\text{kg};$ ideal gas assumed	0.08553	0.07278
$C_p, \text{kJ/kg K}$	2.470	2.514
$\Delta H_v, \text{kJ/kg}$	310.6	302.3

Solution

The heating rate  $q$  is determined by Eq. (9-109)

$$q = \frac{1}{2} C_V \left[ \left( \frac{dT}{dt} \right)_s + \left( \frac{dT}{dt} \right)_m \right]$$

Assuming  $C_V = C_p$

$$\begin{aligned} q &= \frac{1}{2} (2.470 \text{ kJ/kg K}) [0.493 + 0.662] (\text{K/s}) \\ &= 1.426 \frac{\text{kJ}}{\text{kg s}} \end{aligned}$$

The mass flux through the relief  $G$  is given by Eq. (9-99), assuming  $L/D = 0$  and  $\psi = 1.0$

$$\begin{aligned} G &= \frac{Q_m}{A} = 0.9\psi \frac{\Delta H_v}{v_{fg}} \left( \frac{g_c}{C_p T_s} \right)^{0.5} \\ G &= \frac{(0.9)(1.0)(310660 \text{ J/kg}) [1 \text{ (Nm)} / \text{J}]}{(0.08553 - 0.001388) \text{ m}^3/\text{kg}} \\ &\quad \left\{ \frac{[1 \text{ (kg m/s}^2 \text{)} / \text{N}]}{(2470 \text{ J/kg K})(482.5 \text{ K}) [1 \text{ (Nm)} / \text{J}]} \right\}^{0.5} \\ &= 3043.81 \frac{\text{kg}}{\text{m}^2 \text{ s}} \end{aligned}$$

The relief area is determined by Eq. (9-108)

$$A = \frac{M_O q}{G \left[ \left( \frac{V}{M_O} \frac{\Delta H_v}{v_{fg}} \right)^{0.5} + (C_V \Delta T)^{0.5} \right]^2}$$

The change in temperature  $\Delta T$  is  $T_m - T_s$

$$\Delta T = 492.7 - 482.5 = 10.2 \text{ K}$$

$$A = \frac{(9500 \text{ kg})(1426 \text{ J/kg s}) [1 \text{ (Nm)} / \text{J}]}{(3043.81 \text{ kg/m}^2 \text{ s}) \left[ \left( \frac{13.16 \text{ m}^3}{9500 \text{ kg}} \right) \left( \frac{310660 \text{ J/kg} [1 \text{ (Nm)} / \text{J}]}{0.08414 \text{ m}^3/\text{kg}} \right) \right]^{0.5} + \left[ \left( \frac{2470}{\text{kg K}} \right) (10.2 \text{ K}) [1 \text{ (Nm)} / \text{J}] \right]^{0.5}}^2$$

$$A = 0.0839 \text{ m}^2$$

The required relief diameter is given by

$$\begin{aligned} d &= \left( \frac{4A}{\pi} \right)^{0.5} \\ &= \left[ \frac{(4)(0.0839 \text{ m}^2)}{3.14} \right]^{0.5} = 0.327 \text{ m} \\ d &= 327 \text{ mm (12.87 in.)} \end{aligned}$$

We shall consider the situation that involves all vapor relief. The size of a vapor phase rupture disk required is determined by assuming that all of the heat energy is absorbed by the vaporization of the liquid. At the set temperature, the heat release rate  $q$  is given by

$$\begin{aligned} q &= C_V \left( \frac{dT}{dt} \right)_s = \left( 2.470 \frac{\text{kJ}}{\text{kg K}} \right) \left( 0.493 \frac{\text{K}}{\text{s}} \right) \\ q &= 1.218 \frac{\text{kJ}}{\text{kg s}} \end{aligned}$$

The vapor mass flow through the relief is then

$$\begin{aligned} Q_m &= \frac{qm_o}{\Delta H_v} \\ &= \frac{(1218 \text{ J/kg})(9500 \text{ kg})}{(310660 \text{ J/kg})} \\ &= 37.25 \frac{\text{kg}}{\text{s}} \end{aligned}$$

The required relief area for vapor is determined by

$$A = \frac{Q_m}{C_o P} \left[ \frac{R_g T}{\gamma g_c M} \left( \frac{2}{\gamma + 1} \right)^{\frac{\gamma + 1}{\gamma - 1}} \right]^{0.5}$$

where

$Q_m$  = discharge mass flow, kg/s

$C_o$  = discharge coefficient

$A$  = area of discharge

$P$  = absolute upstream pressure

$\gamma$  = heat capacity ratio for the gas

$g_c$  = gravitational constant

$M$  = molecular weight of the gas ( $M = 104$  for styrene)

$R_g$  = ideal gas constant ( $8314 \text{ Pa m}^3/\text{kg mol K}$ )

$T$  = absolute temperature of the discharge.

(continued)

**EXAMPLE 9-26—(continued)**Assuming  $C_o = 1$  and  $\gamma = 1.32$ 

$$A = \frac{(37.25 \text{ kg/s})}{(1.0)(4.5 \text{ bar})(10^5 \text{ Pa/bar})[1(\text{N/m}^2)/\text{Pa}]} \\ \times \left\{ \frac{(8314 \text{ Pa m}^3/\text{kg mol})(482.5 \text{ K})[1(\text{N/m}^2)/\text{Pa}]}{(1.32)[1(\text{kg m/s}^2)/\text{N}](104 \text{ kg/kg mol})} \right\}^{0.5} \\ \times \left[ \left( \frac{2}{2.32} \right)^{\left( \frac{-0.32}{-0.32} \right)} \right]^{0.5}$$

$A = 0.0242 \text{ m}^2.$

The relief diameter = 0.176 m

$d = 176 \text{ mm (6.9 in.)}$

Thus, the size of the relief device is significantly smaller than for two-phase flow. Sizing for all vapor relief will undoubtedly give an incorrect result, and the reactor would be severely tested during this runaway occurrence. Table 9-41 shows the results of the VENT software program of Example 9-26.

Articles describing procedures related to the DIERS development of this entire subject have been published by some members of the DIERS group. These are referenced here and the detailed descriptions and illustrations in the noted articles can be most helpful to the potential user. A useful website for two-phase runaway flow system is [www.fauske.com](http://www.fauske.com).

**Table 9-41 Vent Sizing for Two-phase (Runaway Reactions) Flow for a Tempered System of Example 9-26**

Mass of reactant in the vessel, kg:	9500.000
Reactor volume, $\text{m}^3$ :	13.160
Slope of vapor pressure temp. curve: $\text{Nm}^{-2}\text{K}$ :	7649.734
Latent heat of vaporization, $\text{kJ/kg}$ :	.31060000E+03
Specific heat capacity of liquid, $\text{kJ/kg.K}$ :	2.470
Set temperature, $^\circ\text{C}$ :	209.400
Maximum temperature, $^\circ\text{C}$ :	219.500
Specific volume of gas, $\text{m}^3/\text{kg}$ :	0.085530
Specific volume of liquid, $\text{m}^3/\text{kg}$ :	0.001388
Difference between gas and liquid specific volume $\text{m}^3/\text{kg}$ :	0.084142
Self heat rate at set temperature, $\text{K/s}$ :	0.4930
Self heat rate at maximum temperature, $\text{K/s}$ :	0.6620
Mass flux per unit area, $\text{kg/m}^2\text{s}$ :	3043.067
Heat release rate per unit mass, $\text{kW/kg}$ :	1.426
Vent area, $\text{m}^2$ :	0.84579520E-01
Vent size, $\text{m}$ :	0.32816150E+00
Vent size, $\text{mm}$ :	0.32816150E+03
Vent size, in.:	0.12919750E+02

gases to fall to the ground and become ignited. The atmospheric dispersion calculations are important for the safety of the plant. Computer models can be used to evaluate the plume position when the flare leaves the stack under various atmospheric wind conditions. This should be examined under alternate possibilities of summer through winter conditions (also see [119]).

The velocities of the discharge of relief devices through a stack usually exceed 500 ft/s. Because this stream exits as a jet into the air, it is sufficient to cause turbulent mixing [5].

For a flare stack to function properly and to handle the capacity that may be required, the flows under emergency conditions from each of the potential sources must be carefully evaluated. These include, but may not be limited to, pressure-relief valves and rupture disks, process blow-down for startup, shutdown, upset conditions, and plant fires creating the need to empty or blowdown all or parts of a system.

## 9.127 SIZING

Diameter: sizing based on stack velocity [5c], solve for "d"

$$\text{Mach} = (1.702)(10^{-5}) \left( \frac{W}{P_t d^2} \right) \sqrt{\frac{T}{(kM)}} \quad (9-124)$$

In metric units,

$$\text{Mach} = (11.61)(10^{-2}) \left( \frac{W}{P_t d^2} \right) \sqrt{\frac{T}{(kM)}} \quad (9-125)$$

where

Mach = ratio of vapor velocity to sonic velocity in vapor, dimensionless. Mach = 0.5 for peak for short-term flow, and 0.2 for more normal and frequent conditions [5c]

$W$  = vapor relief rate to stack, lb/h ( $\text{kg/s}$ )

$P_t$  = pressure of the vapor just inside flare tips (at top), psia. For atmospheric release,  $P_t = 14.7$  psia (101.3 kPa)

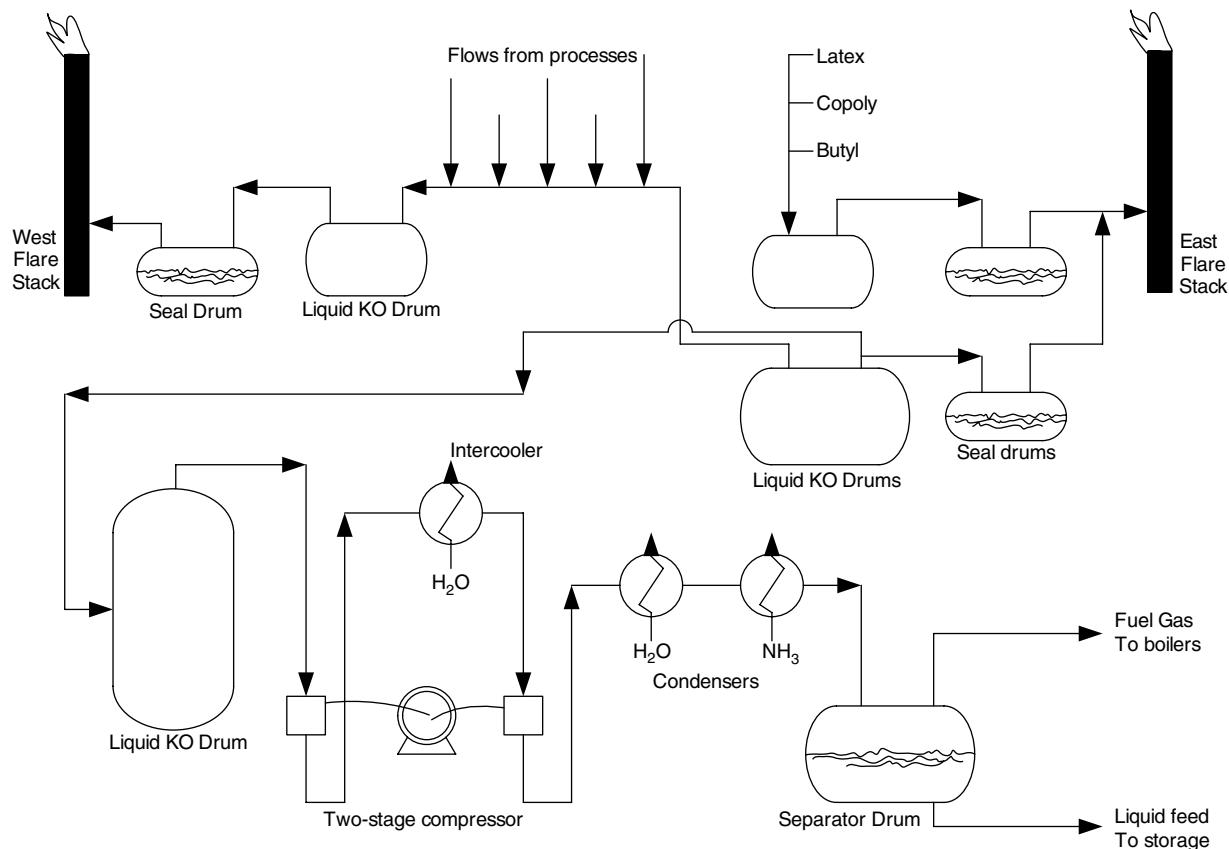
$d'$  = flare tip diameter, ft (m) (end or smallest diameter)

$T$  = temperatures of vapors just inside flare tip,  $^\circ\text{R} = ^\circ\text{F} + 460 (K = ^\circ\text{C} + 273)$

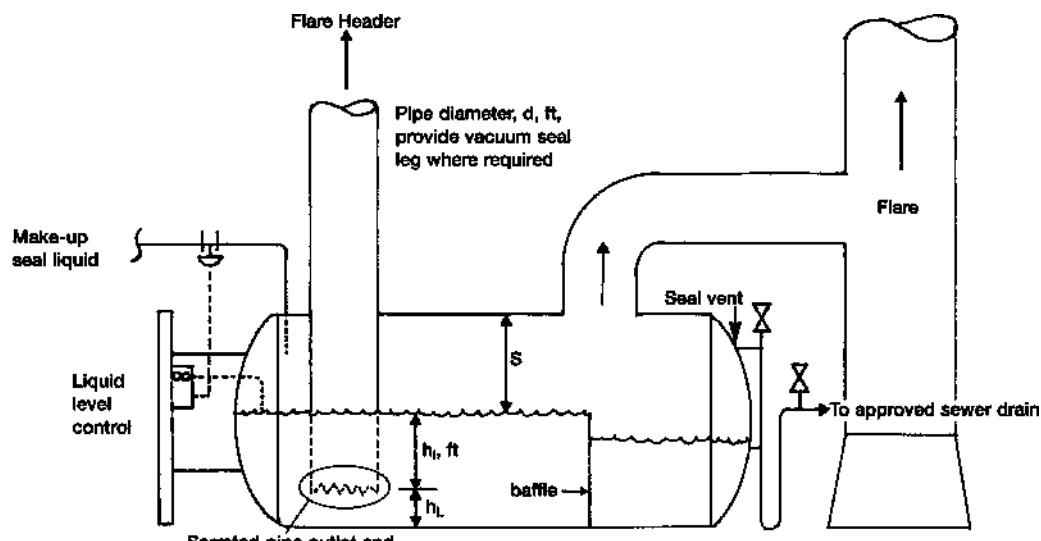
$k$  = ratio of specific heats,  $C_p/C_v$  for vapor being relieved

$M$  = molecular weight of vapor.

A peak velocity through the flare end (tip) of as much as 0.5 Mach is generally considered a peak, short term. A more normal steady

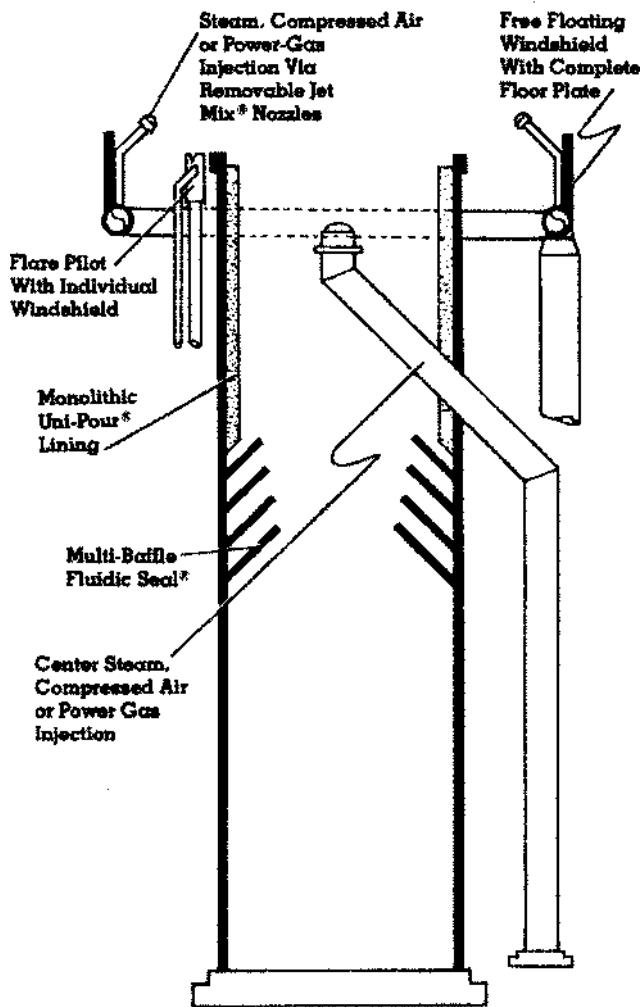


**Figure 9-99** Illustration of one of many collection arrangements for process flow and/or relief valve discharge collections to relieve to one or more plant flare stacks. (By permission from Livingston, D.D., *Oil Gas J*, Apr 28, 1980.)



$h_L$  = Seal pipe clearance to bottom, normally use 12 inch–18 inch  
 $h_f$  = Submergence, seal, ft  
 $S$  = Equivalent area diameter for vapor release

**Figure 9-100** Suggested seal pot/drum for flare stack system (see API RP – 521, Fig. B-1, 3rd ed. 1990). (Design adapted with permission from E.E. Ludwig [34] from API RP-521, 3rd ed. (1990) American Petroleum Institute [5].)

**NRC: NAC's Ring-and-Center Smokeless Flare**

**Figure 9-101** Flare stack arrangement for smokeless burning and back-flash protection with Fluidic Seal® molecular seal. Steam can be injected into the flare to introduce air to the fuel by use of jets inside the stream and around the periphery. (By permission from Straitz, J.F. III, "Make the Flare Protect the Environment", *Hydrocarbon Processing*, Oct 1977, p. 131.)

state velocity of 0.2 Mach is for normal conditions and prevents flare/lift off [120]. Smokeless (with steam injection) flare should be sized for conditions of operating smokeless, which means vapor flow plus steam flow [5c]. Pressure drops across the tip of the flare have been used satisfactorily up to 2 psi. It is important not to be too low and get flashback (without a molecular seal) or blow off where the flame blows off the tip (see [121]), Figure 9-101.

Another similar equation yielding close results [122]:

$$d_t^2 = (W/1370) \sqrt{T/M}, \text{ (generally for smokeless flares)} \quad (9-126)$$

based on Mach 0.2 limitation velocity,  $k = C_p/C_v = 0.2$  and gas constant  $R = 1545$  (ft. lbf/°R)(mole).

$d_t$  = flare tip diameter, in

$W$  = gas vent rate, lb/h

$T$  = gas temperature in stack, °R

$M$  = molecular weight of gas/vapor.

For non-smokeless flares (no steam injection) about 30% higher capacity can be allowed [122]. Therefore, the diameter of a non-smokeless flare stack is approximately 0.85 (diameter of the smokeless flare stack).

The amount of steam injection required for smokeless flares is:

$$W_{\text{steam}} = W_{\text{hc}} (0.68 - 10.8/M) \quad (9-127)$$

where

$W_{\text{steam}}$  = steam injected, lb/h

$W_{\text{hc}}$  = hydrocarbons to be flared, lb/h

$M$  = molecular weight of hydrocarbons (average for mixture, hydrocarbons only).

For specific details, consult a flare system design manufacturer.

This calculation is based on a steam- $\text{CO}_2$  weight ratio of approximately 0.7 [33A, Par. 5.4.3.2.1].

These should be sized for conditions under which they will operate smokelessly.

**9.128 FLAME LENGTH [5C]**

$$\text{Heat liberated or released by flame, } Q_f = (W) (H_c) \quad (9-128)$$

where

$Q_f$  = heat released by flame, Btu/h

$W_{\text{hc}}$  =  $W$  = gas/vapor flow rate, lb/h.

$H_c$  = heat of combustion of gas/vapor, Btu/lb

The heat liberated,  $Q$  in Btu/h (kW), is calculated as follows (see API Recommended Practice 521, Figures 9-6a and b):

$$Q = (100,000) (21,500)$$

$$= 2.15 \times 10^9 \text{ Btu/h}$$

In metric units,

$$Q = (12.6) (50 \times 10^3)$$

$$= 6.3 \times 10^5 \text{ kW.}$$

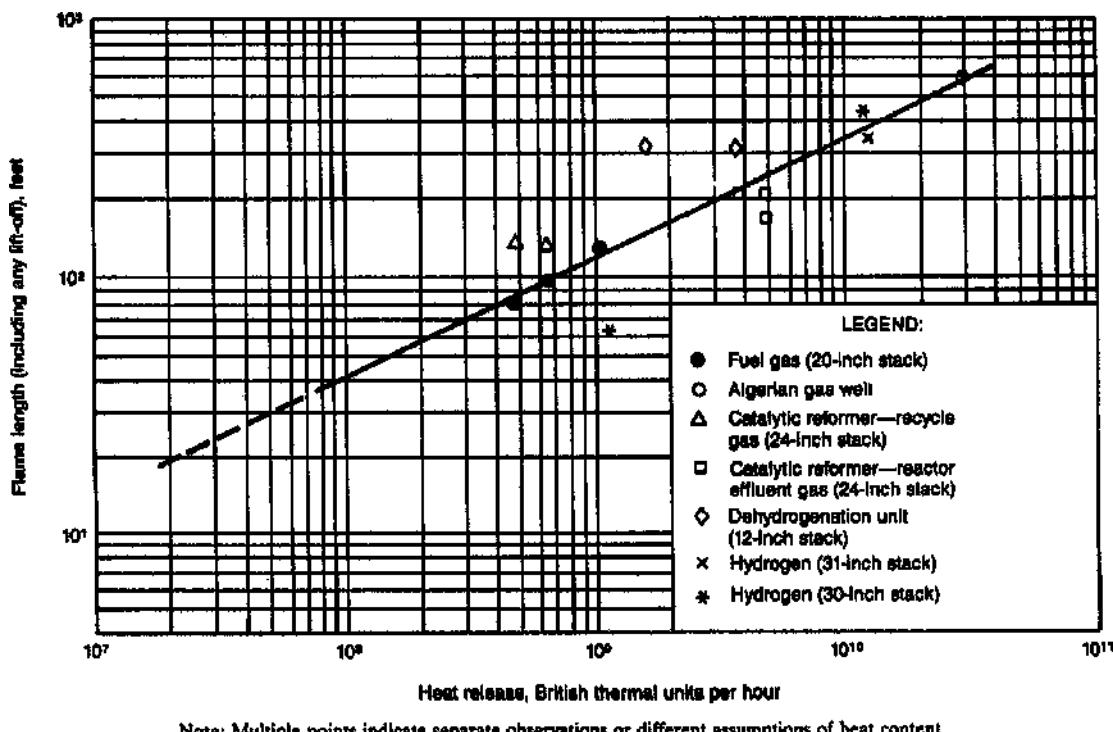
*Note:* For many hydrocarbon-air mixtures, the value of  $H_c$  ranges from 20,000 to 22,000 Btu/lb. Referring to Figure 9-102 at the calculated heat release,  $H_c$ , read the flame length, and refer to dimensional diagram for flame plume from a stack, Figure 9-103.

**9.129 FLAME DISTORTION [5C] CAUSED BY WIND VELOCITY**

Referring to Figure 9-104, the flame distortion is determined as  $\frac{\Delta x}{L}$  or  $\frac{\Delta y}{L}$ . Calculate  $U_j$  using the "d" determined for the selected Mach number in earlier paragraph.

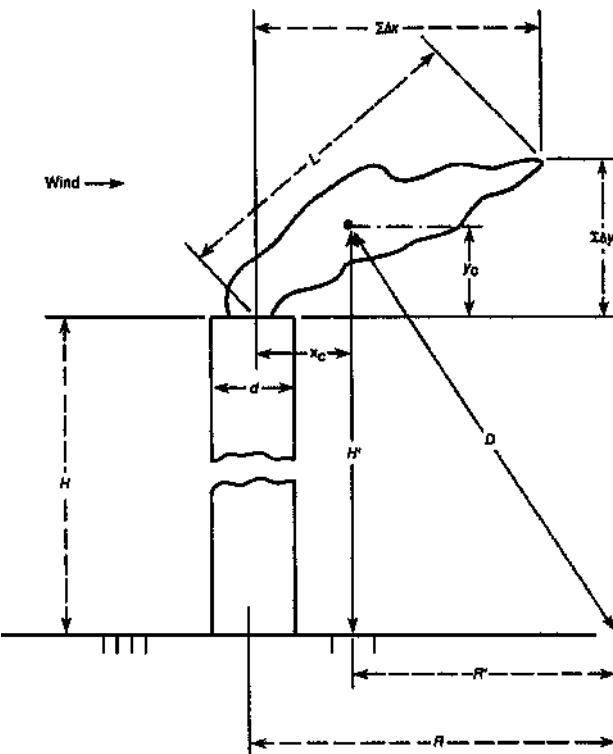
$$\frac{U_\infty}{U_j} = \frac{\text{wind velocity}}{\text{flare tip velocity}} \quad (9-129)$$

$$U_j = (\text{flow}) / (\pi d^2 / 4), \text{ ft/s} \quad (9-130)$$

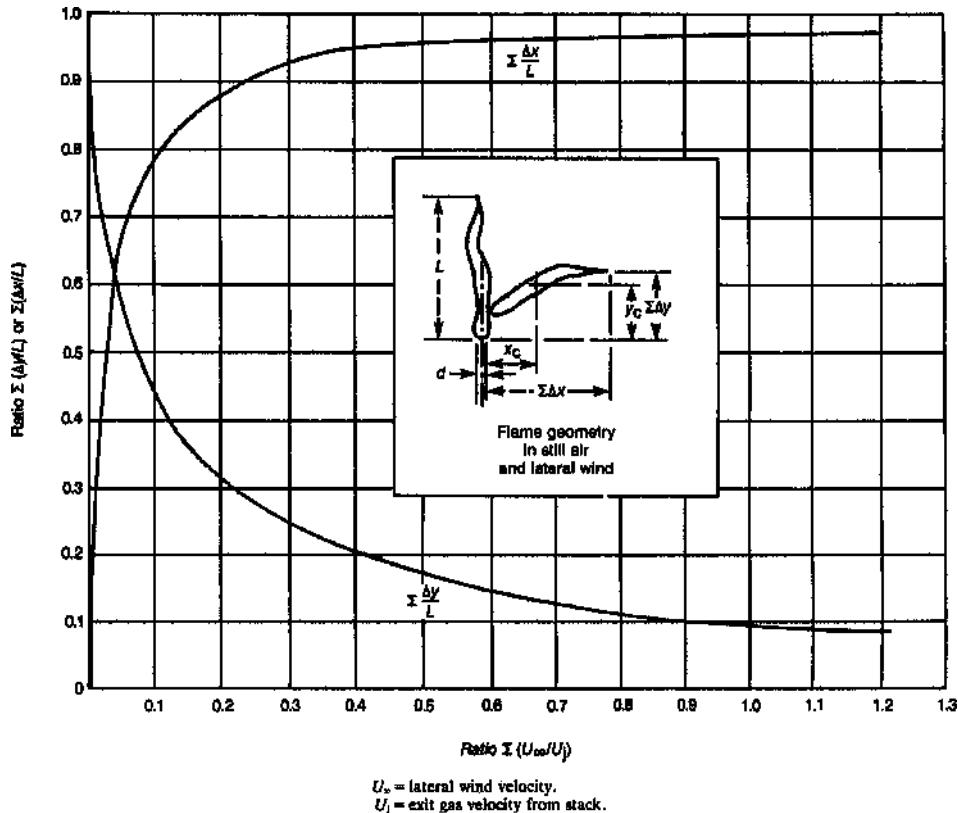


Note: Multiple points indicate separate observations or different assumptions of heat content.

**Figure 9-102** Flame length versus heat release: industrial sizes and releases (customary units). (Reprinted by permission from American Petroleum Institute, API RP-521, *Guide for Pressure Relieving and Depressurizing Systems*, 3rd ed. Nov 1990 [5].)



**Figure 9-103** Dimensional references for sizing a flare stack. (Reprinted by permission from American Petroleum Institute, API RP-521, *Guide for Pressure Relieving and Depressurizing Systems*, 3rd ed. Nov 1990 [36].)



**Figure 9-104** Approximate flame distortion due to lateral wind on jet velocity from flare stack. (Reprinted by permission from American Petroleum Institute, API RP-521, *Guide for Pressure Relieving and Depressuring Systems*, 3rd Nov 1990 [5].)

Flow,  $F_1 = (W/3600)(379.1/MW)(460 + {}^\circ\text{F}/520)\text{ft}^3/\text{s}$  based on  
 $60^\circ\text{F}$  and 14.7 psia, and  $359\text{ ft}^3/\text{mol}$

$U_\infty$  = lateral wind velocity, ft/s  
 $U_j$  = exit gas velocity from stack, ft/s  
 ${}^\circ\text{F}$  = flowing temperature

Kent [123] presents an alternate calculation method for flame distortion.

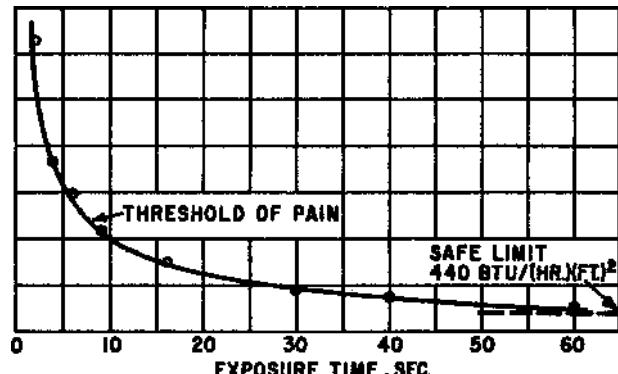
Read  $U_\infty/U_j$  on Figure 9-104 and determine ratio:

$\Delta y/L_f = a$  (vertical)  
 $\Delta x/L_f = b$  (horizontal)

Then, vertical :  $\Delta y = L_f(a)$   
 horizontal:  $\Delta x = L_f(b)$   
 $L_f$  = length of flame, ft

### 9.130 FLARE STACK HEIGHT

The importance of the stack height (see Figure 9-103) is (a) to discharge the burning venting gases/vapors sufficiently high into the air so as to allow safe dispersion, (b) to keep the flare flame of burning material sufficiently high to prevent the radiated heat from damaging equipment and facilities and from creating a life safety hazard to ground personnel. Figure 9-105 summarizes the accepted data for heat radiation related to human exposure time. Figure 9-106 summarizes the maximum radiation intensity related to a human escape time, allowing a 5-s reaction time to take action



**Figure 9-105** Heat radiation intensity vs. exposure time or bare skin at the threshold of pain. (By permission from Kent, *Hydrocarbon Processing*, Vol. 43, No. 8 (1964), p. 121 [123].)

to escape, before the heat intensity injures the individual. Kent [123] suggests an escape velocity of 20 ft/s. The heat radiation is an important factor in locating/spacing of equipment with respect to one or more flares. The use of protective clothing and safety hard hats aids in extending the time of exposure when compared to bare skin.

The distance required between a flare stack venting and a point of exposure to thermal radiation is expressed [5c], [121] as

$$D_F = \sqrt{\tau F Q_r / (4\pi K)} \quad (9-131)$$

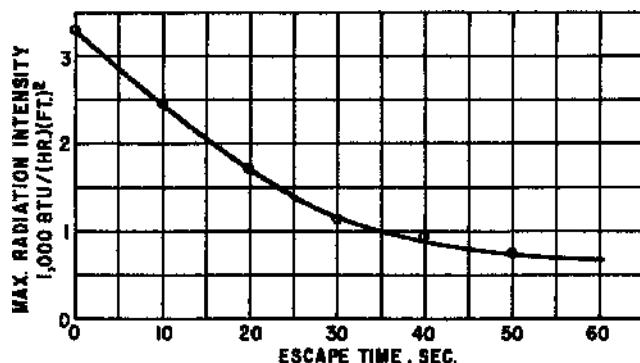


Figure 9-106 Maximum radiation intensity vs. escape time based on 5 s reaction time. (By permission from Kent, *Hydrocarbon Processing*, Vol. 43, No. 8 (1964), p. 121 [123].)

where

$D_F$  = minimum distance from the midpoint of a flame to the object, at ground level, ft (see Figure 9-103) (Note that this is not the flare stack height, but a part of calculation procedure)

$F$  = fraction of heat radiated.

This references to the total heat of combustion of a flame and selected values are [5c, 121] as follows.

Hydrocarbon	F range	F range average
Methane	0.10–0.20*	0.15
Natural Gas	0.19–0.23	0.21
Propane	—	0.33**
Butane	0.21 to 0.30	0.28
Hydrogen	0.10 to 0.17	0.15

\* 0.20 used for methane with carbon weight ratio of 0.333.

\*\* With weight ratio of 0.222.

When in doubt, to be safe, use 0.4 [121] or 1.0 [5c].

TABLE 9-42 Recommended Design Flare Radiation Levels Including Solar Radiation

Permissible Design Level ( $K$ )		
British Thermal Units per Hour per Square Foot	Kilowatts per Square Meter	Conditions
5000	15.77	Heat intensity on structures and in areas where operators are not likely to be performing duties and where shelter from radiant heat is available (for example, behind equipment).
3000	9.46	Value of $K$ at design flare release at any location to which people have access (for example, at grade below the flare or a service platform of a nearby tower); exposure should be eliminated to a few seconds, sufficient for escape only.
2000	6.31	Heat intensity in areas where emergency actions lasting up to 1 min may be required by personnel without shielding but with appropriate clothing.
1500	4.73	Heat intensity in areas where emergency actions lasting several minutes may be required by personnel without shielding but with appropriate clothing.
500	1.58	Value of $K$ at design flare release at any location where personnel are continuously exposed.

Note: On towers or other elevated structures where rapid escape is not possible ladders must be provided on the side, away from the flare, so the structure can provide some shielding when  $K$  is greater than 2000 Btu/h/ft<sup>2</sup>(6.31 kW/m<sup>2</sup>). Reprinted by permission from API RP-521, *Guide for Pressure Relieving and Depressuring Systems*, 3rd ed. Nov 1990, American Petroleum Institute [5].

$\tau$  = fraction heat intensity  $k$  transmitted through the atmosphere, usually assumed 1.0 (see later equation for modifying) [5c]

$Q_r$  = heat release (lower heating valve), Btu/h.

Kent [123] proposes total heat release:

$$Q_r = W \sum nh_c (379/M) \quad (9-132)$$

$$\text{or } (59) Q_n = 20,000 \text{ W}$$

where

$M$  = molecular weight

$h_c$  = net calorific heat value, Btu/scf

$h_c = 50M + 100$  for hydrocarbons Btu/scf (LHV) at 14.7 psia and 60°F

$h_c = \Sigma nh_c$  for gas mixtures, Btu/scf

$n$  = mol fraction combustion compound(s)

$f$  = fraction of radiated heat =  $0.20[h_c/900]^{1/2}$

$W$  = gas/vapors flow, lb/h

$K$  = allowable radiation, Btu/h/ft<sup>2</sup> (Table 9-42) Select acceptable value for "conditions assumed".

Reasonable heat intensity,  $K$ , values are 1500 Btu/h/ft<sup>2</sup>. When referred to Table 9-42

$F$  = fraction of heat radiated

$$\tau = 0.79(100/r)^{1/16} (100/D_F)^{1/16}, \text{ from [5c].}$$

$r$  = relative humidity, %.

When steam is injected at a rate of approximately 0.3 lb of steam per pound of flare gas, the fraction of heat radiated is decreased by 20%.  $\tau$  is based on hydrocarbon flame at 2240°F, 80°F dry bulb air, relative humidity > 10%, distance from flame between 100 and 500 ft, and is acceptable to estimate under wide conditions.

A slightly altered form of the  $D_F$  equation above [124, 125] for spherical radiation:

$$I = (\text{flow}) (\text{NHV}) (\varepsilon) / (4\pi D_F^2), \text{ Btu/h/ft}^2 \quad (9-133)$$

where

- $I$  = radiation intensity at point of object on ground level from midpoint of flame, Figures 9-103 and 9-104  
 Flow = gas flow rate, lb/h (or scfh)  
 NHV = net heating value of flare gas, Btu/lb, or (Btu/scf)  
 $\varepsilon$  = emissivity.

Height of stack for still air [123]:

$$H = \left( L^2 + \frac{fQ_r}{\pi q_M} \right)^{0.5} - L \quad (9-134)$$

The shortest stack exists when  $q_M = 3,300 \text{ Btu/h ft}^2$  (Figure 9-106). The limiting radial distance from the flame, allowing for speed of escape of 20 ft/s is given as [123]:

$$y = 20t_e = [x^2 - H(H+L)]^{0.5} \quad (9-135)$$

where

- $H$  = height of flare stack, ft, Figure 9-103  
 $L$  = height of flame (length of flame from top of stack to flame tip), ft  
 $D = X$  = radial distance from flame core (center) to grade, ft  
 $R = y$  = radial distance from base of stack, ft, to grade intersection with  $D$   
 $t_e$  = time interval for escape, s  
*Note:* For safety, personnel and equipment should be outside the "y" distance.  
 $R$  = distance from flame center to point  $X$  on ground (Figure 9-107).

This has been shown to be quite accurate for distances as close to the flame as one flame length [124].

#### Emissivity Values [125]

Carbon monoxide	0.075
Hydrogen	0.075
Hydrogen sulfide	0.070
Ammonia	0.070
Methane	0.10
Propane	0.11
Butane	0.12
Ethylene	0.12
Propylene	0.13
Maximum	0.13

Length of Flame [125] (see Figure 9-107)

$$L_f = 10(D)(\Delta P_t/55)^{1/2} \quad (9-136)$$

where

- $L_f$  = length of flame, ft  
 $D$  = flare tip diameter, in.  
 $\Delta P_t$  = pressure drop at the tip, in. of water.

This gives flame length for conditions other than maximum flow.

The center of the flame is assumed to be located a distance of one-third the length of the flame from the tip,  $L_f/3$  [125]. The flame angle is the vector addition of the wind velocity and the gas exit velocity.

$$V_{\text{exit, gas exit velocity}} = 550\sqrt{\Delta P_t/55}, \text{ ft/s} \quad (9-137)$$

From Figure 9-107,

$$\begin{aligned} X_c &= (L_f/3)(\sin \theta) \\ Y_c &= (L_f/3)(\cos \theta) \end{aligned}$$

$$\text{Distance, } R = \sqrt{(X - X_c)^2 + (H + Y_c)^2} \quad (9-138)$$

For the worst condition of gas flow and wind velocities, vertically below flame center,

$$\begin{aligned} R &= H + Y_c \\ H &= R - (L_f/3)(\cos \theta) \\ \theta &= \tan^{-1}(V_{\text{wind}}/V_{\text{gas exit}}) \end{aligned}$$

This assumes that the flame length stays the same for any wind velocity that is not rigidly true. With a wind greater than 60 mi/h, the flame tends to shorten. Straitz [125] suggests that practically this can be neglected.

Design values for radiation levels usually used are as follows [125].

1. Equipment protection: 3000 Btu/h/ft<sup>2</sup>
2. Personnel, short time exposure: 1500 Btu/h/ft<sup>2</sup>
3. Personnel, continuous exposure: 440 Btu/h/ft<sup>2</sup>
4. Solar radiation adds to the exposure, so on sunny days, continuous personnel exposure: 200–300 Btu/h/ft<sup>2</sup>.

Determine flare stack height above ground (grade):

Refer to Figure 9-103. Based on the Mach velocity of the vapor/gases leaving the top tip of the flare stack (see Eq. 9-124), determine the Mach number, for example 0.2, then from Figure 9-103

$$\text{where } H' = H + 1/2(\Delta y) \quad (9-139)$$

$$\text{and } R' = R - 1/2(\Delta x) \quad (9-140)$$

$\Delta y$  and  $\Delta x$  from previous calculations under flame distortion.

Refer to Table 9-42 and select the "condition" for radiation level,  $K$ , and ground distance,  $R$ , from stack.

Solve for  $R'$  using the ground distance selected,  $R$ , from stack, and use the  $\Delta x$  previously calculated.

Then, determine height of stack,  $H$ , by

$$D^2 = (R')^2 + (H')^2 \quad (9-141)$$

Substitute the previously calculated value of the distance from center of flame to grade  $D$  and also  $R'$ .

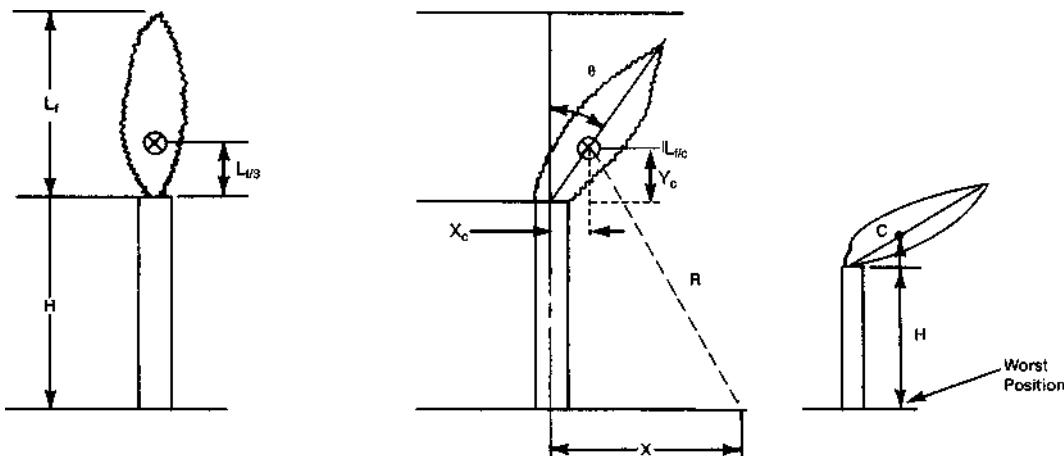
First solve for  $H'$ , then

$$\begin{aligned} H(\text{height of stack}) &= H' - 1/2(\Delta y) \\ &\quad (\text{previously calculated}) \end{aligned} \quad (9-142)$$

#### 9.131 PURGING OF FLARE STACKS AND VESSELS/PIPING

- Vacuum cycle
- Pressure cycle
- Continuous, flow through.

There are several different approaches to purging: Purging a system of flammable gas-vapor mixtures generally involves adding an inert gas such as nitrogen to the system. Sometimes the volumes of nitrogen are large, but it is still less expensive than most other non-flammable gas (even CO and CO<sub>2</sub> have to be used cautiously) and certainly air cannot be used because it introduces oxygen that could aggravate the flammability problem of flammability limits (also see [126]).



**Figure 9-107** Diagrams for alternate flare stack designs of Straitz. (By permission from Straitz, J.F. III and Altube, R.J., NAO, Inc. [125].)

### PRESSURE PURGING

The inert gas is added under pressure to the system to be purged. This is then vented or purged to the atmosphere, usually more than one cycle of pressurization followed by venting is necessary to drop the concentration of a specific flammable or toxic component to a pre-established level 1.

To determine the number of purge cycles and achieve a specified component concentration after "j" purge cycles of pressure (or vacuum) and relief [41]:

$$y_j = y_o (n_L/n_H)^j - y_o (P_L/P_H)^j \quad (9-143)$$

Repeat the process as required to decrease the oxidant concentration to the desired level.

where

$P_H$  = initial high pressures, mmHg

$P_L$  = initial low pressure or vacuum, mmHg

$y_o$  = initial concentration of component (oxidant) under low pressure, mol fraction

$n_H$  = number of mols at pressure condition

$n_L$  = number of mols at atmospheric pressure or low pressure conditions

$j$  = number of purge cycles (pressuring and relief)

$y_j$  = specified component concentration after "j" purges.

*Note:* The above equation assumes pressure limits  $P_H$  and  $P_L$  are identical for each cycle and the total moles of nitrogen added for each cycle is constant [41].

#### EXAMPLE 9-27

#### Purge Vessel by Pressurization Following the Method of [41]

A process vessel of 800-gal capacity is to have the oxygen content reduced from 21% oxygen (air). The system before process startup is at ambient conditions of 14.7 psia and 80°F. Determine the number of purges to reduce the oxygen content to 1 ppm ( $10^{-6}$  lb mol) using purchased nitrogen and used at 70 psig and 80°F to protect the strength of the vessel. How much nitrogen would be required?

Using Eq. (9-143):

$y_o$  = initial mol fraction of oxygen. This is now the concentration of oxygen at end of the first pressuring cycle (not venting or purging).

At high pressure pressurization:

$y_o = 21 \text{ lb mol oxygen}/100 \text{ total moles in vessel (initial)}$

$y_o = (0.21)(P_o/P_H)$ , composition for the high pressure condition

$P_o$  = beginning pressure in vessel, 14.7 psia

$P_H$  = high pressure of the purge nitrogen

$y_o = (0.21)[14.7/(70 + 14.7)] = 0.03644$

The final oxygen concentration  $y_f$  is to be 1 ppm ( $10^{-6}$  lb mol/total moles)

$$y_f = y_o (P_L/P_H)^j \quad (9-144)$$

$$10^{-6} = 0.21 [14.7/(70 + 14.7)]^j$$

Solving by taking the natural logarithms:

$$\ln[y_f/y_o] = j \ln[P_L/P_H]$$

$$\ln[10^{-6}/0.21] = j \ln[14.7/84.7]$$

$$j = 6.99 \text{ cycles}$$

Use seven minimum, perhaps use eight, for assurance that purging is complete. Note that the above relationships hold for vacuum purging. Keep in mind the relationships between high and low pressure of the system and use mmHg for pressure if it is more convenient. For sweep-through purging, see [41].

(continued)

**EXAMPLE 9-27—(continued)**

Total moles of nitrogen required [41]:

$$\begin{aligned}\Delta n_{N_2} &= j(P_H - P_L) \left[ V / (R_g T) \right] \quad (9-145) \\ &= 7.0(84.7 - 14.7)[(800/7.48)/10.3(80 + 460)] \\ &= 8.98 \text{ moles nitrogen}\end{aligned}$$

$$\text{lb nitrogen} = 8.98(28) = 125.72 \text{ lb}$$

$$V = 800 \text{ gal volume}$$

$$R_g = 10.73 \text{ psi ft}^3/\text{lb mol}^\circ\text{R}$$

$$T = \text{nitrogen temperature, } 80^\circ\text{F} + 460 = 540^\circ\text{R}$$

### 9.132 STATIC ELECTRICITY

Static electrical charges cause major damage in chemical and refining plants, yet they are not often recognized in the planning and design details of many plant areas. One of the least commonly recognized situations is that dusts generated in plant operations can be ignited to explosive violence by static electrical charges built up on the small particles. Of course, there are other dangers of explosions/fires being ignited by static discharges involving flammable vapor and mists and liquid particles (larger than mists) (also see [127]).

Although static discharges are small electrical phenomena, they are significantly different from a high voltage electrical discharge to ground from a power system or an arcing condition. These latter can certainly ignite dusts and vapors, but usually design efforts are made to properly insulate to prevent occurrences. Electrical codes aid in this design.

The use of intrinsically safe electrical and instrumentation equipment in appropriately designed environments can guard against many electrically related discharges. Reference should be made to authoritative books on this subject.

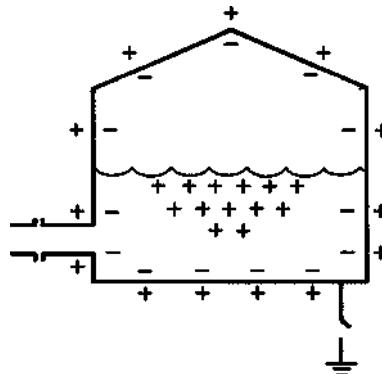
Static electricity is caused by the contact and separation of a good conductor material from a poor or non-conductor material or the separation of two non-conductor (or poor) materials. Static electricity is the accumulation of bound charges of the same sign and is prevented from reuniting quickly with charges of the opposite sign. These electrostatic phenomena are often characterized by the presence of high potential but small currents or charge quantities [128].

When two objects/particles separate after being in contact (equal charges), one particle loses electrons and becomes positively charged while the other gains electrons and becomes negatively charged.

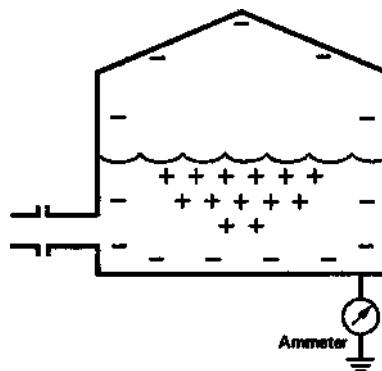
Low humidity allows the resistance of insulating surfaces to increase to a very high level, and this allows electrostatic charge separation and accumulation to occur [129]. Static electricity is usually present in some degree in many industrial situations, but ignitions caused by static discharge are preventable. The charging process arises at an interface between dissimilar materials, that is between hydrocarbons and metal or hydrocarbons and water [129]. The charge separation process occurs at the molecular level, but does not occur while the materials are in contact. When the charges are separated by moving the materials apart, the voltage potential rises. In a pipeline there is a "streaming current" established by charges off the inner pipe wall being carried in the fluid by the flow.

In storage or process tanks, a charge generation can occur if a liquid enters above the liquid surface by the spraying or splashing of the liquid and a charged mist may form [129] and the bulk liquid will become charged.

Figures 9-108a and b illustrate that when a charged fuel enters a metal tank, it attracts a charge of equal magnitude but opposite sign to the inside surface of the tank shell. At the same time a



**Figure 9-108a** Electrical charge induction in tank shell. (By permission from Busting and Dukek, *Electrostatic Hazards in the Petroleum Industry*, Research Studies Press, Somerset, England [129].)



**Figure 9-108b** Electrical-induced positive charge, grounded tank. (By permission from Busting and Dukek, *Electrostatic Hazards in the Petroleum Industry*, Research Studies Press, Somerset, England [129].)

charge of the same magnitude and sign as the charge in the fuel is repelled to the outside surface of the tank. Note that Bustin [129] states the choice of signs on the fuel is arbitrary.

If the tank is grounded, the repelled charge is neutralized; hence, the tank stays at zero voltage (see Figure 9-108b). The charge from this neutralization current is equal in magnitude and sign to the charge carried into the tank by the liquid. The ammeter is exactly equal to the "streaming" current entering the tank [129].

Inside the tank a voltage difference exists between the negative charge on the shell and the positive charge in the liquid. For a grounded tank, the voltage is zero at the shell. Note that grounding a closed metal tank has no effect on the voltage difference between

the two parts in the tank. Grounding a metal tank does not alter the risk of an electrostatic spark being generated within, but it does eliminate the possibility of an external spark discharge from the tank to ground.

Static electricity is classed as (a) spark discharges, and (b) corona discharges. The spark is a quick, instantaneous release of charge across an air gap from one “electrode source” to another. The corona is a discharge that branches in a diffuse manner, spreading over a large area of a poor conductor or ending in space [129]. The current (less) from a corona is weaker than that from a spark [129]. For a flammable mixture to ignite, the electrical discharge must release sufficient minimum energy to allow ignition to take place, and this minimum energy varies between flammable hydrocarbons and between dusts.

To avoid electrostatic discharging or even charging, the following list of conditions suggested by Haase [128] should be considered.

- electrostatic grounding of all conducting surfaces
- increasing the conductivity of the materials
- increasing the surface conductivity through the raising of the relative humidity or through surface treatment
- increasing the conductivity of the air
- relative low working speeds
- proper choice of contact materials
- proper control of the contact temperatures of the surfaces
- or a combination of the above.

Note that charges can be transported by persons or containers from a non-hazardous area into an unsuspecting unsafe (or hazardous) area and ignition cold then can take place [128].

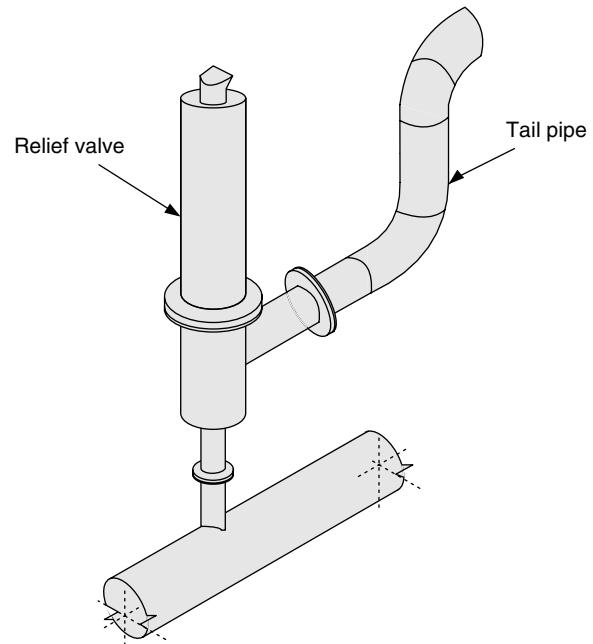
It is essential that the process hazardous atmosphere and the process system and handling of combustible hydrocarbons/chemicals be recognized in the physical designs by conforming to the appropriate class of atmosphere/environment codes specified by the National Electrical Code [63, 130, 131].

### 9.133 COMPRESSIBLE FLOW FOR DISCHARGE PIPING

The design of discharge piping or headers from relief valves for gases generally relates closely to isothermal conditions. Lapple [116] presented equations for compressible flow for both isothermal and adiabatic conditions. An important guideline for sizing discharge lines and headers from a relief valve using compressible fluids is to prevent the back pressure at relief valve outlets from reducing the fluid-relieving capacity of the valve and header system. Sometimes designs that do that can also cause vibration in the discharge lines and unacceptable noise.

Conversely, if the back pressure is excessive, the relief valve may fail to lift at its set pressure. Conventional relief valves tolerate back pressures up to 10% of their set pressures, while balance bellows type can tolerate up to 30–50% of set pressure. The capacity of the valves is reduced above these tolerances.

The design of relief valves is governed by well-established guides such as API RP-520 [8] which employs a kinetic energy correction factor and API RP-521 [5c] which relies on the Lapple chart. The limitation in successfully employing these methods is that they are based upon the unknown back pressure or header inlet pressure when the valve is discharging. Therefore, these methods often require a tedious trial-and-error solution. Figure 9-109 illustrates a typical discharge line (or tail pipe) from a safety relief valve.



**Figure 9-109** A typical relief valve and tail pipe.

### 9.134 DESIGN EQUATIONS FOR COMPRESSIBLE FLUID FLOW FOR DISCHARGE PIPING

The following equations are used to determine the pressure drop for compressible fluid flow.

The isothermal flow equation based on inlet pressure is given by

$$f_D \frac{L}{D} = \left( \frac{1}{M_1^2} \right) \left[ 1 - \left( \frac{P_2}{P_1} \right)^2 \right] - \ln \left( \frac{P_1}{P_2} \right)^2 \quad (9-147)$$

where

$$r = P_1 / P_2 \quad (9-148)$$

Substituting Eq. (9-148) into Eq. (9-147) gives

$$f_D \frac{L}{D} = \frac{1}{M_1^2} \left[ 1 - \frac{1}{r^2} \right] - \ln r^2 \quad (9-149)$$

Re-arranging Eq. (9-149) as a function of  $r$  gives

$$F(r) = M_1^2 \left( f_D \frac{L}{D} \right) - 1 + \frac{1}{r^2} + M_1^2 \ln r^2 \quad (9-150)$$

The isothermal flow equation based on outlet pressure

$$f_D \frac{L}{D} = \left( \frac{1}{M_2^2} \right) \left( \frac{P_1}{P_2} \right)^2 \left[ 1 - \left( \frac{P_2}{P_1} \right)^2 \right] - \ln \left( \frac{P_1}{P_2} \right)^2 \quad (9-151)$$

where

$$r = \frac{P_1}{P_2}$$

Substituting Eq. (9-148) into Eq. (9-151) gives

$$f_D \frac{L}{D} = \left( \frac{1}{M_2^2} \right) r^2 \left[ 1 - \frac{1}{r^2} \right] - \ln r^2 \quad (9-152)$$

Re-arranging Eq. (9-152) as a function of  $r$  gives

$$F(r) = r^2 - 1 - M_2^2 \ln r^2 - M_2^2 f_D \frac{L}{D} \quad (9-153)$$

where

- $D$  = header diameter, ft (meters)
- $f_D$  = Moody (Darcy) friction factor
- $L$  = header equivalent length, ft (meters)
- $M_1$  = Mach number of the inlet pipe
- $M_2$  = Mach number of the outlet pipe
- $P_1, P_2$  = inlet and outlet header pressures, psia (kPa).

$$\begin{aligned} M &= \frac{v_g}{v_s} \\ v_g &= \frac{W}{\rho} \quad \text{or} \quad v_g = \frac{0.0509 W}{(\rho)(d^2)} \\ v_s &= 223 \left( \frac{kT}{M_w} \right)^{0.5} \quad \text{or} \quad v_s = 68 \left( \frac{P_1}{\rho} \right)^{0.5} \\ v_g &= \left( \frac{W}{A} \right) \left( \frac{ZRT}{PM_w} \right) \end{aligned} \quad (9-154)$$

The Mach number can be determined by

$$M = \left( \frac{W}{A} \right) \left( \frac{ZRT}{PM_w} \right) \left( \frac{1}{233} \right) \left( \frac{M_w}{T} \right)^{0.5} \quad (9-156)$$

or

$$M = 0.00001336 \left( \frac{W}{PA} \right) \left( \frac{ZRT}{M_w} \right)^{0.5} \quad (9-157)$$

The outlet Mach number is given by

$$M_2 = 1.702 \times 10^{-5} \left( \frac{W}{P_2 D^2} \right) \left( \frac{ZT}{kM_w} \right)^{0.5} \quad (9-158)$$

In S.I units:

$$M_2 = 3.23 \times 10^{-5} \left( \frac{W}{P_2 D^2} \right) \left( \frac{ZT}{kM_w} \right)^{0.5} \quad (9-159)$$

where

- $A$  = pipe internal cross-sectional area,  $\text{ft}^2$
- $W$  = gas flow rate,  $\text{lb/h}$  ( $\text{kg/h}$ )
- $Z$  = gas compressibility factor
- $T$  = flowing temperature,  $^\circ\text{R}$  ( $^\circ\text{F} + 460$ ),  $K = (273.15 + ^\circ\text{C})$
- $M_w$  = gas molecular weight
- $P$  =  $P_1$  or  $P_2$  depending on input parameter, psia (kPa).
- $R$  = Individual gas constant =  $MR/M_w = 1545/\text{MW}$
- $MR$  = Universal gas constant

In S.I units

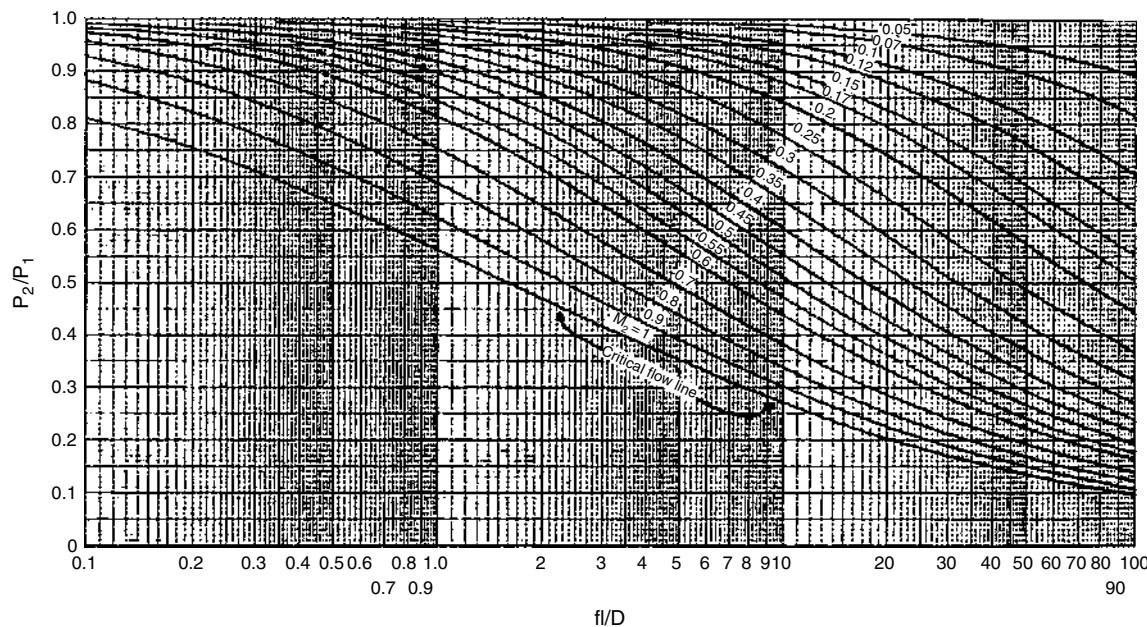
- $R$  = Individual gas constant =  $R_o/M_w$ ,  $\text{J/kg K}$
- $R_o$  = Universal gas constant =  $8314 \text{ J/kg mol K}$

Both graphical and computerized methods have been developed for solving Eq. (9-147) and (9-151) and calculating pipe inlet pressure [132, 133]. Figure 9-110 gives a typical graphical representation of Eq. (9-147). The figure may be used to calculate the inlet pressure,  $P_1$ , for a line segment of constant diameter where the outlet pressure is known.

#### CRITICAL PRESSURE, $P_{\text{CRIT}}$

If both high and low pressure-relief valves need to relieve simultaneously, parallel high and low pressure headers terminating at the flare knockout drum are the economical choice. It is essential to check for critical flow at key points in the high pressure header. The critical pressure at the pipe outlet can be determined by setting  $M_2 = 1.0$  (sonic flow) in Eq. 9-152 as follows.

$$P_{\text{critical}} = 1.702 \times 10^{-5} \left( \frac{W}{D^2} \right) \left( \frac{ZT}{kM_w} \right)^{0.5} \quad (9-160)$$



**Figure 9-110** Isothermal flow chart. (From Mah, H.Y. [132].)

or

alternatively, the critical pressure can be expressed by

$$P_{\text{crit}} = \left( \frac{W}{408d^2} \right) \left( \frac{ZT}{M_w} \right)^{0.5} \quad (9-161)$$

where

$P_{\text{crit}}$  = critical pressure, psia

$W$  = gas flow rate, lb/h

$d$  = pipe diameter, in.

$Z$  = gas compressibility factor

$T$  = gas temperature, °R

$M_w$  = as molecular weight.

In metric units:

$$P_{\text{critical}} = 3.23 \times 10^{-5} \left( \frac{W}{D^2} \right) \left( \frac{ZT}{kM_w} \right)^{0.5} \quad (9-162)$$

where  $P_{\text{critical}}$  is critical pressure, psia, (kpa)

If the critical pressure is less than the pipe outlet pressure, the flow is subsonic. If the critical pressure is greater than the pipe outlet pressure, the flow is sonic and  $M_2 = 1$ . Therefore, the pipe inlet pressure,  $P_1$ , is calculated from Eq. (9-147) with  $P_2$  equal to the critical pressure.

The specific gravity is defined by

$$S_{60g} = \frac{\rho_{60g}}{\rho_{60a}} = \frac{M_g}{M_a} \quad (9-163)$$

### 9.135 COMPRESSIBILITY FACTOR Z

Compressibility factors ( $Z$ ) are available in charts or tables as a function of pseudo-reduced temperatures and pressures,  $T_r$  and  $P_r$ , as shown in Chapter 3. Use of these charts is often time consuming and sometimes requires difficult calculations. The following equations allow the compressibility factor to be determined. This method gives a compressibility factor to within 5% for natural hydrocarbon gases with specific gravities between 0.5 and 0.8 and for pressures up to 5000 psia. Equation (3-38) gives the compressibility factor  $Z$  as:

$$Z = F_1 \left\{ \frac{1}{\left[ 1 + \frac{\left( A_6 P \times 10^{(1.785S_g)} \right)}{T^{3.825}} \right]} + F_2 F_3 \right\} + F_4 + F_5$$

where

$$F_1 = P (0.251S_g - 0.15) - 0.202S_g + 1.106$$

$$F_2 = 1.4e^{-0.0054(T-460)}$$

$$F_3 = A_1 P^5 + A_2 P^4 + A_3 P^3 + A_4 P^2 + A_5 P$$

$$F_4 = (0.154 - 0.152S_g) P^{(3.18S_g - 1.0)} e^{(-0.5P)} - 0.02$$

$$F_5 = 0.35 \left\{ (0.6 - S_g) e^{[-1.039(P - 1.8)^2]} \right\}.$$

The values of the constants  $A_1, A_2, A_3, A_4, A_5$ , and  $A_6$  are as follows.

$$A_1 = 0.001946$$

$$A_2 = -0.027635$$

$$A_3 = 0.136315$$

$$A_4 = -0.23849$$

$$A_5 = 0.1055168$$

$$A_6 = 3.44 \times 10^8.$$

The specific gravity of natural gas can be calculated from its density or molecular weight. This is expressed as the ratio of the gas density at 60°F and 1 atm (14.7 psia),  $\rho_{\text{gas}, 60^\circ\text{F}}$ , to the density of air,  $\rho_{\text{air}, 60^\circ\text{F}}$ , under the same conditions.

$$S_g = \frac{\text{density of gas}}{\text{density of air}}$$

$$= \frac{\rho_{\text{gas}, 60^\circ\text{F}}}{\rho_{\text{air}, 60^\circ\text{F}}}$$

Using the molecular weight of the gas,  $S_g$ , can be expressed as

$$S_g = \frac{\text{molecular weight of gas}}{\text{molecular weight of air}} = \frac{M_{w,gas}}{M_{w,air}}$$

Average molecular weight and viscosity:

$$M_w = \sum W / \sum (W/M_w) \quad (9-164)$$

$$T = \sum W_i T / \sum W_i \quad (9-165)$$

$$\mu = \sum x_i \mu_i (M_w)_i^{0.5} / \sum x_i (M_w)_i^{0.5} \quad (9-166)$$

Friction factor,  $f$

The explicit equation for the friction factor (Chen friction factor,  $f_C$ ) is expressed by Eq. (4-35):

$$\frac{1}{\sqrt{f_C}} = -4 \log \left[ \frac{\varepsilon}{3.7D} - \frac{5.02}{Re} \log A \right]$$

where

$$A = \frac{\varepsilon/D}{3.7} + \left( \frac{6.7}{Re} \right)^{0.9}$$

$\varepsilon$  = pipe roughness, ft

The Darcy friction factor  $f_D = 4f_C$ .

The Newton–Raphson method is employed to solve the iterative process from the ratio of the inlet and the outlet pressures. This is expressed in the form

$$X_{i+1} = X_i - \frac{F(X_i)}{F'(X_i)} \quad (9-167)$$

$X_i$  is the guessed or assumed root of the equation given by  $F(X) = 0.0F(X_i)$  is the value of the objective function.  $F'(X_i)$  is the value of the differential of the objective function. The  $i$  is the iteration counter, and  $i_{\text{max}}$  is the maximum iteration.

where  $i = 1, 2, 3, \dots, i_{\text{max}}$ .

For isothermal flow equation based on the inlet and the outlet Mach numbers, differentiating Eqs (9-150) and (9-153) with respect to  $r$  gives

$$F'(r) = -\frac{2}{r^3} + \frac{2M_1^2}{r} \quad (9-168)$$

$$F'(r) = 2r - \frac{2M_2^2}{r} \quad (9-169)$$

A computer program (KAFLO [133]) has been developed to calculate the pressure drop ( $\Delta P$ ), and Mach number for compressible fluid flow in a network of connecting pipes from a pressure-relief valve or valves. The program is based on the assumptions that the flow of gas through the discharge lines is isothermal, and that either the inlet or exit pressure is known. The Mach numbers are evaluated at both the inlet and the outlet. The types of pipe fittings

are incorporated in the program for selection. The program displays a message if the exit Mach number is greater than 0.7, signifying that the outlet gas velocity is too close to sonic velocity (i.e., the pipe size is too small). A larger pipe size is then required before the program proceeds to calculate the Reynolds number and the pressure drop in the pipe system.

### 9.136 DISCHARGE LINE SIZING

The following steps are used to size flare manifolds and relief valve blowdown systems:

1. The design starts at the flare tip where the outlet pressure is atmospheric. The calculation is worked back toward each relief valve in the system.
2. A size is assumed for each pipe section, and the maximum allowable velocity at each section inlet and outlet corresponds to a Mach number of 0.7. This criterion is applied to avoid pipe vibration and noise generation caused by excess velocity in the lines.
3. Properties in the common headers may be estimated from the relationships given in Eqs (9-164), (9-165), and (9-166). In those equations,  $i$  is the  $i^{\text{th}}$  component.
4. The inlet pressure is calculated for each section of the line. At each downstream line,  $P_1$  is taken as the outlet pressure of the upstream line,  $P_2$ , and a new upstream pressure,  $P_1$ , is calculated. The operation is repeated, working back toward each relief valve.
5. The maximum allowable back pressure, MABP, is taken as 10% of the set pressure for conventional relief valves, and 40% of the set pressure for balanced-bellows relief valves.
6. Check all relief valves against their MABP.
  - Case I The calculated back pressure at the lowest set relief valve on a header is much smaller than its MABP. Reduce the header size.
  - Case II The calculated back pressure at the lowest set relief valve on a header is close to and below its MABP. The header size is correct.
  - Case III The calculated back pressure at the lowest set relief valve on a header is above its MABP. Increase the header size.
7. If there is a great difference between the calculated back pressure and the MABP, the longest header should be decreased in size until the calculated back pressure is close to the MABP.

### 9.137 VENT PIPING

In general, the discharge piping or tail pipe should be as direct and as vertical as possible. Horizontal runs and elbows should be limited, or at best avoided.

Vent lines must never have any pockets (traps), and valves should never be installed between the relieving device and the vessel or system it is protecting. Pipe fittings should be kept to a minimum.

If in doubt about pipe size, use a size larger.

### 9.138 DISCHARGE REACTIVE FORCE

When a pressure-relief valve discharges without supported discharge piping, its discharge will impose a reaction force due to the flowing fluid. This force will be transmitted into the valve structure, the mounting nozzle, and supporting vessel shell. All reactive loading and resulting stresses are dependent on the reaction force and the piping configuration. Therefore, the designer must ensure that the reaction forces and associated bending moments will not cause excessive stresses on the system's components. Furthermore the location of an elbow and any support in the discharge system to direct the fluid up into a vent pipe must be considered in the analysis of the bending moments.

The following equation is based on the condition of critical steady state of a compressible fluid that discharges to the atmosphere through an elbow and a vertical discharge pipe. The reaction force  $F$  includes the effects of both momentum and static pressure. For any gas or vapor

$$F = \frac{W \sqrt{\frac{kT}{(k+1)M}}}{366} + (A_o P_2) \quad (9-170)$$

where

$F$  = reaction force at the point of discharge to the atmosphere, pounds (Newtons)

$W$  = flow of any gas or vapors, lb/h (kg/s)

$k$  = ratio of specific heats ( $C_p/C_v$ )

$C_p$  = specific heat at constant pressure

$C_v$  = specific heat at constant volume

$T$  = temperature at inlet  $^{\circ}\text{R}$  =  $^{\circ}\text{F} + 460$

$M$  = molecular weight of the process fluid

$A_o$  = area of the outlet at the point of discharge, in.<sup>2</sup> (mm<sup>2</sup>)

$P_2$  = static pressure at the point of discharge, psig (barg).

#### EXAMPLE 9-28

Size the tail pipe for steam flowing through a 6-in. Sch. 40 pipe from a pressure-relief valve at a set pressure of 110.4 psig, under the following conditions:

#### Parameters

Flow rate, lb/h	20,000
Temperature, $^{\circ}\text{F}$	320.0
Outlet pressure, psia	14.7
Viscosity, cP	0.0144
Ratio of specific heats ( $C_p/C_v$ )	1.3
Compressibility factor, $Z$	1
Molecular weight, $M_w$	18
Gas density, lb/ft <sup>3</sup>	0.0455
Pipe length, ft	12

Pipe fittings	Number
90° ell (long radius)	1
45° ell (long radius)	1
Entrance	1
Exit	1

#### Solution

A computer program KAFLO for sizing the discharge line or tail pipe of a relief valve or valves for compressible, isothermal gas flow is employed to solve Example 9-28. The program uses the Newton-Raphson method (Eq. 9-167) to solve the iterative process of Eqs (9-168) and (9-169), using  $r = P_1/P_2 = 1.0$  as the starting default value. The KAFLO calculates the pressure drop, the inlet and the outlet Mach numbers, Reynolds number, friction factor, the total length of pipe (using the 2-K method to determine the equivalent length of pipe from pipe fittings) and summing with the straight length of pipe, upstream gas velocity, sonic velocity, and critical pressure. It determines whether fluid flow is sonic or subsonic.

Table 9-43 shows the results of the computer program.

**TABLE 9-43 Computer Results of Example 9-28**

Line	
Normal size, in.	6
Schedule number:	40
Internal diameter: $d$ , in.	6.065
Flow rate: $W$ , lb/hr.	20,000
Compressibility factor: $Z$	1
Gas density: $\rho$ , lb/ft <sup>3</sup>	0.042
Gas viscosity: $\mu$ cP	0.0144
Ratio of specific heats: $k$ , ( $C_p/C_v$ )	1.32
Gas molecular weight: $M_w$ , lb/lb. mole	18.0
Gas temperature: °F	320
Actual pipe length: $L$ , ft.	12
$r = (P_1/P_2 = 1)$	
Equivalent length of pipe: $L_{eq}$ , ft.	62.549
Gas Reynolds number: $Re$	6,961,150
Pipe roughness: $\varepsilon$ , ft.	0.00015
Darcy friction factor: $f_D$	0.0154
Total length of pipe: ft.	74.549
Gas inlet pressure: $P_1$ , psia.	19.5367
Gas outlet pressure: $P_2$ , psia.	14.7
Pressure drop: $\Delta P$ , psi.	4.8367
Mach number at the inlet pipe: $M_1$	0.3906
Mach number at the outlet pipe: $M_2$	0.5191
Upstream gas velocity: $v_g$ , ft/sec.	659.332
Gas sonic velocity: $v_s$ , ft/sec.	1686.33
Gas critical pressure: $P_c$ , psia	8.7724
Fluid flow pattern	Subsonic

### 9.139 A RAPID SOLUTION FOR SIZING DEPRESSURING LINES [5C]

This is based on the technique developed by Lapple [116]. The methods employs a theoretical critical mass flow based on an ideal nozzle and adiabatic flow conditions and assumes a known upstream low velocity source pressure. The mass flux, where  $k = C_p/C_v = 1.0$ , can be determined by the following:

$$G_{ci} = 12.6P_1 \left( \frac{M_w}{ZT_1} \right)^{0.5} \quad (9-171)$$

In metric units:

$$G_{ci} = 6.7P_1 \left( \frac{M_w}{ZT_1} \right)^{0.5} \quad (9-172)$$

where

$G_{ci}$  = critical mass flux, lb/s ft<sup>2</sup>(kg/s m<sup>2</sup>)

$P_1$  = pressure at the upstream low velocity source  
(see Figure 9-111), psia, (kPa)

$M_w$  = molecular weight of the vapor

$T_1$  = upstream temperature, °R = 460 + °F, ( $K = 273.15 + °C$ )

$Z$  = compressibility factor.

The compressibility factor  $Z$  should be taken at flow conditions, and thus will change as the fluid moves down the line with resulting pressure drop. A stepwise calculation may be employed to allow for this variation. An accurate solution using this method is tedious. But sufficiently accurate results can be obtained by performing the calculation over relatively large increments of pipe lengths, using an average compressibility factor over those lengths. Regardless of which equation is used, actual mass flux ( $G$ ) is a function of critical mass flux ( $G_{ci}$ ), frictional resistance ( $N$ ), and the ratio of

downstream to upstream pressure. Figure 9-112 illustrates these relationships. Lapple [116] has developed similar charts for adiabatic cases with ratios of specific heats of 1.4 and 1.8. In the area below the diagonal line (Figure 9-112), the ratio  $G/G_{ci}$  remains constant, which indicates that sonic flow has been established. The total frictional resistance for use with the chart is expressed by [5c]:

$$N = f \frac{L}{D} + \sum K \quad (9-173)$$

where

$N$  = line resistance factor (dimensionless)

$F$  = Moody friction factor

$L$  = actual length of the line, ft (m)

$D$  = diameter of the line, ft (m)

$K$  = resistance coefficients of fittings (see Chapter 4).

If a Fanning friction factor is used,  $N = 4fL/D$ . These methods assume that there are no enlargements or contractions in the piping, and no variation in the Mach number that results from a change in area. Coulter [134] provides a more comprehensive treatment of ideal gas flow through sudden enlargements and contractions. Another method of calculating pressure drops for ideal gases at high velocities is the use of Fanno lines. Fanno lines are the loci of enthalpy/entropy conditions that result from adiabatic flow with friction in a pipe of constant cross section. Fanno lines extend into both supersonic and subsonic flow zones. However, for relief disposal systems, only the subsonic is of concern. The use of Fanno lines permits the calculation of pressure drops for ideal gases under adiabatic or isothermal flow conditions, with the total piping resistance as a parameter [135]. Generally, the velocity in gas discharging piping cannot exceed the sonic or critical velocity limit (this limit is shown on Lapple's charts [116] or Fanno lines).

In most disposal systems, the gases being handled are not ideal. For gases, deviations from the ideal are expressed as compressibility factors, which in turn are normally correlated with reduced pressure and reduced temperature. For hydrocarbon gases, the compressibility factor is less than 1.0 if the reduced temperature does not exceed 2.0 and the reduced pressure does not exceed about 6. Since most refinery pressure-relief valve disposal system fall within these limits, the compressibility of the gases will usually be less than 1.0. As long as compressibility is less than 1.0, the pressure drop calculated for an ideal gas will be larger than that calculated for the same gas incorporating the compressibility factor [5c]. The following steps are employed in the use of Figure 9-112.

1. Calculate the  $N$  (number of velocity heads) from Eq. (9-173)
2. Calculate  $P_3/P_1$  or  $P_2/P_1$

where

$P_1$  = pressure at upstream low velocity source, psia (kpa)

$P_2$  = pressure in the pipe at the exit or any point distance  $L$  downstream from the source, psia (kilopascals absolute)

$P_3$  = pressure in reservoir into which pipe discharges 14.7 psia (101 kpa) with atmospheric discharge

3. Calculate  $G_{ci}$  from Eq. (9-171) or Eq. (9-172)

4. From  $P_3/P_1$  or  $P_2/P_1$  and  $N$ , read  $G/G_{ci}$

5. Calculate  $G$  in pounds per second per square foot (kg/s m<sup>2</sup>)

6. Calculate  $W$  in actual flow in pounds per second (kg/s).

where

$W = G \times \text{cross-sectional area of pipe, ft}^2 (\text{m}^2)$ .

**EXAMPLE 9-29****Flare and Relief Blowdown System**

Size the flare manifold with relief loads and flow conditions shown in Figure 9-111.

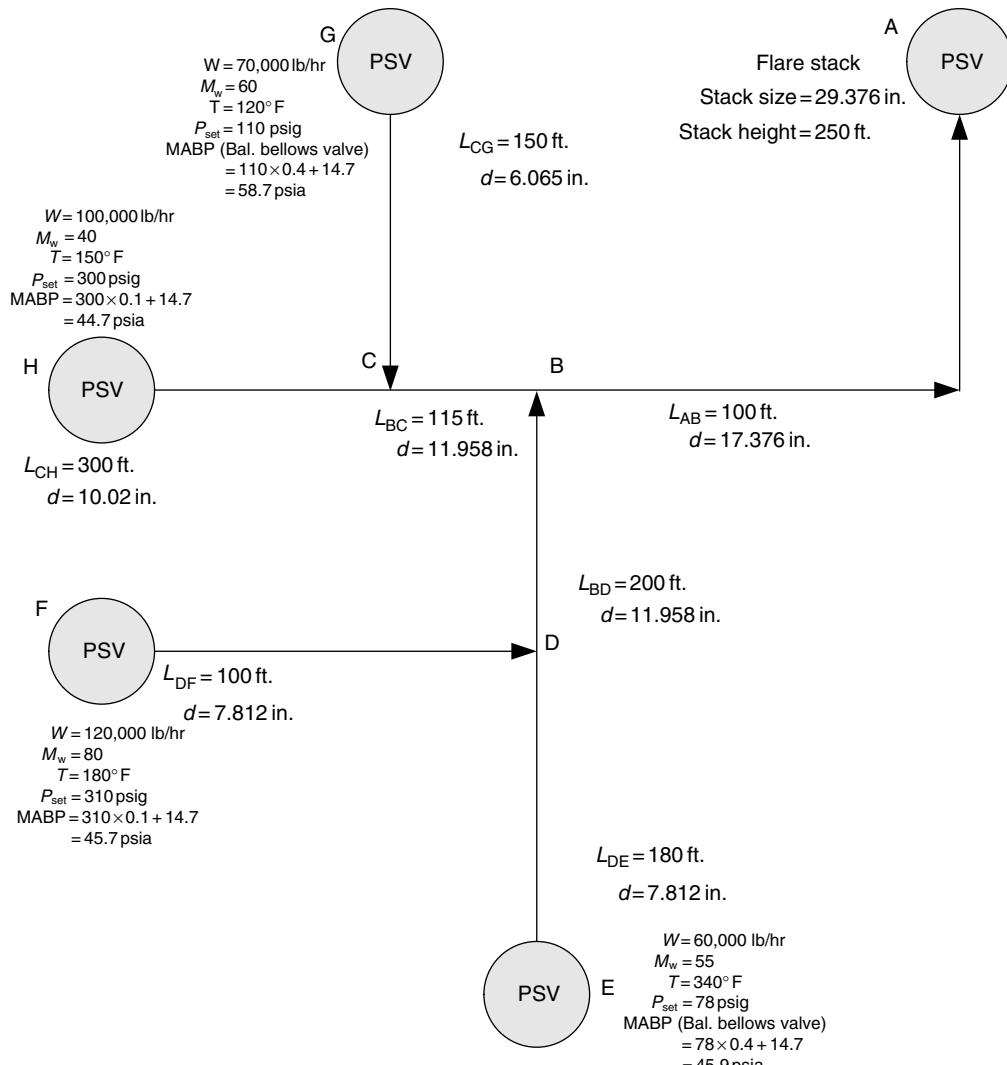


Figure 9-111 Flare and relief blowdown system

### 9.140 HAZARD AND OPERABILITY (HAZOP) STUDIES

The design and operation of a process plant form an integral part of safety, and systematic procedures should be employed to identify hazards and operability, and where necessary they should be quantified. During the design of a new plant, the hazard identification procedure is repeated at intervals. This is first carried out on the pilot plant before the full-scale version as the design progresses. Potential hazards whose significance can be assessed with the help of experiments are often revealed by this study.

The HAZOP study identifies a potential hazard. It provides little information on risk and consequences or its seriousness.

**Solution**

Table 9-44 summarizes the computer results using the computer program KAFLO. Note that the back pressures are close to, but less than the MABPs, showing that the line sizing is acceptable.

However, judgment is required, and sometimes the designer may decide that the consequences of the hazard are either trivial or unlikely to be ignored. In certain instances, the solution is obvious and the design is modified. A fault tree analysis (FTA) is useful where the consequences of the hazard are severe, or where its causes are many. The fault tree indicates how various events or combinations of events can give rise to a hazard. It can be employed to identify the most likely causes of the hazard, and thus to show where additional safety precautions will be most effective.

The HAZOP studies have formed an integral part of process safety management (PSM); several books and standards on this aspect have been published since its introduction to the CPI in

**TABLE 9-44 Computer Results of Vent Lines to the Flare Stack of Example 9-29**

Line	Stack	AB	BD	DE	DF	BC	CH	CG
Normal size: in.	30	18	12	8	8	12	10	6
Schedule number:	10	20	40	40	40	40	24	40
Internal diameter: d, in.	29.376	17.376	11.958	7.981	7.981	11.958	10.20	6.065
Flow rate: W, lb/hr.	350,000	350,000	180,000	60,000	120,000	170,000	100,000	70,000
Compressibility factor: Z	1	1	1	1	1	1	1	1
Gas density: $\rho$ , lb/ft <sup>3</sup>	0.122	0.249	0.347	0.263	0.487	0.262	0.250	0.445
Gas viscosity: $\mu$ , cP	0.0108	0.0108	0.0118	0.0130	0.0110	0.0099	0.010	0.0098
Ratio of specific heats: k ( $C_p/C_v$ )	1.27	1.27	1.27	1.27	1.27	1.27	1.27	1.27
Gas molecular weight: M <sub>w</sub> , lb/lb. mole	56.0	56.0	69.5.0	55.0	80.0	46.4	40.0	60.
Gas temperature: °F	186.6	186.6	233.3	340.0	180.0	137.6	150.0	120.0
Actual pipe length: L, ft $r = (P_1/P_2 = 1)$	250.0	1000.0	200.0	180.0	100.0	115.0	300.0	150.0
MAPB, psia	—	—	—	45.9	45.7	—	44.7	58.7
Equivalent length of pipe: $L_{eq}$ , ft	—	—	—	—	—	—	—	—
Gas Reynolds number: Re	6,961,150	11,768,570	8,049,359	3,649,051	8,625,030	9,061,175	6,297,405	7,431,389
Pipe roughness: $\epsilon$ , ft	0.00015	0.00015	0.00015	0.00015	0.00015	0.00015	0.00015	0.00015
Darcy friction factor: $f_D$	0.0113	0.0122	0.0131	0.0143	0.0142	0.0131	0.0136	0.0150
Total length of pipe: L, ft	250.0	1000.0	200.0	180.0	100.0	115.0	300.0	150.0
Gas inlet pressure: P <sub>1</sub> , psia	15.0632	30.8268	37.0825	40.9915	41.7929	36.1365	40.9063	46.0759
Gas outlet pressure: P <sub>2</sub> , psia	14.7	15.0632	34.1452	37.8690	37.8689	34.1452	36.1365	36.1365
Pressure drop: ΔP, psi	0.3632	15.7636	2.9373	3.1225	3.9239	1.9913	4.7698	9.9394
Mach number at the inlet pipe: M <sub>1</sub>	0.1989	0.2777	0.2330	0.1905	0.2771	0.2566	0.2067	0.2791
Mach number at the outlet pipe: M <sub>2</sub>	0.2038	0.5684	0.2531	0.2062	0.3058	0.2716	0.2340	0.3559
Upstream gas velocity: v <sub>g</sub> , ft/s	169.973	237.384	185.135	182.740	197.160	231.656	203.006	218.275
Gas sonic velocity: v <sub>s</sub> , ft/s	853.838	853.828	793.624	958.319	710.708	901.763	981.254	781.240
Gas critical pressure: P <sub>c</sub> , psia	3.3799	9.6546	9.7446	8.8052	13.0602	10.4573	9.5332	14.5016
Fluid flow pattern	Subsonic	Subsonic	Subsonic	Subsonic	Subsonic	Subsonic	Subsonic	Subsonic

the 1960s [136–140]. Over the years, practitioners have gained extensive experience with this methodology and in the US, 29 CFR 1910 is mandatory. A HAZOP study is a structured review of the design of a plant and operating procedures. The main aims are the following.

- identify potential maloperations
- assess their consequences
- recommend corrective actions.

The recommended actions may eliminate a potential cause or interrupt the consequences. They involve the following.

- pipework or other hardware changes
- changes in operating conditions
- more precise operating instructions
- addition of alarms with prescribed operator responses
- addition of automatic trip systems.

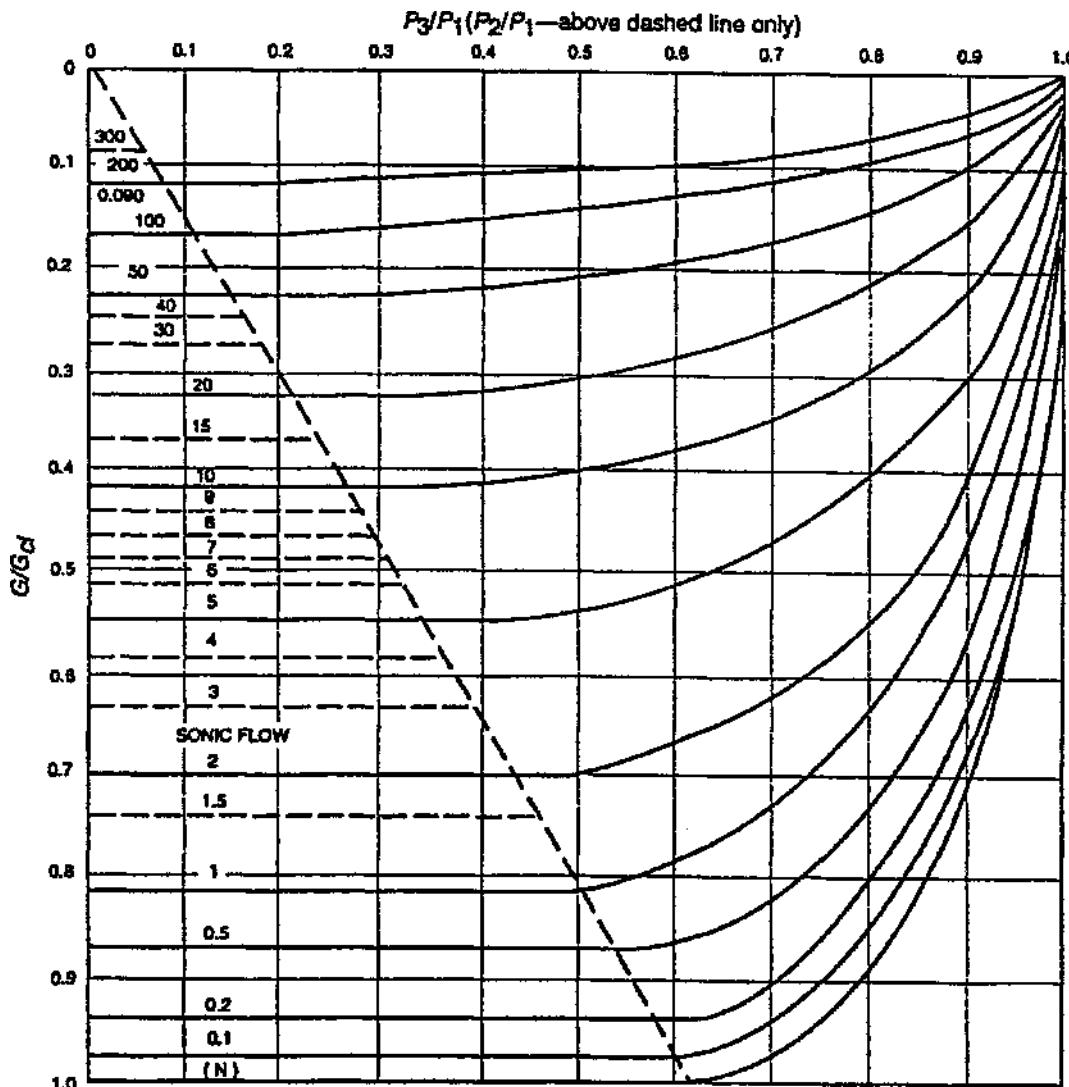
The procedure for a HAZOP study is to apply a number of guidewords to various sections of the process design intention. The design intention informs what the process is expected to carry out. Table 9-45 shows these guidewords, and Figure 9-113 summarizes the whole procedure. Some companies have developed their own set of guidewords for particular technologies; however, while clear recommendations can be made as to which guidewords should be considered, it is not possible to provide such firm advice regarding parameters. The selection of parameters is a task each team must

address for each system studied. Table 9-46 gives examples of parameters that might be used in the analysis of a process operation. The list is intended to show the depth and breadth of the parameter and guideword search that can be used. Further, the extent of this list emphasizes the need for the HAZOP team to form a clear conceptual model of the step and to use it to decide which parameters should be used in the search for possible deviations. When deviations are being sought, it must be remembered that not every guideword combines with a parameter to give a meaningful deviation. Therefore, it is futile and a waste of time to discuss combinations which do not have a physical meaning. Table 9-47 shows examples of meaningful combinations. In general, HAZOP study is most effective when it is a creative process and the use of checklists for guidewords or parameters can stultify creativity. Nonetheless, checklists can be helpful for an experienced team.

#### 9.141 STUDY CO-ORDINATION

The HAZOP is a structured review exercise carried out by a team of between three and six people, one of who acts as a chairman. Another member of the team acts as a secretary and records the results of the proceeding. A typical team comprises of the following.

- Chairman: study leader, experienced in HAZOP.
- Project or design engineer: usually a mechanical engineer, responsible for keeping the costs within budget.



**Figure 9-112** Adiabatic Flow of  $k = 1.0$ . Compressible Fluids Through Pipes at High Pressure Drop. (From API, recommended Practice 521, 4th ed. Mar 1997 [5c].)

- Process engineer: usually the chemical engineer who drew up the flowsheet.
- Commissioning manager: usually a chemical engineer, who will have to start up and operate the plant.
- Control system design engineer: modern plants contain sophisticated control and trip systems and HAZOPS often result in the addition of yet more instrumentation.
- Research chemist: if new chemistry is involved.
- Independent team leader: an expert in the HAZOP technique, not the plant, whose job is to ensure that the team follows the procedure.

The team should have a wide range of knowledge and experience. If a contractor designs the plant, then the HAZOP team should include people from both the contractor and the client organizations. On a computer-controlled batch plant, the software engineer should be a member of the HAZOP team, which should include at least one other person who understands the computer logic.

In an existing plant, the team should include several people with experience of the plant. A typical team is as follows.

- Plant manager: responsible for plant operation.
- Process foreman: knows what actually happens rather than what is supposed to happen.
- Plant engineer: responsible for mechanical maintenance such as testing of alarms and trips, as well as installation of new instruments.
- Process investigation manager: responsible for investigating technical problems and for transferring laboratory results to plant-scale operations.
- Independent team leader.

## 9.142 HAZOP OF A BATCH PROCESS

A HAZOP study of a batch processing plant involves a review of the process as a series of discrete stages, as the conditions in a single reactor change with time in a given cycle. Variations in

**TABLE 9-45 List of Guidewords used for the HAZOP Procedure**

Guideword	Meaning	Explanation
No (not, none)	Complete negation of the design intention.	Application to flow, concentration, react, heat transfer, separate and similar functions. No level means an empty vessel or a two-phase interface is lost.
More (more of, higher)	Quantitative increase in a parameter.	Applicable to all property words.
Less (less of, lower)	Quantitative decrease in a parameter.	Applicable to all property words.
As well as (more than)	Design intention achieved together with something else.	Flow as well as describes contamination in a pipeline. React as well as covers side reactions.
Part of	Design intentions only partly achieved.	It is more precise to use the guideword less wherever possible. But fluctuations in a property word are covered by part of.
Reverse	Logical opposite of the design intention occurs.	Applicable principally to flow.
Other than (other)	Complete substitution – another activity takes place.	Applicable where the wrong material flows in a line or the wrong reagent is charged to a reactor or the required reaction does not occur but others do.

Other useful guidewords include the following:

Guideword	Meaning
Where else	Applicable for flows, transfers, sources, and destinations.
Before/after	The step (or some part of it) is effected out of sequence.
Early/late	The timing is different from the intention.
Faster/slower	The step is done/not done with the right timing.

the rate of change as well as changes in the duration of settling, mixing, and reacting are important deviations. A HAZOP on a batch process includes the flowrates of service fluids: it is more common to look at the reactants in terms of amounts charged or discharged rather than flowrates.

Sequential operating instructions are applied to identify the design intent through various stages of the process. The HAZOP guidewords can then be applied to each design intent instead of the process line. For example, raise concentration or temperature to a specified value for a given time. The HAZOP would then consider the consequences of not achieving or exceeding the desired concentration or temperature in the specified time. A further example – if an instruction states that 1 tonne of A has to be charged to a reactor, then the team should consider deviations such as

DON'T CHARGE	A
CHARGE MORE	A
CHARGE LESS	A
CHARGE AS WELL AS	A
CHARGE PART OF	A ( <i>if A is a mixture</i> )
CHARGE OTHER THAN	A

**REVERSE CHARGE A** (i.e., can flow occur from the reactor to the A container?). This can be the most serious deviation if

A IS ADDED EARLY	
A IS ADDED LATE	
A IS ADDED TOO QUICKLY	
A IS ADDED TOO SLOWLY	

Many accidents have occurred because process materials flowed in the wrong direction to that expected. For example, ethylene oxide and ammonia were reacted to make ethanolamine. Some ammonia flowed from the reactor in the opposite direction, along the ethylene

oxide transfer line into the ethylene oxide tank, past several non-return valves and a positive displacement pump. It got past the pump through the relief valve, which discharged into the pump suction line. The ammonia reacted with 30 m<sup>3</sup> of ethylene oxide in the tank, which ruptured violently. The released ethylene oxide vapor exploded causing damage and destruction over a wide area. A HAZOP study might have disclosed the facts that reverse flow could occur [97].

### 9.143 LIMITATIONS OF HAZOP STUDIES

While HAZOP can readily identify the majority of operational problems, it is still fallible to what it can miss. The HAZOP-Guide to best practice [137] states “A HAZOP study is not an infallible method of identifying every possible hazard or operability problem that could arise during the actual operations. Expertise and experience within the team is crucial to the quality and completeness of a study. The accuracy and extend of the information available to the team, the scope of the study and the manner of the study all influence its success.” Krishnan [141] has illustrated the limitations of HAZOP studies as exemplified in Table 9-48. The table serves as an illustration to gain insight with regard to accidents and their predictability by HAZOP. Table 9-49 shows pertinent points that should be kept in mind while performing a thorough HAZOP study.

### 9.144 HAZARD ANALYSIS (HAZAN)

The HAZAN is a quantitative way of assessing the likelihood of failure. Other names associated with this technique are risk analysis, quantitative risk assessment (QRA), and probability risk assessment (PRA). Kletz [139] expressed the view that HAZAN is a selective technique whilst HAZOP can be readily applied to new design and major modification. Some limitations of HAZOP are its inability to detect every weakness in design such as weaknesses in plant

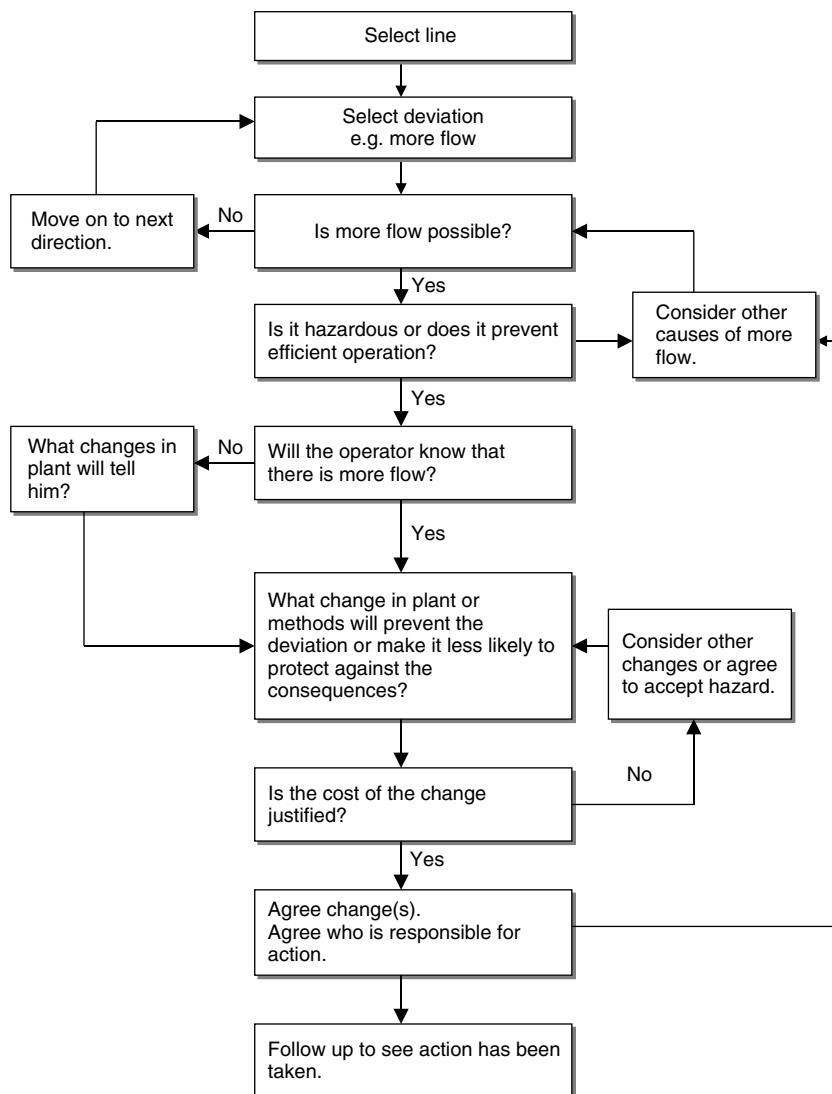


Figure 9-113 HAZOP procedure. (From Trevor kletz: HAZOP and HAZAN, 4th ed., IChemE, 1999.)

**TABLE 9-46 Examples of Possible Parameters for Process Operations**

<ul style="list-style-type: none"> <li>• Addition</li> <li>• Composition</li> <li>• Level</li> <li>• Flow or amount</li> <li>• Mixing</li> <li>• Transfer</li> <li>• Temperature</li> <li>• Viscosity</li> <li>• Measure</li> <li>• Stirring</li> <li>• Pressure</li> <li>• Separation</li> <li>• Communication</li> </ul>	<ul style="list-style-type: none"> <li>• Phase</li> <li>• Speed</li> <li>• Particle size</li> <li>• Reaction</li> <li>• Control</li> <li>• pH</li> <li>• Sequence</li> <li>• Signal</li> <li>• Start/stop</li> <li>• Operate</li> <li>• Maintain</li> <li>• Services</li> <li>• Time</li> </ul>
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(Source [145], By permission from IChemE, All rights reserved.)

**TABLE 9-47 Examples of Meaningful Combinations of Parameters and Guidewords**

Parameter	Guidewords That Can Give Meaningful Combinations
Flow	None; more of; less of; reverse; elsewhere; as well as
Temperature	Higher; lower
Pressure	Higher; lower; reverse
Level	Higher; lower; none
Mixing	Less; more; none
Reaction	Higher (rate of); lower (rate of); none; reverse; as well as/other than; part of
Phase	Other; reverse; as well as
Composition	Part of; as well as
Communication	None; part of; more of; less of; other; as well as

(Source [145] By permission from IChemE, All rights reserved.)

**TABLE 9-48 Summary of Recent Major Industrial Accidents That Are Sourced to Process Safety Failure**

Incident	Causes	HAZOP Identification of the Cause
Flixborough, UK (1974) Cyclohexane vapor cloud explosion	The cause was determined as the failure of a bypass line connecting two reactors handling high quantities of cyclohexane. The bypass line was installed in a hurry without proper engineering or safety review.	HAZOP would not have been able to predict the actions of the plant staff.
Bhopal, India (1984) Release of poisonous MIC gas (Methyl isocyanate)	The multiple causes that led to this accident included storage of a highly poisonous chemical MIC in a facility that was shutdown and idle, failure of the scrubbing and flare system to absorb the toxic vapor, and lack of knowledge on the nature of toxicity of the chemical itself.	A well-done HAZOP would have predicted the hazards due to the non-operability of the relief system. But it would not have predicted the nature of the risk and catastrophe in case of a loss of containment unless all details of the chemical was known. (This raises a question as to how much we know about the long-term and short-term effects of chemicals).
Seveso, Italy (1976) Release of poisonous gas TCDD	A runaway reaction caused the chemical TCDD (2,3,7,8 tetrachlorodibenzo-paradioxin—one of the most potent toxins known to man) to be released through the relief system in a white cloud over the town of Seveso. A heavy rain washed the TCDD into the soil. Lack of knowledge and poor communication with public delayed response from authorities.	HAZOP would have predicted the consequences of runaway reaction and release of the vapor. But the lack of knowledge about the chemical itself and its consequences would not have been predictable by HAZOP.
Three Mile Island nuclear plant (1979) Equipment malfunction and shutdown resulting in partial meltdown of reactor core	In the Three Mile Island nuclear plant, shutdown of main feed water pumps caused chain shutdown of steam generators and the reactor. Pressure build-up in the reactor system occurred and the relief valve on system opened to relieve the pressure, but failed to seat back. Result was loss of containment of coolant through the relief valve. But somehow there was no direct way the operators could know that the level of coolant was dangerously low in the reactor and that the reactor was overheating.	In Three Mile Island, there were alarms in the control room but the operator did not know that the relief valve was stuck open and the coolant level was getting low. Alarms in the control room resulted in confused initial actions by the operating staff that actually worsened the situation. Though the reactor core suffered a partial meltdown, worst case scenarios were avoided. A HAZOP would have predicted lack of critical alarms but would not have predicted the actions by the operating staff. Reports indicate that there were, in fact, too many alarms (nearly 100!) in the control room.
Piper Alpha Offshore platform (1988) Leaked condensate caught fire resulting in massive fire and explosion and loss of the platform	It is believed that the leak came from pipe-work connected to a condensate pump. A safety valve had been removed from the pipe-work for overhaul and maintenance. The pump itself was undergoing maintenance work. When the pipe-work from which the safety valve was removed was pressurized at startup, it is believed that the leak occurred.	HAZOP would have predicted and corrected a lot of items that were found lacking in the platform during the subsequent inquiry including unit spacing and locations, safety provisions, etc., but would not have predicted the causes of the incident and further the reason for escalation of the incident to a catastrophe.

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layout, or miss hazards due to leaks on lines that pass through or close to a unit but carry material that is not used on that unit. However, hazards should generally be avoided by changing the design. Assessing hazard by HAZAN or any other technique should always be the alternative choice.

A small team similar to that used in HAZOP carries out HAZAN. The five steps in HAZAN are:

- Estimate how frequently the incident will occur.
- Estimate the consequences to employees, members of the public.
- Estimate the plant and profits.
- Compare the results of the first two steps with a target or criterion.
- Decide whether it is necessary to act to reduce the incident's frequency or its severity.

## 9.145 FAULT TREE ANALYSIS

Fault tree analysis (FTA) is used to assess the frequency of an incident. A fault tree is a diagram that shows how primary causes produce events, which can contribute to a particular hazard. There are several pathways in which a single primary cause can combine with other primary causes or events. Therefore a single cause may be found in more than one hazard, and may occur at different locations in the fault tree.

The graphical structure of the fault tree enables the primary causes, and secondary events are combined to produce the hazards. We can compare the relative contributions of the different events to the probability of the hazardous outcome by employing the probability of occurrence of causes and events on the fault tree.

**TABLE 9-49 Checklist for a Productive HAZOP Study**

1. HAZOP is not an infallible method. It can achieve many items, but users must be aware of study's limitations.
2. An overall Process Safety management (PSM) system should be in place and HAZOP should be a part of the same. For complex systems, always conduct another study augmenting the HAZOP study including Quantitative Risk Analysis (QRA), FMEA and/or event tree analysis.
3. HAZOP is not a substitute for design. The process design should be sound with regard to codes, standards and good engineering practice. Also, the design stage should not be rushed with the attitude that the HAZOP team will catch any errors or omissions at a later time. This is the biggest judgment lapse an engineering team can make.
4. Composition of the HAZOP team is critical. The best individuals in a positive group dynamics environment are necessary to produce a high-quality HAZOP report.
5. Management must support and act on the HAZOP recommendations. Many recommendations made with much effort and time are forgotten due to lack of follow-up action and no support from management.
6. Low-probability major-consequence hazards such as total power failure, utilities shutdown, location suitability, high loss of containment, natural disasters, etc. should be reviewed separately in a major hazards review.
7. Identify safety critical equipment and control systems, analyze them carefully and provide redundancy efforts if required.
8. Always consider facility/unit startup and shutdown cases.
9. Operator response in an abnormal or emergency situation should be identified and later tested and verified.
10. Understand double jeopardy situations correctly. Double jeopardy definition only includes situations occurring *at the same instant*. Remember: More than one situation can develop over a period of time although not at the same instant as the first event.
11. When a HAZOP study is done on modifications to an existing plant, thoroughly review the history of the facility. Many changes must have occurred. Without understanding the design intentions of each modification, the HAZOP study will not be meaningful. Proper management of change procedures are a must.
12. Adding more hardware to the system is not always the best solution to a problem. When such a situation arises, risk-based assessment would be helpful.
13. Analyze perceived low risks for escalation potential to a major accident.
14. Thoroughly review the "incident database" for similar plant or equipment. Such reviews enable focusing on other potential areas.
15. Remember: A HAZOP cannot predict a deliberate act of sabotage.

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## 9.146 INHERENTLY SAFER PLANT DESIGN

Hazards should be considered and if possible eliminated at the design stage or in the process development where necessary. This involves considering alternative processes, reduction or elimination of hazardous chemicals, site selection, or spacing of process units. It is essential to consider inherently safer principles in the design stage, because designers may have various constraints imposed upon them by the time the process is developed. The safety (i.e., safety, health, environmental and loss prevention) performance of any plant is a function of many factors but can be considered to be primarily dependent on the following aspects [142]:

- The qualities of the people who design, operate, and maintain it.
- The effectiveness of the management and management systems in design, operation, maintenance, and incident response.
- The effectiveness of engineered safety systems and control hazards.
- The risk potential of the plant and the process being carried out.

A given level of safety performance can be attained in different ways by allocating different standards to each of these aspects. For example, use of safety systems with good personnel and management systems can result in a high hazard plant being made "tolerably" safe. While a company may place an emphasis on its management systems, another may rely more on engineered safeguards. The exercise becomes a balancing act between the risk potential of the plant, the risk reduction afforded by the various safety control and management efforts, and the cost of these efforts. In reducing or eliminating the hazard potential of the plant by careful selection of the process and good engineering design of the plant, the need for "add on" safety systems and detailed management controls is reduced. The plant is "inherently safer"

because its safety performance is less reliant on "add on" engineered systems and management controls which can and do fail. Inherently safer approach has the advantage of providing a means to address safety, health, environmental and loss prevention issues in a strategic and integrated manner by dealing with the hazards at source, rather than trying to find ways to live with them. Minimizing the inherent hazards of the plant offers savings by reducing the need for expensive safety systems and instrumentation, easing the burden on personnel and procedures, and simplifying on-site and off-site emergency plans. Englund [143] has given details of inherently safer plants of various equipment in the CPI. He emphasized the importance of user-friendly plants as originally expatiated by Kletz [144]. This strategy is based on a hierarchy of four approaches to process plant safety:

- 1. Intensification** Reducing the hazardous materials.
- 2. Substitution** Substituting the hazardous materials with less hazardous ones. For example, if the hazardous material is an intermediate product, alternate chemical reaction pathways might be used.
- 3. Attenuation** Using the hazardous materials or processes in a way that limits their hazard potential by lowering the temperature/pressure or adding stabilizing additives.
- 4. Simplification** Making the plant and process simpler to design, build, and operate, hence less prone to equipment, control, and human failings.

The goal of these approaches is summarized by the notion of a "friendly plant" as shown in Tables 9-50 which provides a means to visualize the "inherently safer" concept [142]. This concept needs to be integrated into the overall project and addressed together with other constraints and objectives. A framework is required, which enables the basis of the project, process, or plant to be challenged in

**TABLE 9-50 The Friendly Plant Concept (A Risk-based Approach)**

Consequence reduction	
Friendly plant	Classical Inherently Safer Plant Less Hazardous: <ul style="list-style-type: none"><li>• processes</li><li>• materials</li><li>• conditions</li></ul> and lower inventories Needs Extra: <ul style="list-style-type: none"><li>• hardware</li><li>• control systems</li><li>• engineered safeguards</li></ul> ... to control hazards, leading to a complex (unfriendly) plant
Tolerant of People or Equipment Failings	
Alternative-Unfriendly plant "High Consequence" Plant	
Frequency reduction	
Friendly plant Less Prone to People or Equipment Failings	Simpler Plant Simpler to: <ul style="list-style-type: none"><li>• design</li><li>• build</li><li>• operate</li><li>• maintain</li></ul>
Alternative-Unfriendly plant Prone to Failings	Complex Plant Needs Extra: <ul style="list-style-type: none"><li>• hardware</li><li>• control systems</li><li>• engineered safeguards</li></ul> ... to prevent or control failings, leading to a complex (unfriendly) plant, and extra: <ul style="list-style-type: none"><li>• manpower</li><li>• training</li><li>• procedures</li><li>• management controls</li></ul> ... to make plant operable

(Source: Mansfield, D. and K. Cassidy [140], By permission from IChemE, All rights reserved.)

a systematic way at each of the key steps in the project's lifecycle. This framework considers alternatives and thus allows them to be determined in terms of their inherent safety, feasibility, and cost. Table 9-51 shows the essential steps to be considered when starting from initial specification to process detailed design.

### INHERENTLY SAFER PLANT DESIGN IN REACTOR SYSTEMS

Here, we shall review the practical application to reactor systems. The following should be considered in applying inherently safer plant involving reactor systems.

1. A good understanding of reaction kinetics is required to establish safe conditions for operation of exothermic reactions.

2. Use continuous reactors if possible. It is usually easier to control continuous reactors than batch reactors. If a batch reaction system is required, minimize the amount of unreacted hazardous materials in the reactor. Figures 9-114 and 9-115 show typical examples.

Methods have been developed for improving batch process productivity in the manufacture of styrene–butadiene latex by the continuous addition of reactants so the reaction takes place as the reactor is being filled. These are not continuous processes even though the reactants are added continuously during most of a batch cycle. The net result is that reactants can be added about as fast as heat can be removed. There is relatively little hazardous material in the reactor at any time because the reactants, which are flammable or combustible, are converted to non-hazardous and non-volatile polymer almost as fast as they are added.

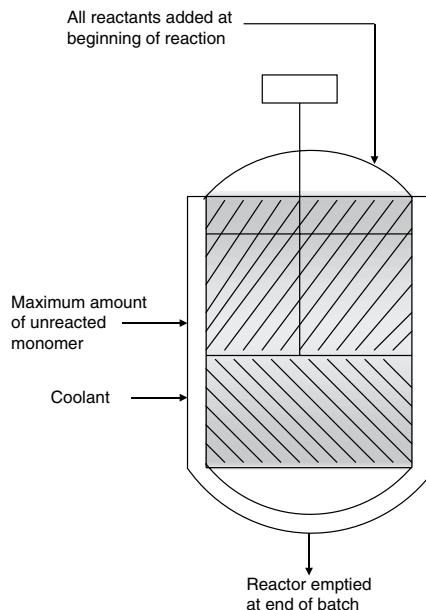
3. If possible, produce and consume hazardous raw materials in situ. Some process raw materials are so hazardous to ship and store that it is very desirable to minimize the amount of these materials on hand. Sometimes it is possible to achieve this by using less hazardous chemicals, so there is only a small amount of the hazardous material in the reactor at any time.
4. Liquid-phase of solid-phase reactors contains more material than vapor-phase reactors and thus contains more stored energy than vapor-phase reactors.
5. Using high purity raw materials and products can reduce the amount the waste material that must be handled.
6. Consider designing the reactor for the highest pressure that could be expected in case of a runaway reaction to reduce the possibility of release of material to the environment. For example, for a certain process for the manufacture of polystyrene, the composition in the reactor included monomer, polymer, and solvent. The maximum pressure that could be reached by adiabatic polymerization of the mixture in the reactor, beginning at the reaction temperature of 120°C (248°F), was about 300 psig (2068 kPa (gauge)). With this knowledge, it was possible to design polymerization equipment that will withstand this pressure, plus a reasonable safety factor, with considerable confidence that a runaway will not cause a release of material through a pressure relief system or because of an equipment rupture.
7. Limit the total charge possible to a batch reactor by using a pre-charge or feed tank of limited capacity. Alternatively, limit the rate of addition by selection of a pump with a maximum capacity lower than the safe maximum rate of addition for the process, or by using restriction orifices.
8. The maximum or minimum temperature attainable in a vessel can be limited by properly designed jacket heating systems. If steam heating is used, maximum temperatures can be limited by controlling steam pressure. A steam desuperheater may be needed to avoid excessive temperature of superheated steam from a pressure let down station.
9. Tubular reactors often offer the greatest potential for inventory reduction. They are usually simple, have no moving parts, and a minimum number of joints and connections that can leak.
10. Mass transfer is often the rate-limiting step in gas–liquid reactions. Novel reactor designs that increase mass transfer can reduce reactor size and may also improve process yields.

The dangerous substances and explosive atmosphere regulations act (DSEAR) is the UK implementation of the European Directive 1999/92/EC *Minimum requirements for improving the safety and health protection of workers potentially at risk from explosive atmospheres, also known as the ATEX 137 directive*. This directive sets minimum requirements aimed at protecting employees, contractors,

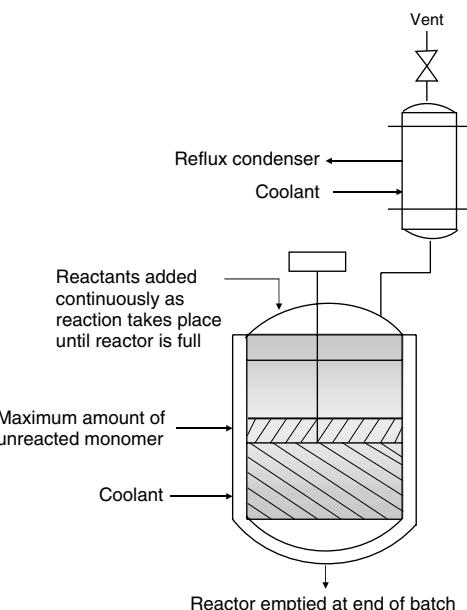
**TABLE 9-51** The Steps to Plant Design

Decision Point	Key Questions/Decisions	Information Used
Initial Specification	What Product What Throughput	Market Research R&D New Product
Process Synthesis Route	How to make the product What route What reactions, materials starting point	R&D Chemists research Known synthesis routes and techniques
Chemical Flowsheet	Flowrates, Conversion Factors, and Basic Unit Operation Selection Temperatures, Pressures solvents and catalyst selection	Process synthesis route Lab and pilot scale trials Knowledge of existing processes
Process Flowsheet	Batch vs Continuous operation Unit operation selection Control/operation philosophy	Information above plus Process engineering design principles and experience
Process Conceptual Design	Equipment selection and sizing Inventory of process Single vs Multiple Trains Utility requirements Overdesign/Flexibility Recycles and Buffer capacities Instrumentation and Control Location/Siting of plant Preliminary plant layout Materials of construction	As above plus equipment suppliers data, raw materials data, Company design procedures and requirements
Process Detailed Design	Detailed specification based on concept design	Process conceptual design and codes/standards and procedures on past projects/designs

(Source: [140], By permission from IChemE, All rights reserved.)



Batch reaction with all reactants added at beginning of reaction. Considerable amount of flammable and hazardous material in reactor at beginning.



Batch reaction with reactants added during reaction. Little flammable and hazardous material present at any time. Reflux (or knockback) condenser used to provide additional heat transfer.

**Figure 9-114** Process A. (Source: S.M. Englund, "Inherently Safer Plants: Practical Applications", *Process Safety Prog.*, Vol. 14, No. 1, pp. 63–70, AIChE, 1995.)

**Figure 9-115** Process B. (Source: S.M. Englund, "Inherently Safer Plants: Practical Applications", *Process Safety Prog.*, Vol. 14, No. 1, pp. 63–70, AIChE, 1995.)

visitors, and the public from hazards posed by fire and explosions. The DSEAR is concerned with the storage, handling, and use of dangerous substances which are defined as substances and preparations classified under the Chemicals Hazards Information and Packaging for Supply (CHIP) regulations. Applying inherent safety principles early in the development of the process can be a useful technique in achieving compliance with DSEAR regulations act. Bell [145] has illustrated how compliance with DSEAR will embed a more inherently safe design basis within the CPI.

## GLOSSARY

**Accumulation:** The buildup of unreacted reagent or intermediates, usually associated with reactant added during a semi-batch operations.

**Activation energy  $E_a$ :** The constant  $E_a$  in the exponential part of the Arrhenius equation, associated with the minimum energy difference between the reactants and an activated complex (transition state which has a structure intermediate to those of the reactants and the products), or with the minimum collision energy between molecules that is required to enable a reaction to occur.

**Adiabatic:** A system condition in which no heat is exchanged between the chemical system and its environment.

**Adiabatic induction time:** Induction period or time to an event (spontaneous ignition, explosion, etc.) under adiabatic conditions, starting at operating conditions.

**Adiabatic temperature rise:** Maximum increase in temperature that can be achieved. This increase occurs when the substance or reaction mixture decomposes or reacts completely under adiabatic conditions. The adiabatic temperature rise follows from:

$$\Delta T_{\text{adia}} = x_o (\Delta H_{\text{RX}}) / C\phi$$

where  $x_o$  = Initial mass fraction,  $\Delta H_{\text{RX}}$  = Heat of reaction J/kg,  $C$  = Liquid heat capacity J/kg K,  $\phi$  = Dimensionless thermal inertial factor (Phi-factor).

**Autocatalytic reaction:** A reaction, the rate of which is increased by the catalyzing effect of its reaction products.

**Autoignition temperature:** The AIT of a substance, whether solid, liquid, or gaseous, is the minimum temperature required to initiate or cause self-sustained combustion (e.g., in air, chlorine, or other oxidant) with no other source of ignition. For example, if a gas is flammable and its AIT is exceeded, there is an explosion.

**Back pressure:** The static pressure existing at the outlet of a pressure-relief device as a result of the pressure in the discharge system. It is the sum of the superimposed and built-up back pressure.

**Blowdown:** The difference between actual popping pressure of a pressure-relief valve and actual re-seating pressure expressed as a percentage of set pressure.

**Blowdown pressure:** The value of decreasing inlet static pressure at which no further discharge is detected at the outlet of a safety relief valve of the resilient disk type after the valve has been subjected to a pressure equal to or above the popping pressure.

**Boiling Liquid Expanding Vapor Explosive (BLEVE):** The violent rupture of a pressure vessel containing saturated liquid/vapor at a temperature well above its atmospheric boiling point. The sudden decrease in pressure results in explosive vaporization of a fraction of the liquid and a cloud of vapor and mist, with accompanying blast effects. The resulting flash vaporization of a large fraction of the liquid produces a large cloud. If the vapor is flammable and if an ignition source is present at the time of vessel rupture, the vapor cloud burns in the form of a large rising fireball.

**Built-up back pressure:** Pressure existing at the outlet of a pressure-relief device caused by flow through that particular device into a discharge system.

**Burst pressure:** The value of inlet static pressure at which a rupture disk device functions.

**Chatter:** Abnormal rapid reciprocating motion of the movable parts of a pressure-relief valve in which the disk contacts the seat.

**Combustible:** A term used to classify certain liquids that will burn on the basis of flash points. Both the NFPA and the Department of Transportation (DOT) define "combustible liquids" as having a flashing point of 100°F (37.8°C) or lower.

**Combustible Dusts:** Dusts are particularly hazardous; they have a very high surface area-to-volume ratio. When finely divided as powders or dusts, solids burn quite differently from the original material in the bulk. Many combustible dusts produced by industrial processes are explosive when they are suspended as a cloud in air. A spark may be sufficient to ignite them. After ignition, flame spreads rapidly through the dust cloud as successive layers are heated to ignition temperature.

**Condensed phase explosion:** An explosion that occurs when the fuel is present in the form of a liquid or solid.

**Confined explosion:** An explosion of a fuel–oxidant mixture inside a closed system (e.g. A vessel or building).

**Confined Vapor Cloud Explosion (CVCE):** A condensed phase explosion occurring in confinement (equipment, building, and/or congested surroundings). Explosions in vessels and pipes, processing or storing reactive chemicals at elevated conditions are examples of CVCE. The excessive buildup of pressure in the confinement leads to this type of explosions leading to high over-pressure, shock waves, and heat load (if the chemical is flammable and ignites). The fragments of exploded vessels and other objects hit by blast waves become airborne and act as missiles.

**Containment:** A physical system in which under all conditions no reactants or products are exchanged between the system and its environment.

**Cubic law:** The correlation of the vessel volume with the maximum rate of pressure rise.  $V^{1/3} (dP/dt)_{\text{max}} = \text{constant} = K_{\text{max}}$

**Decomposition energy:** The maximum amount of energy which can be released upon decomposition. The product of decomposition energy and total mass is an important parameter for determining the effects of a sudden energy release, for example in an explosion.

**Deflagration:** The chemical reaction of a substance in which the reaction front advances into the unreacted substance at less than sonic velocity. Where a blast wave is produced that has the potential to cause damage, the term "explosive deflagration" is used.

**Detonation:** A release of energy caused by the extremely rapid chemical reaction of a substance in which the reaction front advances into the unreacted substance at equal to or greater than sonic velocity.

**Design Institute for Emergency Relief Systems (DIERS):** Institute under the auspices of the AIChE funded to investigate design requirements for vent lines in the case of two-phase venting.

**Disk:** The pressure-containing movable element of a pressure-relief valve which effects closure.

**Dow Fire and Explosion Index (F&EI):** A method (developed by Dow Chemical Company) for ranking the relative fire and explosion risk associated with a process. Analysts calculate various hazard and explosion indexes using material characteristics and process data.

**Dust:** Solid mixture with a maximum particle size of 500 μm.

**Dust explosion class, St:** Dusts are classified in accordance with the  $K_{\text{max}}$  values.

**Dust explosion constant ( $K_{\text{st}}$ ):**  $K_{\text{st}}$  is defined as the maximum rate of pressure rise during a dust explosion in an equidimensional vessel, times the cube root of the vessel volume, that

is  $K_{st} = (dP/dt)_{max} V^{1/3}$ .  $K_{st}$  (bar m/s) is numerically equal to the maximum rate of pressure rise (bar/s) in the 1 m<sup>3</sup> standard ISO (International Standards Organization, 1985) test. The St class was determined using the modified Hartmann tube with a hinged lid at the top. St 1 dust means  $K_{st} \leq 200$  bar m/s, St 2 dust means that  $200$  bar m/s  $\leq K_{st} < 300$  bar m/s, and St 3 dust means that  $K_{st} \geq 300$  bar m/s.

**Exotherm:** A reaction is called exothermic if energy is released during the reaction.

**Explosion:** Propagation of a flame in a premixture of combustible gases, suspended dust(s), combustible vapor(s), mist(s), or mixtures of thereof, in a gaseous oxidant such as air, in a closed, or substantially closed vessel.

**Explosion rupture disk device:** A rupture disk device designed for use at high rates of pressure rise.

**Fail-safe:** Design features which provide for the maintenance of safe operating conditions in the event of a malfunction of control devices or an interruption of an energy source (e.g., direction of failure of a motor-operated valve on loss of motive power).

**Failure:** An unacceptable difference between expected and observed performance.

**Fire point:** The temperature at which a material continues to burn when the ignition source is removed.

**Flammability limits:** The range of gas or vapor compositions in air that will burn or explode if a flame or other ignition source is present. *Note:* The range represents an unsafe gas or vapor mixture with air that may ignite or explode. Generally, the wider the range the greater the fire potential.

**Flammable:** A "flammable liquid" is defined as a liquid with a flash point below 100° F (37.8° C). Flammable liquids provide ignitable vapor at room temperatures and must be handled with caution. Flammable liquids are Class I liquids and are subdivided as follows:

Class 1A: Those having flash points below 73° F and having a boiling point below 100° F.

Class 1B: Those having flash points below 73° F and having a boiling point at or above 100° F.

**Flares:** Flares are used to burn the combustible or toxic gas to produce combustion products, which are neither toxic nor combustible. The diameter of a flare must be suitable to maintain a stable flame and prevent a blowdown (when vapor velocities are greater than 20% of the sonic velocity).

**Flash fire:** The combustion of a flammable vapor and air mixture in which flame passes through that mixture at less than sonic velocity, such that negligible damaging overpressure is generated.

**Flash point:** The lowest temperature at which vapors above a liquid will ignite. The temperature at which vapor will burn while in contact with an ignition source, but which will not continue to burn after the ignition source is removed.

**Gases:** Flammable gases are usually very easily ignited if mixed with air. Flammable gases are often stored under pressure, in some cases as a liquid. Even small leaks of a liquefied flammable gas can form relatively large quantities of gas, which is ready for combustion.

**Gassy system:** In gassy systems, the pressure is due to a permanent gas which is generated by the reaction.

**Hazard:** An inherent chemical or physical characteristic that has the potential for causing damage to people, property, or the environment.

**HAZAN:** The identification of undesired events that lead to the materialization of a hazard, the analysis of the mechanisms by which these undesired events could occur and usually the estimation of the consequences.

**HAZOP:** A systematic qualitative technique to identify process hazards and potential operating problems using a series of guidewords to study process deviations. A HAZOP is used to question every part of the process to discover what deviations from the start of the design can occur and what their causes and consequences may be. This is done systematically by applying suitable guidewords. This is a systematic detailed review technique for both batch and continuous plants, which can be applied to new or existing processes to identify hazards.

**Hazardous chemical reactivity:** Any chemical reaction with the potential to exhibit rates of increase in temperature and/or pressure too high to be absorbed by the environment surrounding the system. Included are reactive materials and unstable materials.

**Hybrid mixture:** A suspension of dust in air/vapor. Such mixtures may be flammable below the lower explosive limit of the vapor and can be ignited by low energy sparks.

**Hybrid system:** Hybrid systems are those in which the total pressure is due to both vapor pressure and permanent gas.

**Inherently safe:** A system is inherently safe if it remains in a non-hazardous situation after the occurrence of non-acceptable deviations from normal operating conditions.

**Inhibition:** A protective method where the reaction can be stopped by addition of another material.

**Interlock system:** A system that detects out-of-limits or abnormal conditions or improper sequences and either halts further action or starts corrective action.

**Isothermal:** A system condition in which the temperature remains constant. This implies that temperature increases and decreases that would otherwise occur are compensated by sufficient heat exchange with the environment of the system.

**Likelihood:** A measuring of the expected frequency in which an event occurs. This may be expressed as a frequency (e.g., events per year), a probability of occurrence during a time interval (e.g., annual probability), or a conditional probability (e.g., probability of occurrence, given that a precursor event has occurred).

**Limiting oxygen concentration (LOC):** Minimum concentration of oxygen in a mixture with gas, vapor, or dust that will allow it to burn.

**Liquids:** A vapor has to be produced at the surface of a liquid before it burns. Many common liquids give off a flammable concentration of vapor in air without being heated, sometimes at well below room temperature. Gasoline, for example, gives off ignitable vapors above -40° C, depending on the blend. The vapors are easily ignited by a small spark of flame.

**Lower explosive limit (LEL):** The concentration of a powder finely dispersed in air, below which no mixture likely to explode will be present.

**Lower flammable limit (LFL):** The lowest concentration of a vapor or gas (the lowest percentage of the substance in air) that will produce a flash of fire when an ignition source (heat, arc, or flame) is present.

**Maximum allowable working pressure (MAWP):** The maximum allowed pressure at the top of the vessel in its normal operating position at the operating temperature specified for that pressure.

**Maximum explosion overpressure,  $P_{max}$ :** The maximum pressure reached during an explosion in a closed vessel through systematically changing the concentration of dust-air mixture.

**Maximum reduced explosion overpressure,  $P_{red,max}$ :** The maximum pressure generated by an explosion of a dust-air mixture in a vented or suppressed vessel under systematically varied dust concentrations.

**Maximum explosion constant,  $K_{max}$ :** Dust and test-specific characteristic calculated from the cubic law. It is equivalent to the maximum rate of pressure rise in a 1-m<sup>3</sup> vessel.

**Maximum explosion pressure ( $P_{\max}$ ):** The maximum expected pressure for an explosion of the optimum concentration of the powder concerned in air, in a closed vessel under atmospheric starting conditions.

**Maximum explosion pressure rise ( $dP/dt)_{\max}$ :** The maximum pressure rise for an explosion of the optimum concentration of the powder concerned in air, in a closed vessel under atmospheric starting conditions. This explosion property depends on the volume of the vessel.

**Minimum Ignition Energy (MIE):** Used to measure the lowest energy at which an electrical discharge is just able to ignite the most sensitive mixture of the material in air.

**Minimum ignition temperature (MIT):** The lowest temperature of a hot surface which will cause a dust cloud to ignite and flame to propagate.

**Minimum oxygen concentration (MOC):** The concentration of oxygen in air, below which no mixture likely to explode will be formed with the presence of dust/air/inert mixture.

**Mitigation:** Lessening the risk of an accident event. A sequence of action on the source in a preventive way by reducing the likelihood of occurrence of the event, or in a protective way by reducing the magnitude of the event and for the exposure of local persons or property.

**Onset temperature:** The temperature at which the heat released by a reaction can no longer be completely removed from the reaction vessel, and consequently results in a detectable temperature increase. The onset temperature depends on detection sensitivity, reaction kinetics, on vessel size and on cooling, flow, and agitation characteristics.

**Oxidant:** Any gaseous material that can react with a fuel (either gas, dust, or mist) to produce combustion. Oxygen in air is the common oxidant.

**Overpressure:** A pressure increase over the set pressure of the relief device usually expressed as a percentage of gage set pressure.

**Phi-factor  $\phi$ :** A correction factor which is based on the ratio of the total heat capacity (mass  $\times$  specific heat) of a vessel to the total heat capacity of the vessel contents.

$$\phi = \frac{\text{Heat capacity of sample} + \text{Heat capacity of vessel}}{\text{Heat capacity of sample}}$$

The  $\phi$  factor enables temperature rises to be corrected for heat lost to the container or vessel. The  $\phi$  factor approaches the value of one for large vessels and for extremely low mass vessels.

**Pressure-relief device:** Designed to open to prevent a rise of internal fluid pressure in excess of a specified value due to exposure to emergency or abnormal conditions. It may also be designed to prevent excessive internal vacuum. It may be a pressure-relief valve, a non-reclosing pressure-relief device or a vacuum-relief valve.

**Process safety:** A discipline that focuses on the prevention of fires, explosions, and accidental chemical releases at chemical process facilities. Excludes classic worker health and safety issues involving working surfaces, ladders, protective equipment, and so on.

**Purge gas:** A gas that is continuously or intermittently added to a system to render the atmosphere non-ignitable. The purge gas may be inert or combustible.

**Quenching:** Rapid cooling from an elevated temperature, for example severe cooling of the reaction system in a short time (almost instantaneously), "freezes" the status of a reaction and prevents further decomposition.

**Relieving pressure:** Set pressure plus overpressure.

**Risk:** The likelihood of a specified undesired event occurring within a specified period or in specified circumstances.

**Risk analysis:** A methodical examination of a process plant and procedure which identifies hazards, assesses risks, and proposes measures which will reduce risks to an acceptable level.

**Runaway:** A thermally unstable reaction system, which shows an accelerating increase of temperature and reaction rate. The runaway can finally result in an explosion.

**Rupture disk device:** A non-reclosing pressure-relief device actuated by inlet static pressure and designed to function by the bursting of a pressure-containing disk.

**Safety relief valve:** A pressure-relief valve characterized by rapid opening pop action or by opening generally proportional to the increase in pressure over the opening pressure. It may be used for either compressible or incompressible fluids, depending on design, adjustment, or application.

**Set pressure:** The inlet pressure at which the relief device is set to open (burst).

**Stagnation pressure:** The pressure that would be observed if a flowing fluid were brought to rest along an isentropic path.

**Static activation pressure,  $P_{\text{stat}}$ :** Pressure which activates a rupture disk or an explosion door.

**Superimposed back pressure:** The static pressure existing at the outlet of a pressure-relief device at the time the device is required to operate. It is the result of pressure in the discharge system from other sources.

**Temperature of no-return:** Temperature of a system at which the rate of heat generation of a reactant or decomposition just exceeds the rate of heat loss and will lead to a runaway reaction or thermal explosion.

**Thermally unstable:** Chemicals and materials are thermally unstable if they decompose, degrade, or react as a function of temperature and time at or about the temperature of use.

**Thermodynamic data:** Data associated with the aspects of a reaction that are based on the thermodynamic laws of energy, such as Gibbs' free energy, and the enthalpy (heat) of reaction.

**Time to maximum rate (TMR):** The time taken for a material to self-heat to the maximum rate of decomposition from a specific temperature.

**Unconfined vapor cloud explosion (UCVE):** Occurs when sufficient amount of flammable material (gas or liquid having high vapor pressure) gets released and mixes with air to form a flammable cloud such that the average concentration of the material in the cloud is higher than the lower *limit of explosion*. The resulting explosion has a high potential of damage as it occurs in an open space covering large areas. The flame speed may accelerate to high velocities and produce significantly blast overpressure. Vapor cloud explosions in densely packed plant areas (pipe lanes, units, etc.) may show accelerations in flame speeds and intensification of blast.

**Upper explosive limit (UEL) or upper flammable limit (UFL):** The highest concentration of a vapor or gas (the highest percentage of the substance in the oxidant) that will produce a flash or fire when an ignition source (heat, arc, or flame) is present.

**Vapor specific gravity:** The weight of a vapor or gas compared to the weight of an equal volume of air, an expression of the density of the vapor or gas. Materials lighter than air have vapor specific gravity less than 1.0 (e.g., acetylene, methane, hydrogen). Materials heavier than air (e.g., ethane, propane, butane, hydrogen, sulphide, chlorine, sulfur dioxide) have vapor specific gravity greater than 1.0.

**Vapor pressure:** The pressure exerted by a vapor above its own liquid. The higher the vapor pressure, the easier it is for a liquid to evaporate and fill the work area with vapors which can cause health or fire hazards.

**Vapor-pressure system:** A vapor-pressure system is one in which the pressure generated by the runaway reaction is solely due to the

increasing vapor pressure of the reactants, products, and/or solvents as the temperature rises.

**Venting (emergency relief):** Emergency flow of vessel contents out of the vessel. The pressure is reduced by venting, thus avoiding a failure of the vessel by overpressurization. The emergency flow can be one-phase or multiphase, each of which results in different flow and pressure characteristics. Multiphase flow, for example vapor and or gas/liquid flow, requires substantially larger vent openings than single-phase vapor (and/or gas) flow for the same depressurization rate.

**Vent area, A:** Area of an opening for explosion venting.

## ACRONYMS AND ABBREVIATIONS

AGA	American Gas Association	$a$	area, in. <sup>2</sup>
AIChE	American Institute of Chemical Engineers	$a_p$	cross-sectional area of the inlet pipe, ft <sup>2</sup>
AIChE/CCPS	American Institute of Chemical Engineers – Center for Chemical Process Safety	$A$	area, m <sup>2</sup> , ft <sup>2</sup> , or in. <sup>2</sup> ; consistent with equation units
AIChE/DIERS	American Institute of Chemical Engineers – Design Institute for Emergency Relief Systems	or $A$	nozzle throat area, or orifice flow area, effective discharge area (calculations required)
AIT	Auto-Ignition Temperature	$A_1$	or from manufacturer's standard orifice areas, in. <sup>2</sup>
API	American Petroleum Institute	$A_2$	initial vessel relief area, in. <sup>2</sup> or m <sup>2</sup>
ARC	Accelerating Rate Calorimeter	$A_3$	second vessel relief area, in. <sup>2</sup> or m <sup>2</sup>
ASME	American Society of Mechanical Engineers	$A_s$	exposed surface area of vessel, ft <sup>2</sup>
ASTM	American Society for Testing Materials	$A_v$	internal surface area of enclosure, ft <sup>2</sup> or m <sup>2</sup>
bar-m/sec	Bar-meter per second	$A_w$	vent area, m <sup>2</sup> or ft <sup>2</sup>
BLEVE	Boiling Liquid Expanding Vapor Explosion	$B$	total wetted surface area, ft <sup>2</sup>
CFD	Computational fluid dynamics	BP	cubical expansion coefficient per of liquid at expected temperature (see tabulation in text)
CSB	US Chemical Safety and Hazard Investigation Board	B.P.	boiling point, °C or °F
CPI	Chemical Process Industry	Bar	burst pressure, either psig or psia
DIERS	Design Institute for Emergency Relief Systems.	$C1 = c = C$	14.5 psi = 0.987 atm = 100 kPa atm; 14.7 psia = 1.01 bar
DOE	Department of Energy	$C_p/C_v$	gas/vapor flow constant depending on ratio of specific heats $C_p/C_v$
EFCE	European Federation of Chemical Engineers	$C_o$	ratio of specific heats
EPA	U.S. Environmental Protection Agency	$C(\text{psi})^{1/2}$	sonic flow discharge orifice constant, varying with Reynolds number
HAZOP	Hazard and Operability	$C_h$	venting equation constant, fuel characteristic
HAZAN	Hazard Analysis	$C_2$	constant for explosion venting equation
HMSO	Her Majesty's Stationery Office	$c$	specific heat of trapped fluid, Btu/lb° F
HRA	Human Reliability Analysis	$d$	subsonic flow constant for gas or vapor, function of $k = C_p/C_v$
HSE	Health and Safety Executive, United Kingdom	$d'$	orifice coefficients for liquids
IChemE	Institution of Chemical Engineers (UK)	$D = d_t$	diameter, in. (usually of pipe)
ICI	Imperial Chemical Industries	$D_F$	flare tip diameter, ft
LFL	Lower Flammable Limit	$dp/dt$	flare tip diameter, in.
LNG	Liquefied Natural Gas	$E$	minimum distance from midpoint of flame to the object, ft
LPG	Liquefied Petroleum Gas	$e$	rate of pressure rise, bar/s or psi/s
MSDS	Material safety data sheet	$e_t$	joint efficiency in cylindrical or spherical shells or ligaments between openings (see ASME code Par.UW-12 or UG-53)
NFPA	National Fire Protection Agency	$F$	natural logarithm base, e = 2.718
NIOSH	National Institute for Occupational Safety and Health	$F'_{\text{gs}}$	TNT equivalent (explosion)
OSHA	Occupational Safety and Health Administration	$F = F_h$	environment factor for
PFD	Process Flow Diagram	$F'$	relief valve factor for non-insulated vessels in gas service exposed to open fires
PHA	Preliminary Hazard Analysis	$F$	fraction of heat radiated
P&ID	Piping and Instrumentation Diagram	$F_u$	operating environment factor for safety relief of gas only vessels
TNT	Trinitrotoluene	$F_y$	Flow gas/vapor, cubic feet per minute at 14.7 psia and 60° F
TLV	Threshold Limit Values	$f_1$ or $f_2$	The ratio of the ultimate stress of the vessel to the allowable stress of the vessel
UFL	Upper Flammable Limit	$F_2$	ratio of the yield stress of the vessel to the allowable stress of the vessel
VCDM	Vapor Cloud Dispersion Modeling	°F	relief area for vessels 1 or 2 respectively, ft <sup>2</sup>
VCE	Vapor Cloud Explosion	$f$	coefficient of subcritical flow
VDI	Verein Deutscher Ingenieure	$f_q$	temperature, °F
VSP	Vent Sizing Package	$G$	specific relief area, sq meter/cubic meter, or area/unit volume
			stream quality, degrees fraction
			specific gravity of gas (air = 1), or specific gravity of liquid (water = 1) at actual discharge temperature

## NOMENCLATURE

GPM	gallons per minute flow	MAWP	maximum allowable working pressure of a pressure vessel, psi gauge (or psi absolute if so specifically noted)
$G$	acceleration of gravity, $32.2 \text{ ft/s}^2$		
$H_c$	heat of combustion of gas/vapor, Btu/lb	MP	melting point (freezing point), $^\circ\text{C}$ or $^\circ\text{F}$
$H$	total heat transfer rate, Btu/h	$MR$	universal gas constant = $1545 \text{ ft lb}_f/\text{lb sec-sec}$ .
$h_1$	seal, submerged, ft		Units depend on consistency with other symbols in equation, or manufacturing range for metal bursting/rupture disks.
$h_L$	seal pipe clearance		spark energy, milli-joules
$h$	head of liquid, ft		mass of TNT, lb
$h_c$	net calorific-heat value, Btu/scf		moles of specified components
$j$	number of purge cycles (pressurizing and relief)		total number moles at pressure or atmospheric condition
$K$	permissible design level for flare radiation (including solar radiation), Btu/h/ $\text{ft}^2$		total number mols at atmospheric pressure or low pressure or vacuum condition
$K_p$	liquid capacity correction factor for overpressures lower than 25%. Non-code equations only.	$P$	relieving pressure, psia = valve set pressure + permissible overpressure, psig, + 14.7, or any pressure, bar (gauge), or a consistent set of pressure units. Minimum overpressure is 3 psi pressure, psia
$K_b$	vapor or gas flow correction factor for constant back pressures above critical pressure	$P_1 = P'$	maximum header exit pressure into seal, psig
$K_v$	vapor or gas flow factor for variable back pressures. Applies to Balanced Seal valves only.	$p''$	stamped bursting pressure plus overpressure allowance (ASME 10% or 3 psi, whichever is greater) plus atmospheric pressure (14.7), psia
$K_w$	liquid correction factor for variable back pressures. Applies to balanced seal valves only. Conventional valves require no correction.	$P_b$	critical pressure of a gas system, psia
$K_u$	liquid viscosity correction factor	$P_c = P_{\text{crit}}$	design pressure of vessel or system to prevent deformation due to internal deflagration, psig
$K_{sh}$	steam superheat correction factor	$P_d$	ASME code design pressure (or maximum allowable working pressure), psi
$K_n$	Napier steam correction factor for set pressures between 1500 and 2900 psig	$P_d$	pressure on outlet side of rupture disk, psia
$K = K_d$	coefficient of discharge: <sup>*</sup> 0.975 for air, steam, vapors and gases 0.724 for ASME code liquids** 0.64 for non-ASME code liquids 0.62 for bursting/rupture disk	$P_{do}$	exit or back pressure, psia, stamped burst pressure
$K_d$	discharge coefficient orifice or nozzle	$P_e$	perimeter of a cross section, ft or meters
$K_c$	deflagration index, maximum rate of pressure rise for gases, bar m/s = bar m/s	$P_{er}$	initial high pressure, mmHg
$K_{st}$	deflagration index, maximum rate of pressure rise for dusts, bar m/s = bar m/s	$P_H$	maximum initial pressure at which the combustible atmosphere exists, psig
$K_w$	variable or constant back pressure sizing factor, balanced valves, liquids only	$P_i$	initial pressure of system, psia
$k$	ratio of specific heats, $C_p/C_v$	$P'_i$	initial low pressure or vacuum, mmHg
$W_L$	liquid flow, gallons per minute	$P_L$	maximum explosion pressure, bar, or other consistent pressure units
$L_f$	length of flame, ft	$P_{\max}$	maximum pressure developed in an unvented vessel, bar (gauge) or psig
$L_v = L$	latent heat of vaporization, Btu/lb	$P_{uv}$	Normal expected or maximum expected operating pressure, psia
$L_X$	distance between adjacent vents, m or ft	$P_{op}$	relieving pressure, psia, or sometimes upstream pressure, psia, or initial pressure of system
LEL	lower explosive, or lower flammable limit, per cent of mixture of flammable gases only in air	$P_i, P_o$	design pressure to prevent rupture due to internal deflagration, psig
$L_1, L_2$ , and so on	lower flammability limits, vol % for each flammable gas in mixture	$P_r$	reduced pressure termed maximum internal overpressure that can be withstood by the weakest structural element, psig, or barga, or kPa (gas/vapor), or maximum pressure actually developed during a vented deflagration
$L_3$	longest dimension of the enclosure, ft	$P_{red}$	maximum pressure developed during venting, barga (dusts)
$L/D$	length-to-diameter ratio, dimensionless	$P_{stat}$	vent closure release pressure, barga (dusts)
$M$	molecular weight	$P_\eta$	normal operating gas pressure, psia
$m$	meter or percent moisture, or 100 minus steam quality	$\Delta P_t$	pressure drop at flare tip, inches water
		$P_t$	pressure of the vapor just inside flare tips (at top), psia
		$P_1$	upstream relieving pressure, or set pressure at inlet to safety relief device, psig (or psia, if consistent)
		$P_2$	back pressure or downstream at outlet of safety relief device, psig, or psia, depending on usage
		$p$	rupture pressure for disk, psig or psia
		$P_o$	overpressure (explosion), lb force/in. <sup>2</sup>

\* Where the pressure-relief valve is used in series with a rupture disk, a combination capacity factor of 0.8 must be applied to the denominator of the above valve equations. Consult the valve manufacturer (also see specific section this chapter of text). For higher factors based on National Board flow test results conducted with various rupture disk designs/arrangements (see Table 9-12).

\*\* For saturated water see ASME code, Appendix 11-2.

$p'$	pressure, psia	$v$	velocity, ft/s, or dust vessel volume, $\text{ft}^3$ or
$\Delta P$	pressure differential across safety relief valve, inlet pressure minus back pressure or down stream pressure, psi. Also = set pressure + overpressure, psig-back pressure, psig. At 10% over-pressure $\Delta P$ equals $1.1 P_1 - P_2$ . Below 30 psig set $\Delta P$ equals $P_1 + 3 - P_2$ .	$V$	vessel volume, cubic meters or cubic feet, or required gas capacity in scfm or
$\Delta P$	differential pressure across liquid relief rupture disk, usually equals $p$ , psig	$V$	vapor flow required through valve (sub-critical), -Std
$\Delta P$ (dusts)	pressure differential, bar or psi	$\bar{V}$	$\text{ft}^3/\text{min}$ at 14.7 psia and $60^\circ\text{F}$
$Q$	total heat absorption from external fire (input) to the wetted surface of the vessel, Btu/h	$V_a$	specific volume of fluid, $\text{ft}^3/\text{lb}$
$Q'$	liquid flow, $\text{ft}^3/\text{s}$	$V_c$	required air capacity, scfm
$Q_A$	required flow, $\text{ft}^3/\text{min}$ at actual flowing temperature and pressure, acfm		cubic feet of free air per hour which is 14.7 psia and $60^\circ\text{F}$ , or from Eq. (9-49), for wetted area
$Q_f$	heat released by flame, Btu/h		$A_w > 2800 \text{ ft}^2$
$Q_r$	heat release, lower heating value, Btu/h	$V'$	venting requirement, cubic feet free air per hour at 14.7 psia and $60^\circ\text{F}$
$Q_r$	required flow, $\text{ft}^3/\text{min}$ at standard conditions of 14.7 psia and $60^\circ\text{F}$ , scfm	$V_L$	flow rate at flowing temperature, US gpm, or required liquid capacity in US gpm
$q$	Average unit heat absorption, Btu/h/ $\text{ft}^2$ of wetted surface	$v$	shock velocity, ft/s or ft/min (depends on units selected)
$R$	ratio of the maximum deflagration pressure to the maximum initial pressure, as described in NFPA code-69, Par 5-3.3.1	$V_1$	specific volume of gas or vapor at upstream or relief pressure and temperature conditions, $\text{ft}^3/\text{lb}$
also $R$	individual gas constant = $MR/M = 1545/M$	$v_s$	sonic velocity of gas, ft/s
$R'$	adjusted value of $R$ , for NFPA code-69	$V_1, V_2$	volume percent of each combustible mixture, free from air or inert gas
$R_{\exp}$	distance from center of explosion source the point of interest, ft.	$W$	required vapor capacity in pounds per hour any flow rate in pounds per hour, vapor relief rate to flare stack, lb/h
$Re$	Reynolds number (or sometimes, $N_{Re}$ )	$W_c$	charge weight of explosive of interest, lb
$R_g$	Universal gas constant = $1545 = MR$	$W_e$	effective charge weight, pounds of TNT for estimating surface burst effects in free air
$R_i$	inside radius of vessel, no corrosion allowance added, in.	$W_s$	required steam capacity flow or rate in pounds per hour, or other flow rate, lb/h
$^\circ R$	temperature, absolute, degrees Rankine	$W_{he}$	hydrocarbon to be flared, lb/h
$r = rc$	ratio of back pressure to upstream pressure, $P_2/P_1$ , or critical pressure ratio, $P_c/P_1$	$W_{TNT}$	equivalent charge weight of TNT, lb
$r$	relative humidity, percent	$W_L$	liquid flow rate, gpm
$S$	maximum allowable stress in vessel wall, from ASME code, psi., UCS-23.1-23.5; UHA-23, UHT-23	$W_{steam}$	steam injected into flare, lb/h
$S' = \text{sp gr}$	specific gravity of liquid, referenced to water at the same temperature	$Y_f$	final oxidant concentration, mol fraction
$S_G = Sg$	sp gr of gas relative to air, equals ratio of mol wt of gas to that of air, or liquid fluid	$y_j$	specified component concentration after "j" purges
	specific gravity relative to water, with water = 1.0 at $60^\circ\text{F}$	$y_o$	initial concentration of component (oxidant) under low pressure, mol fraction
sp gr	specific gravity of fluid, relative to water = 1.0	$Z$	compressibility factor, deviation of actual gas from perfect gas law. Usually $Z = 1.0$ at low pressure below 300 psig.
St	dust hazard class St = stainless steel	$Z$ or $Z_{TNT}$	scaled distance for explosive blasts, $\text{ft}/(\text{lb})^{1/3}$
SSU	viscosity Saybolt universal seconds	$Z$	actual distance for explosion damage, ft
$^\circ S$	degrees of superheat, $^\circ\text{F}$		
$T$	absolute inlet or gas temperature, degrees Rankin $^\circ R = ^\circ F + 460$ , or temperature of relief vapor [48]; $^\circ R$		
$T_\eta$	normal operating gas temperature, $^\circ R$		
$T_1$	operating temperature, $^\circ\text{C}$ (NFPA code-59)		
$T_o$	temperature of service, $^\circ R$		
$T_W$	vessel wall temperature, $^\circ R$		
$T_1$	gas temperature, $^\circ R$ , at the upstream pressure, determined from $T_1 = (P_1/P_\eta)(T_\eta)$		
$t$	minimum required thickness of shell of vessel, no corrosion, in.		
$U$	viscosity at flowing temperature, SSU		
$U_\infty$	lateral wind velocity, ft/s		
$U_j$	flare tip velocity, ft/s		
UEL	upper explosive or flammable limit, percent of mixture of flammable gases only in air		

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62. Leung, J.C. and M. Epstein, "A Generalized Critical Flow Model for Nonideal Gases", *AIChEJ*, Vol. 34, No. 9, Sep 1988, pp. 1568–1572.

## WORLD WIDE WEB ON DUST EXPLOSIONS AND TWO-PHASE RELIEF SYSTEMS

[www.csb.gov](http://www.csb.gov)  
[www.sciencedirect.com](http://www.sciencedirect.com)  
[www.MegaSpider.com](http://www.MegaSpider.com)  
[www.chilworth.com](http://www.chilworth.com)  
[www.fauske.com](http://www.fauske.com)  
[www.helgroup.co.uk](http://www.helgroup.co.uk)  
[www.chemvillage.org](http://www.chemvillage.org)  
[www.chemeng.ed.ac.uk](http://www.chemeng.ed.ac.uk)  
[www.chemsafety.gov](http://www.chemsafety.gov)  
[www.kbintl.com](http://www.kbintl.com)  
[www.rcostello.com](http://www.rcostello.com)  
[www.che.ufl.edu](http://www.che.ufl.edu)  
[www.osha-slc.gov](http://www.osha-slc.gov)  
[www.icHEME.org](http://www.icHEME.org)  
[www.harsnet.de/presEng.htm](http://www.harsnet.de/presEng.htm)

---

## Appendix A

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# A LIST OF ENGINEERING PROCESS FLOW DIAGRAMS AND PROCESS DATA SHEETS

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### A-1 PROCESS FLOW DIAGRAMS USING VISIO 2002 SOFTWARE

Figure 1-12b Process flow diagram (Feed and fuel desulfurization section).

Figure 1-12c Typical process flow diagram for the production of Methyl Tertiary Butyl Ether (MTBE).

Figure 1-14 Piping and instrumentation diagram for Ammonia plant CO<sub>2</sub> removal.

Figure 1-15 Piping and instrumentation diagram: Ammonia synthesis and refrigeration unit (2).

### A-2 PROCESS DATA SHEETS

1. Air cooled heat exchanger process data sheet
2. Centrifugal pump schedule: driver
3. Centrifugal pump schedule: pump
4. Centrifugal pump summary
5. Column schedule
6. Deaerator process data sheet: Deaerated water storage tank
7. Deaerator process data sheet: Deaerator head
8. Drum process data sheet
9. Effluent schedule
10. Equilibrium flash calculation
11. Fabricated equipment schedule
12. Fan/Compressor process duty specification
13. Fractionator calculation summary
14. General services and utilities checklist
15. Hazardous chemical and conditions schedule
16. Heat and mass balances
17. Heat exchanger rating sheet
18. Hydrocarbon dew point calculation
19. Line list schedule
20. Line schedule
21. Line schedule sheet
22. Line summary table
23. Mass balance
24. Mechanical equipment schedule
25. Pipe line list
26. Pipe list
27. Piping process conditions summary
28. Plate heat exchanger data sheet
29. Calculation of pressure drop in fixed catalyst beds
30. Process engineering job analysis summary
31. Pump calculation sheet
32. Pump schedule
33. Relief device philosophy sheet
34. Tank and vessel agitator data sheet
35. Tank process data sheet
36. Tank schedule
37. Tie-in-schedule
38. Tower process data sheet
39. Tray loading summary
40. Trip schedule
41. Utility summary sheet
42. Vessel and tank schedule
43. Vessel and tank summary: driver
44. Vessel schedule
45. Water analysis sheet





**CENTRIFUGAL PUMP SCHEDULE:  
DRIVER**



### CENTRIFUGAL PUMP SCHEDULE: PUMP



## CENTRIFUGAL PUMP SUMMARY

	<b>SQ2R</b>	COLUMN SCHEDULE												
	Job No.:			Sheet of	Issue No.:	1	Date	2	Date	3	Date			
	Job Name:				Description									
	Document No.				Made/Revised by									
	Project No.				Checked by									
	Client				Approved Process									
	Location				Approved									
	Plant													
1	Item Number													
2	Item Name													
3	Number Required													
4		Units	Top	Bottom	Top	Bottom	Top	Bottom	Top	Bottom	Top	Bottom		
5	Shell Diameter (ID/OD)	mm												
6	Shell Length (T.L. - T.L.)	mm												
7	Overall Shell Length - (T.L. - T.L.)	mm												
8	Base Elevation	mm												
9	Trays - Type													
10	- Number													
11	- Spacing	mm												
12	- Liquid Flow													
13	Packing - Type													
14	- Number of Beds													
15	- Bed Height/Volume	mm/m <sup>3</sup>												
16	- Density	kg/m <sup>3</sup>												
17	Other Internals													
18														
19														
20														
21	Operating - Pressure													
22	- Temperature													
23	Design - Pressure	barg												
24	- Temperature	°C												
25	Vacuum Design													
26	Material: Shell													
27	Liner													
28	Internals													
29														
30	Shell Corrosion Allowance	mm												
31	Sour Service													
32	Stress Relieved for Process Reasons													
33	Insulation													
34														
35														
36	Notes													
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Σ <small>A.K.C. TECHNOLOGY</small> 2K		DEAERATOR PROCESS DATA SHEET			Document No. Sheet of Rev.																																																																																																																
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Item Name		<b>DEAERATOR HEAD</b>			No. Working	Total No. Off																																																																																																															
					<p>PERFORATED AREA</p> <ol style="list-style-type: none"> <li>1. DESIGN FOR FULL VACUUM. OPERATING TEMPERATURE _____ OPERATING PRESSURE _____</li> <li>2. TRAYS TO BE SEALED TO SHELL WITH STEAM QUALITY RUBBER GASKETS.</li> <li>3. C4 TO BE CAPPED FOR SITE INSTALLATION OF STANDARD SAMPLING EQUIPMENT.</li> <li>4. HEAD TO BE IN S.S OR COATED M.S. BELOW 120°C MAX. EPOXY RESIN BELOW 180°C MAX. STOVED PHENOLIC RESIN (SAKAPHEN). PROCESS TO ADVISE ON SPECIFICATION &amp; APPLICATION.</li> <li>5. TRAYS TO BE IN 18/8 S/S. TRIANGULAR PERFORATION. PATTERN TO BE N.GREENING'S _____ OR EQUAL.</li> <li>6. SPRAY NOZZLE TO BE IN 18/8 S/S. SIMILAR TO DELAVAN WATSON S. _____ AND WITHDRAWABLE THROUGH C-1.</li> </ol>																																																																																																																
<p>NOTE: Bubbling Area Is Any Part Of Tray Within 4" Distance Of A Hole</p> <table border="1" style="width: 100%; border-collapse: collapse;"> <thead> <tr> <th style="width: 15%;">NOZZLES</th> <th style="width: 15%;">MARK</th> <th style="width: 15%;">SIZE</th> <th style="width: 15%;">QTY</th> <th colspan="4" style="width: 40%;"></th> </tr> </thead> <tbody> <tr> <td>Water Inlet</td> <td>C-1</td> <td></td> <td></td> <td colspan="4"></td> </tr> <tr> <td>Vent</td> <td>C-3</td> <td></td> <td></td> <td colspan="4"></td> </tr> <tr> <td>Sample Point</td> <td>C-4</td> <td>STD</td> <td>1</td> <td colspan="4"></td> </tr> </tbody> </table> <p>Perf. Area/Tray Bubbling Area/Tray</p> <table border="1" style="width: 100%; border-collapse: collapse;"> <thead> <tr> <th style="width: 15%;">No. Of Trays</th> <th style="width: 15%;">Tray Thickness</th> <th style="width: 15%;">Hole Diameter</th> <th style="width: 15%;">Pitch. Approx.</th> <th colspan="4" style="width: 40%;"></th> </tr> </thead> <tbody> <tr> <td></td> <td></td> <td></td> <td></td> <td colspan="4"></td> </tr> <tr> <td>No. of Holes/Tray</td> <td></td> <td></td> <td></td> <td colspan="4" style="text-align: center;">* No. Of Hole Is More Important Than Pitch</td> </tr> </tbody> </table> <table border="1" style="width: 100%; border-collapse: collapse;"> <thead> <tr> <th style="width: 15%;">Description</th> <th style="width: 15%;">1</th> <th style="width: 15%;">Date</th> <th style="width: 15%;">2</th> <th style="width: 15%;">Date</th> <th style="width: 15%;">3</th> <th style="width: 15%;">Date</th> <th style="width: 15%;">4</th> <th style="width: 15%;">Date</th> <th style="width: 15%;">5</th> <th style="width: 15%;">Date</th> </tr> </thead> <tbody> <tr> <td>Made/Revised by</td> <td></td> </tr> <tr> <td>Checked by</td> <td></td> </tr> <tr> <td>Approved Process</td> <td></td> </tr> <tr> <td>Approved by</td> <td></td> </tr> </tbody> </table>							NOZZLES	MARK	SIZE	QTY					Water Inlet	C-1							Vent	C-3							Sample Point	C-4	STD	1					No. Of Trays	Tray Thickness	Hole Diameter	Pitch. Approx.													No. of Holes/Tray				* No. Of Hole Is More Important Than Pitch				Description	1	Date	2	Date	3	Date	4	Date	5	Date	Made/Revised by											Checked by											Approved Process											Approved by										
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Job No.:		Sheet of	Issue No.:		1	Date	2	Date
Job Name:			Description					
Document No.			Made/Revised by					
Project No.			Checked by					
Client			Approved- Process					
Location			Approved					
Plant								
STREAM NUMBER	EFFLUENT SOURCE	FLUID	TOTAL FLOWRATE tonne/h.	MAIN CONTAMINANTS	CONTAMINANT kg/h. QUANTITY	CONTAMINANT CONCENTRATION	STREAM TEMP. °C.	
1								
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 <small>A.K.C. TECHNOLOGY</small>	<b>PROCESS DATA SHEET</b>								Job No.			
	Job:								Sheet of			
<b>EQUILIBRIUM FLASH CALCULATION (Form 1 of 2)</b>												
Temperature: ..... °F Pressure: ..... psia Pg: ..... psia												
Assumed $\frac{V}{L} =$  H <sub>2</sub> CO CH <sub>4</sub>  CO <sub>2</sub> C <sub>2</sub> H <sub>6</sub> C <sub>3</sub> H <sub>8</sub>  i-C <sub>4</sub> H <sub>10</sub> n-C <sub>4</sub> H <sub>10</sub> C <sub>2</sub> H <sub>4</sub>  C <sub>3</sub> H <sub>6</sub>  C <sub>6</sub> H <sub>6</sub>	$m$ $K$ $K \frac{V}{L} + 1$ $\frac{m}{K \frac{V}{L} + 1}$	$\frac{V}{L} + 1$ $\frac{m}{K \frac{V}{L} + 1}$ $K \frac{V}{L} + 1$ $\frac{m}{K \frac{V}{L} + 1}$	$\frac{V}{L} + 1$ $\frac{m}{K \frac{V}{L} + 1}$ $K \frac{V}{L} + 1$ $\frac{m}{K \frac{V}{L} + 1}$	$\frac{V}{L} + 1$ $\frac{m}{K \frac{V}{L} + 1}$ $K \frac{V}{L} + 1$ $\frac{m}{K \frac{V}{L} + 1}$	$\frac{V}{L} + 1$ $\frac{m}{K \frac{V}{L} + 1}$ $K \frac{V}{L} + 1$ $\frac{m}{K \frac{V}{L} + 1}$							
	$\sum \frac{m}{K \frac{V}{L} + 1} =$											
	Calculated $\frac{V}{L} =$											
	$\text{Calculated } \frac{V}{L} = \frac{1}{\sum \frac{m}{K \frac{V}{L} + 1}} - 1$											
	Issue No.		1	Date	2	Date	3	Date	4	Date	5	Date
	Made/Revised by											
Checked by												
Approved - Process												
Approved by												



## **PROCESS DATA SHEET**

Job No.

**Item No.**

| Job:

Sheet of

## **EQUILIBRIUM FLASH CALCULATION (Form 2 of 2)**

Temperature: ..... °F  
Pressure: ..... psia  
Pg: ..... psia

	MOLES			MOL. FRACTION	
	Feed m	Liquid $L_x$	Gas $V_y$	Liquid $x$	Gas $y$
H <sub>2</sub>					
CO					
CH <sub>4</sub>					
CO <sub>2</sub>					
C <sub>2</sub> H <sub>6</sub>					
C <sub>3</sub> H <sub>8</sub>					
i-C <sub>4</sub> H <sub>10</sub>					
n-C <sub>4</sub> H <sub>10</sub>					
C <sub>2</sub> H <sub>4</sub>					
C <sub>3</sub> H <sub>6</sub>					
C <sub>6</sub> H <sub>6</sub>					

$$L_x = \frac{m}{K \frac{V}{L} + 1} \quad V_y = m - L_x$$



**FABRICATED EQUIPMENT SCHEDULE**

Notes: 1. H.E. Type Coding - A.C. = Air Cooler; K = Kettle; U.B. = U - Tube Bundle; F.H. = Floating Head; F.T.S. = Fixed Tube Sheet

783





\* ASSUME 1 HORSE POWER = 1 KVA

NOTE: ONE COPY OF THIS LIST TO BE GIVEN TO SERVICE AND UTILITY SECTION.



## HAZARDOUS CHEMICAL AND CONDITIONS SCHEDULE

**Note:**

1. This Schedule is given without any Legal responsibility on the part of AKC Technology.
  2. Column 3; Vapor - V, Liquid - L, Solid - S
  3. Toxicity: Denotes Threshold Limit (T.L.) under which it is believed nearly all workers may be exposed day after day without adverse effect. The figures relate to average concentration for a normal working day.

#### 4. Column 11

A – Water                  E – Foam

5. Column 12 B.A. – Breathing Apparatus

## G.V. – Gloves

B - Powered Talc F - See Remarks

C = CO<sub>2</sub>

C - CO<sub>2</sub>  
D - Dry Chemical

D = Dry Chemical

### **C. – Full Clothing**

*W.* = Footware

W. = Footwear

10 / 10



## HEAT AND MASS BALANCES

Job No.:		Sheet of	Issue No: Made/Revised by Checked by Approved Process Approved by	1	Date	2	Date	3	Date
Job Name:									

*delete unit as necessary		Units	→?	°API	K	M.W.	FLOWING SpGr	*° F °C	*barg psig	B.P.S.D. at 60°F	FLOWING GPM	Mol %	Mol/h	*kg/h lb/h	*kcal/kg Btu/lb	*106 kcal/h 106 Btu/h
Stream	↓															
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HEAT EXCHANGER RATING SHEET				Document No.									
				Sheet of	Rev.								
				Items No. (s)									
<b>PART 1 - TO BE FILLED IN COMPLETELY</b>													
1 Exchanger Name:	Location:		Arrangement: Horiz = 0; Vert = 1 Units: S.I. 0; Engineering = 1										
2 Job:													
3 Type of Unit: F.T.S.=0: FLTG.HEAD=1; U-TUBE=2; FOR KETTLES ADD 3													
4 Hot Side: Either=0; Shell=1; Tubes=2													
5 Units	S.I.	Engineering	HOT FLUID	COLD FLUID									
6 Fluid Circulated	-	-											
7 Total Fluid Entering (Normal)	kg/hr	lb/hr											
8 Flow Margin	%	%											
9 Temperature In/Out	°C	°F											
10 Max Pressure Drop at line 8	bar	psi											
11 Inlet Pressure Operating/Design	barA	psia											
12 Normal Heat Load	kW	Btu/hr											
13 Heat Load Margin	%	%											
14 Design Temperature	°C	°F											
15 Corrosion Allowance	mm	ins											
16 Fouling Resistance	°C ft²/Wt	'F hr ft²/Btu											
17 Line N.B. In/Out	mm	ins											
18 Part 2													
19 GAS (AD VAPOR) IN	-	-											
20 Flow of Vapor and Gas at Inlet	kg/hr	lb/hr											
21 Code Number	-	-											
22 Molecular Weight	-	-											
23 Thermal Conductivity	W/m °C	Btu/hr ft °F											
24 Specific Heat	kJ/kg °C	Btu/lb °F											
25 Compressibility Factor	-	-											
26 Viscosity	cP	lb/ft hr											
27 LIQUID IN	-	-											
28 Total Flow of Liquid at Inlet	kg/hr	lb/hr											
29 Code Number	-	-											
30 Thermal Conductivity	W/m °C	Btu/hr ft °F											
31 Specific Heat	kJ/kg °C	Btu/lb °F											
32 Density	kg/m³	lb/ft³											
33 Viscosity	cP	lb/ft hr											
34 Viscosity at Ave. Temp. other side	cP	lb/ft hr											
35 CONDENSATION AND VAPORIZATION	-	-											
36 Fluid Condensed or Vaporized	kg/hr	lb/hr											
37 Code Number	-	-											
38 Molecular Weight	-	-											
39 Thermal Conductivity	W/m °C	Btu/hr ft °F											
40 Specific Heat	kJ/kg °C	Btu/lb °F											
41 Density (Hot Fluid)	kg/m³	lb/ft³											
42 Compressibility Factor (Cold Fluid)	-	-											
43 Viscosity	cP	lb/ft hr											
44 Latent Heat	kJ/kg	Btu/lb											
45 Surface Tension	N/m	lb/ft											
46 Expansion Coefficient	1/ °C	1/ °F											
47 % Condensate Forming Film	%	%											
48 Thermal Conductivity	W/m °C	Btu/hr ft °F											
49 Density	kg/m³	lb/ft³											
50 Viscosity	cP	lb/ft hr											
51 Temps for Phase Change Start/Finish	°C	°F											
52 LMTD weighting Factor °F	-	-											
53 PART 3													
54 Number of Points in Table Below	%	%	1	2	3	4	5	6	7	8	9	10	
55 % Heat Load	°C	°F											
56 Hot Fluid Temperature	°C	°F											
57 Cold Fluid Temperature	°C	°F											
<b>PART 4 - CODE NUMBERS TO BE FILLED IN COMPLETELY</b>													
59 Material for	Description	Code No.	O.D.	Thickness	Length	Passes	Number	Contents Lethal?					
60 Tubes								Yes/No					
61 Shell								Yes/No					
62 Floating Head					Tube Pitch = mm/in.	<input type="radio"/>	<input checked="" type="radio"/>	<input type="checkbox"/>					
63 Channel													
64 Tube Sheets													
NOTES													
Issue No.	1 Date	2 Date	3 Date	4 Date	5 Date								
Made/Revised by													
Checked by													
Approved- Process													
Approved by													





**LINE LIST SCHEDULE**



**LINE SCHEDULE**  
**TITLE:**

**Job No.:**



**LINE SCHEDULE SHEET**



## LINE SUMMARY TABLE

**TITLE:**



## MASS BALANCE

Job No.:	Sheet of	Issue No.:	1	Date	2	Date	3	Date					
Job Name:		Description											
Document No.		Made/Revised by											
Project No.		Checked by											
Client		Approved Process											
Location		Approved											
Plant													
STREAM NUMBER		▼	▼	▼	▼	▼	▼	▼					
STREAM NAME													
PHASE													
TEMPERATURE (°C)													
COMPONENT	MOL WT	kg mol/h	Mol %	kg mol/h	Mol %	kg mol/h	Mol %	kg mol/h	Mol %	kg mol/h	Mol %	kg mol/h	Mol %
1 Water													
2 Hydrogen													
3 Nitrogen													
4 Carbon monoxide													
5 Carbon dioxide													
6 Methane													
7 Acetylene													
8 Ethylene													
9 Ethane													
10 Propyne													
11 Propadiene													
12 Propylene													
13 Propane													
14 Isobutane													
15 Isobutene													
16 <i>n</i> -Butane													
17 1-Butene													
18 1,3-Butadiene													
19 1-Pentene													
20 neo-Pentane													
21 Isopentane													
22 <i>n</i> -Pentane													
23 Toluene													
24 <i>m</i> -Xylene													
25 Methanol													
26 MTBE													
27 TBA													
28 DIISO													
29 TOTAL (kg mol/h)													
30													
31													
32													
33													
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37													
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40													
41													
42 Notes:													
43													
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45													
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47													
48													



## MECHANICAL EQUIPMENT SCHEDULE

Notes: 1. Rotary Equipment Type Coding: A = Axial; C = Centrifugal; M = Metering; R = Reciprocating; S = Screw

2. All drives electric motor unless specified otherwise.



## **PIPE LINE LIST**

**TITLE:**

Note: 1. Flow Units: G=gpm; C=acf m; P=lb/h



PIPE LIST

**TITLE:**

---

NOTE: 1. FLOW UNITS: G=gpm; C=acf m; P=lb/h



## PIPING PROCESS CONDITIONS SUMMARY

		<b>PLATE HEAT EXCHANGER DATA SHEET</b>		Document No.			
				Sheet	of	Rev.	
Job				Item No.(s)			
Item Name				No. Working		Total No. Off	
1	PROCESS DESIGN	UNITS		FLUID 1	FLUID 2	FLUID 3	
2	Fluid Circulated	-	-				
3	Total Fluid Entering (Normal)	kg/h	lb/h				
4	Flow Margin	%	%				
5	Inlet Vapor and Gas (&MW)	kg/h	lb/h				
6	Inlet Liquid	kg/h	lb/h				
7	Fluid Vaporized/Condensed (&MW)	kg/h	lb/h				
8	Temperature In/Out	°C	°F				
9	Max. Pressure Drop	bar	psi				
10	Inlet Pressure (Operating)	bar g	psig				
11	Normal Heat Load	kW	Btu/h				
12	Heat Load Margin	%	%				
13	Fouling Resistance	m <sup>2</sup> °C/W	ft <sup>2</sup> °Fh/Btu				
14	FLUID PROPERTIES						
15	Specific Heat	kJ/kg °C	Btu/lb °F				
16	Thermal Conductivity	W/m °C	Btu/hr ft °F				
17	Density	kg/m <sup>3</sup>	lb/f <sup>3</sup>				
18	Viscosity	cP	lb/hr.ft				
19	Liquid Viscosity (at Temp.)	cP	lb/hr.ft				
20	Latent Heat	kJ/kg	Btu/lb				
21	ENGINEERING DESIGN						
22	Process Design Pressure	bar g	psig				
23	Process Design Temperature	°C	°F				
24	Corrosion Allowance (Header)	mm	in.				
25	Line N.B. In/Out	in	in.				
26	Minimum Flow Passage	mm	in.				
27	Plate Material						
28	Frame Material						
29	Gasket Material						
30	Design Code						
31	NOTES						
32							
33	1. Unless otherwise stated fluid properties are for mean fluid temperatures.						
34	2. Sour Service Yes / No						
35							
36							
37							
38							
39							
40							
41							
42							
43							
44							
45							
46							
47							
48							
49							
50							
51							
52							
53							
54							
		1	Date	2	Date	3	Date
Description							
Made/Revised by							
Checked by							
Approved Process							
Approved by							

Made		<b>PROCESS DATA SHEET</b>			Job No.																																				
Checked					Item No.																																				
Date					Sheet of																																				
		Job																																							
<b>CALCULATION OF PRESSURE DROP IN FIXED CATALYST BEDS</b>																																									
<b>REQUIRED DATA</b>																																									
VESSEL NO:																																									
SERVICE:																																									
CATALYST:	VOLUME REQUIRED _____ MANUFACTURER & NO. _____ DIMENSIONS _____ EQUIV. PARTICLE DIA. _____ $D_p$ _____ ft. SHAPE/SIZE FACTOR _____ $S_f$ _____			V	ft <sup>3</sup>																																				
GAS DATA:	FLOW RATE _____ $W$ _____ lb/h $W^2$ _____ (lb/h) <sup>2</sup> MIN. MOLECULAR WEIGHT _____ BED OUTLET PRESSURE _____ $P$ _____ psia MAX. BED TEMPERATURE _____ °C ABS. TEMPERATURE (°K = °C + 273) _____ T _____ K																																								
<b>CALCULATION</b>																																									
$k_1 = \frac{(T)(S_f)}{(M)(P)} = \text{_____} =$																																									
BED DIAMETER BED C.S.A. BED DEPTH SUPERFICIAL MASS FLOW MASS FLOW FACTOR $D_p G$ REYNOLDS NO. FACTOR $k_2$ CLEAN PRESSURE DROP/UNIT DEPTH CLEAN OVERALL PRESSURE DROP SAFETY/FOULING FACTOR FLOW SHEET PRESSURE DROP	SYMBOLS	UNITS	CALCULATION	CASE 1	CASE 2	CASE 3	CASE 4																																		
	D	ft	$0.7854D^2$																																						
	A	ft <sup>2</sup>	V/A																																						
	L	ft.	W/A																																						
	G	lb/h.ft <sup>2</sup>																																							
	$G_f$	-																																							
	$Re_f$	-																																							
	$k_1$	-	$k_1 Re_f$																																						
	$\Delta P_c$	psi/ft.	$k_2 G_f$																																						
	$\Delta P_c$	psi	$L \Delta P_c$																																						
f	-		1.25/																																						
$\Delta P$	psi	$f \Delta P_c$																																							
* IF VALUE OF $D_p G > 200$ (750 FOR RING CATALYSTS) PUT Ref = 1.0 IF VALUE OF $D_p G < 200$ (750 FOR RING CATALYSTS) CALCULATE $Re = D_p G / \mu$ (WHERE $\mu$ = DYNAMIC VISCOSITY OF GAS, lb/ft.hr)																																									
<table border="1" style="width: 100%; border-collapse: collapse;"> <tr> <th>Description</th> <th>1</th> <th>2</th> <th>3</th> <th>4</th> <th>5</th> <th>6</th> </tr> <tr> <td>Made/Revised by</td> <td></td> <td></td> <td></td> <td></td> <td></td> <td></td> </tr> <tr> <td>Checked by</td> <td></td> <td></td> <td></td> <td></td> <td></td> <td></td> </tr> <tr> <td>Approved Process</td> <td></td> <td></td> <td></td> <td></td> <td></td> <td></td> </tr> <tr> <td>Approved by</td> <td></td> <td></td> <td></td> <td></td> <td></td> <td></td> </tr> </table>							Description	1	2	3	4	5	6	Made/Revised by							Checked by							Approved Process							Approved by						
Description	1	2	3	4	5	6																																			
Made/Revised by																																									
Checked by																																									
Approved Process																																									
Approved by																																									

		PROCESS ENGINEERING JOB ANALYSIS SUMMARY		
Job Title				
Job No.	Charge No.	Date		
Based Upon Cost Estimated Dated or Actual Construction Cost				
Summary Prepared By			Information Dated	
Production Basis (lb/day, tonnes/day, lb/month)				
SERVICE REQUIREMENTS:		UNIT RATE	UNIT RATE/ PRODUCTION BASIS	
1	Steam (30 lb)	lb/h		
2	Steam (150 lb)	lb/h		
3	Steam (400 lb)	lb/h		
4	Steam ( lb)	lb/h		
5	Treated R.W.	gpm		
6	Untreated R.W.	gpm		
7	Fresh Water	gpm		
8	Sea Water	gpm		
9	Fuel Gas ( psi)	cfm (60° F and 1atm)		
10	Air ( psi)	cfm (60° F and 1atm)		
11	Power ( )			
12	Horsepower			
13	Condensate	lb/h		
14				
	RAW MATERIALS	UNIT RATE		
1	Chlorine			
2	Hydrogen ( %)			
3	Caustic ( %)			
4	Salt			
5	Sat. Brine			
6	Natural Gas			
7	Air			
8	Ethylene			
9				
10				
11				
	PRODUCTS AND BY- PRODUCTS	UNIT RATE		
1	Chlorine			
2	HCl ( %)			
3	Salt ( %)			
4	Caustic ( %)			
5	Ammonia ( %)			
6	H <sub>2</sub> SO <sub>4</sub> ( %)			
7	Gas ( )			
8				
9				
10				
11				

		PUMP CALCULATION SHEET			Document No.	
					Sheet	of
Job Item Name.					Item No. (s)	
		No. Working		Total No. off		
	UNITS		CASE I	CASE II	SKETCH OF PUMP HOOK-UP	
1						
2	Liquid Pumped					
3	Corrosion/Erosion					
4	Due To					
5	Operating Temp. ( $T$ )	$^{\circ}\text{C}$	$^{\circ}\text{F}$			
6	Specific Gravity at $T$					
7	Viscosity	cP	cP			
8	Vapor Pressure at $T$	bara	psia			
9	Normal mass Flowrate	kg/h	lb/h			
10	Normal Vol. Flowrate	$\text{m}^3/\text{h}$	gpm			
11	Min. Vol. Flowrate					
12	Design Vol. Flowrate	$\text{m}^3/\text{h}$	gpm			
13	SUCTION CONDITION					
14	Pressure at Equipment	barg	psig	+	+	
15	Static Head	bar	psi	+ / -		
16	Total – Lines 14 + 15	bar	psi	+	+	
17	Suction Line $\Delta P$	bar	psi	-	-	
18	Filter/Strainer $\Delta P$	bar	psi	-	-	
19						
20	Total Suction Pressure	barg	psig	+	+	
21	DISCHARGE CONDITION					
22	Pressure at Equipment	barg	psig	+	+	
23	Static Head	bar	psi	+ / -	+ / -	
24						
25	Exchanger $\Delta P$	bar	psi	+	+	
26						
27	Furnace $\Delta P$	bar	psi	+	+	
28	Orifice $\Delta P$	bar	psi	+	+	
29	Control Valve $\Delta P$	bar	psi	+	+	
30						
31	Line $\Delta P$	bar	psi	+	+	
32						
33	Total Discharge Press.	barg	psig	+	+	
34	Differential Pressure	bar	psi			
35	Differential Head	bar	psi			
36	NPSH					
37	Total Suction Pressure	bara	psia			
38	Vapor Pressure	bara	psia			
39	NPSH – Lines 37–38	bara	psia			
40	=	m	ft			
41	Safety Margin	m	ft			
42	NPSH – Lines 40–41	m	ft			
43	Hydraulic Power	kW	Hp			
44	Estimated Efficiency	%	%			
45	Estimated Abs. Power	kW	Hp			
46	Type of Pump					
47	Drive					
48						
49	Material – Casing					
50	– Impeller					
51	– Shaft					
52						
53	Sour Service	Yes/No				
54	HEAD	$m = 10.2 \times \text{bar}/\text{SG}$	$m = 10 \times \text{kg}/\text{cm}^2/\text{SG}$	$ft = 2.31 \times \text{psi}/\text{SG}$		
55	VOLUME	$\text{m}^3/\text{h} \times \text{SG} \times 1000 = \text{kg}/\text{h}$	$\text{igpm} \times \text{SG} \times 600 = \text{lb}/\text{h}$			
56	POWER	$\text{kW} = \text{m}^3/\text{h} \times \text{bar}/36.0$	$\text{kW} = \text{m}^3/\text{h} \times \text{kg}/\text{cm}^2/36.71$	$\text{Hp} = \text{igpm} \times \text{psi}/1427$		
57		1 Date	2 Date	3 Date	4 Date	5 Date
58	Description					
59	Made/Revised by					
60	Checked by					
61	Approved Process					
62	Approved					



## PUMP SCHEDULE

		RELIEF DEVICE PHILOSOPHY SHEET		DOCUMENT / ITEM REFERENCE				
		EQUIPMENT No.:						
		DATE:		SHEET No.:		OF		
CHECKED BY:		MADE BY:					1	
							2	
DESIGN CODES:		VESSELS	EXCHANGERS	LINES			3	
							4	
OTHER REQUIREMENTS							5	
							6	
BASIS FOR CALCULATION:							7	
							8	
SET PRESSURE, PSIG:		MAX. BACK PRESSURE:	(a) BEFORE RELIEVING			(b) WHILE RELIEVING		9
NORMAL CONDITIONS UNDER RELIEF DEVICE:					Calculated			10
STATE:	TEMPERATURE, °F:	PRESSURE, PSIG:	POSSIBLE CAUSE?	FLUID RELIEVED	RELIEF RATE, lb/h	ORIFICE AREA, in².	11	
							12	
HAZARDS CONSIDERED							13	
1. Outlets blocked							14	
2. Control Valve malfunction							15	
3. Machine trip/ overspeed/density change							16	
4. Exchanger tube rupture							17	
5. Power failure/ Voltage dip							18	
6. Instrument air failure							19	
7. Cooling failure							20	
8. Reflux failure							21	
9. Abnormal entry of volatile liquid							22	
10. Loss of liquid level							23	
11. Abnormal chemical reaction							24	
12. Boxed in thermal expansion							25	
13. External fire							26	
14. (specify)							27	
15. (specify)							28	
16. (specify)							29	
SELECTED DESIGN CASE:							30	
							31	
RELIEVED FLUID: STATE		DENSITY / MW:	TEMPERATURE:		Cp/Cv:		32	
COMPOSITION:						FLASHING:	33	
RELIEF RATE REQUIRED, lb/h:		ORIFICE SELECTED:	AREA, in²:		TYPE:		34	
ACTUAL CAPACITY, lb/h							35	
REMARKS/SKETCH							36	
							37	
							38	
							39	
							40	
							41	
							42	
							43	
							44	
							45	
Issue No:	1	Date	2	Date	3	Date	46	
Description							47	
Made/Revised by							48	
Checked by							49	
Approved- Process							50	
Approved							51	

 Description Made/Revised by Checked by Approved Process  Approved		TANK AND VESSEL AGITATOR DATA SHEET			Drawing No: Project No.: sheet ..... Of .....																	
		1	Date	2		Date	3	Date														
		Equipment No:			No. Off:	Associated Vessel/Tank* Item No:																
		1. Largest and smallest charge: 2. Components added during mixing: 3. Agitator operating while vessel is being filled or product withdrawn? Yes/No* 4. If continuous, throughput per hour: 5. Process duty: Mixing liquids/Dissolving/Suspensions/Emulsions/Gas absorption/Homogenisation* 6. Mixing effect: Violent/Medium/Moderate* 7. Working Pressure: 8. Working Temperature: 9. Special Remarks:																				
		<b>PROCESS DEPT. INFORMATION</b>  <b>MIXING</b>  10. Components % by weight: 11. Temperature during mixing: 12. Specific gravity of components: 13. Viscosity of mixing at mixing temperature: 14. Specific gravity of product at mixing temperature: 15. Size of solid particles: 16. Special Remarks:																				
					<b>IMPELLER</b>  17. Type of impeller: 18. Number of impellers on shaft: 19. Position of impellers: 20. Distance between shaft end & vessel: Bottom bearing: 21. Preferred impeller speed: Shaft diameter: 22. Type of drive: Direct/Vee belt/Fluid* 23. Type of seal: Vapor/Packed gland/Mechanical/Easy replacement* 24. Method of installation: Assembled in/Assembled out* of vessel 25. Entry position: 26. Materials of construction:  27. Absorbed HP/KW*: Installed HP/KW*: 28. Other information:																	
								<b>TANK OR VESSEL</b>  29. Dimensions: Capacity: 30. Coils, baffles, etc.: 31. Fixing agitator (beams, flanges, etc.): 32. If at atmospheric pressure: Closed/Open* Can stuffing box be greased? Yes/No* 33. Headroom available above agitator: 34. Other information:														
											<b>ELECT.</b>  35. Agitator installed: Indoors/Outdoors Motor enclosure: 36. Electrical specification: 37. Any other electrical information:											
														<b>GENERAL</b>  38. Motor to be included: Yes/No* 39. Motor Will/Will not* be sent to manufacturer for assembly and alignment. 40. Threads: Unified /* 41. All rotating parts must be strictly guarded to BS 1649/ASME 42. Fixing bolts supplied by: 43. Net weight including motor: 44. Witnessed run in air:  45. General notes:								
NOTE: * Indicates delete as necessary.																						

				TANK PROCESS DATA SHEET			Doc. No: Item No. Sheet of	
				Job:				
		Item Name:						
		NOTE: * indicates delete as necessary; ** indicates for other than Code reason						
5								
10								
15								
20								
25								
26	Shell Diameter (O.D. / I.D.):			Shell Length:			No. Required:	
27	Center Line: *Horizontal / Vertical							
28	Pressure:	g	Temperature:	°F / °C	Nozzles	Mark No.	Size	Number
29	Item Number				Inlet	C-		
30	Operating							
31	Design				Vapor Out	C-		
32	Emergency Vacuum Design: * Yes / No							
33	Material		Corr. Allowance		Liquid Out:	C-		
34	Shell							
35	Heads							
36	Liner				Thermocouple	R-		
37	Type of Heads				Pressure Gauge	R-		
38	Code:				Gauge Glass	R-		
39	Stress Relieve**:	Radiography**:			Level Control	R-		
40	Joint Efficiency:				Safety Valve	R-		
41	Density of Contents:		at	*°F / °C				
42	Weight Empty:		Weight Full:					
43	Is vessel subject to mechanical vibration? *Yes / No				Vent	C-		
44	Insulation:	Type: *Frost and Personnel Protection / Cold			Drain	C-		
45	*Yes / No	Anticondensation / Heat Conservation			Steam Out	C-		
46	REMARKS:							
47				Manhole	A-			
48								
49								
50								
51								
52								
53					min. Base Elev'n:		k-Skirt Length	
54					Material:			
	Issue No.	1	Date	2	Date	3	Date	4
	Made/Revised by							
	Checked by							
	Approved - Process							
	Approved by							





**TIE-IN SCHEDULE**

\* ELD=ENGINEERING LINE DRAWING

Made by:			<b>TOWER PROCESS DATA SHEET</b>				Document No.		
Date:			Job				Item No.(s)		
Checked by:			Item Name				Sheet of		
	Tower Name:								
NOTE: * indicates delete as necessary; ** indicates for other than code reason									
1			Top	Bottom					
2	Shell Diameters O.D. - I.D.								
3	No. of Trays								
4	Pressure	Operating							
5	*psig	Design							
6	Temperature	Operating							
7	*°F / °C	Design							
8	Material	Shell							
9		Trays							
10		Caps							
11		Liner or Clad							
12	Corrosion -	Shell							
13	Allowance	Heads							
14		Trays							
15	Tray Spacing								
16	Type of Liquid Flow								
17	Type of Trays								
18	Joint Efficiency								
19	Code	Emergency Vac. Design *Yes / No							
20	Stress Relieved **Yes / No	Radiography **Yes / No							
21	Is vessel subject to mechanical vibration	*Yes / No							
22	Insulation	Type: *Frost and Personnel Protection / Cold /							
23	*Yes / No	/anticondensation / Heat Conservation /							
24	Min. Base Elevation:			Skirt length:					
25	Weight Empty:			Full:					
26	Nozzles	Mark No.	Size	Number					
27	Feed	C-							
28									
29	Overhead Vpr.	C-							
30	Reflux In	C-							
31									
32	Bottoms	C-							
33	Reboiler Vpr	C-							
34	Reboiler liq	C-							
35									
36									
37	Thermocouple	R-	by inst. Gp.						
38	Level Glass	R-							
39	Press Gauge	R-							
40	Level Control	R-							
41	Safety Valve	R-							
42									
43									
44	Vent	C-							
45	Drain	C-							
46	Steam Out	C-							
47	Manholes	A-							
48	Handholes	A-							
49	Cap Type:			Tray Layout Ref:					
50	NOTES								
51									
52									
53									
54									
55		1	Date	2	Date	3	Date	4	Date
56	Description								
57	Made/Revised by								
58	Checked by								
59	Approved Process								
60	Approved by								

		TRAY LOADING SUMMARY				Job No.	
		Job:					
						item No.	
				Sheet of			
	Tower Name:	Type of Tray					
1	Manufacturer					Mfr. Ref:	
2	Pressure at top of Tower	barg	psia				
3	Max. ΔP over Tower	bar	psi	units Used: METRIC/BRITISH (delete one)			
4	No. of Trays	Aboved Feed				Below Feed	
5	Tray location						
6	Tray Number*						
7	Tray Spacing	mm	in.				
8	Tower Internal Diameter I.D.	mm	in.				
9	Vapor to Tray						
10	Temperature	°C	°F				
11	Compressibility						
12	Density	kg/m <sup>3</sup>	lb/ft <sup>3</sup>				
13	Molecular Weight						
14	Rate	kg/h	lb/h				
15							
16	Liquid from Tray						
17	Temperature	°C	°F				
18	Surface Tension	dynes/cm	lb/ft				
19	Viscosity	cps	lb ft/hr				
20	Density	kg/m <sup>3</sup>	lb/ft <sup>3</sup>				
21	Rate	kg/h	lb/h				
22	Foaming Tendency **						
23	Number of Passes						
24	Minimum Hole Diameter ***	mm	in.				
25	Minimum DC Residence Time***	secs	s				
26	Maximum Rate as % Design						
27	Minimum Rate as % Design						
28	Design Rate % Flood Rate***						
29							
30	Tray Material						
31	Valve or Cap Material						
32	Corrosion Allowance	mm	in.				
33	Tray Thickness***	mm	in.				
34							
35							
36							
37							
38	NOTES:	* Trays are numbered from the bottom of the tower upwardstop of the tower downwards					
39		** Indicate whether 'non', 'moderate', 'high' or 'severe'					
40		*** Data to be supplied by tray manufacturer unless special Process/Client requirement entered here.					
41	REMARKS:						
42							
43							
44							
45							
46							
47							
48							
49							
50							
51							
52							
53							
54							
55		1	Date	2	Date	3	Date
56	Description						
57	Made/Revised by						
58	Checked by						
59	Approve - Process						
60	Approved by						



TRIP SCHEDULE



**UTILITY SUMMARY SHEET**

Note 1: State units -  $P$  is

Note 2: Cooling water Source I is:

T

Source II is



## VESSEL AND TANK SCHEDULE

**Figure 1-38** Vessel and tank summary



**VESSEL SCHEDULE**

<b><math>\Sigma \Omega \Sigma</math></b> A.K.C. TECHNOLOGY		<b>WATER ANALYSIS SHEET</b>	Submitted by: Address:		Location:			
					Date:			
**	Analysis		No.					
			Date:					
	pH Value							
	Suspended Solids		mg/l					
	Total Dissolved Solids		mg/l at 110°C					
	Total Dissolved Solids		mg/l at 180°C					
	Alkalinity to Pp		as mg/l CaCO <sub>3</sub>					
	Alkalinity to MO		as mg/l CaCO <sub>3</sub>					
	Sulphate		as mg/l SO <sub>4</sub>					
	Chloride		as mg/l Cl					
	Nitrate		as mg/l NO <sub>3</sub>					
	Silica		as mg/l SiO <sub>2</sub>					
	Phosphate		as mg/l PO <sub>4</sub>					
	Total Anions		as mg/l CaCO <sub>3</sub>					
	Total Hardness		as mg/l CaCO <sub>3</sub>					
	Calcium		as mg/l Ca					
	Magnesium		as mg/l Mg					
	Sodium*		as mg/l Na					
	Potassium*		as mg/l K					
	Iron		as mg/l Fe					
	Manganese		as mg/l Mn					
	Free and Saline Ammonia		as mg/l NH <sub>3</sub>					
	Total Cations		as mg/l CaCO <sub>3</sub>					
	Free Dissolved CO <sub>2</sub>		as mg/l CO <sub>2</sub>					
	Dissolved O <sub>2</sub>		as mg/l O					
	Color Hazaen Units							
	Turbidity Formazin Units <sup>+</sup>		FTU APHA					
	Lead		as mg/l Pb					
	Copper		as mg/l Cu					
	Residual Chlorine							
Flouride		as mg/l F						
Sulfite		as mg/l SO <sub>3</sub>						
<p>NOTES:</p> <ol style="list-style-type: none"> <li>1. Please state if units other than milligrams per liter, or different conditions of test, are used.</li> <li>2. * If Na and/or K form an appreciable amount of total cations, please state Alkalinity to Phenolphthalein (p), Alkalinity to Methyl Orange (MO), and Carbonate Hardness.</li> <li>3. + DO NOT use Formazin Units as defined by British Standard BS. 2690 Pt. 9.</li> <li>4. ** Tick if item definitely to be included in analysis.</li> <li>5. 100 mg/L CaCO<sub>3</sub> = 2 milliequivalents/(m val/L).</li> </ol>								

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# Appendix B

## ETHICS IN ENGINEERING PROFESSION

Ethical issues in the engineering profession are becoming more prevalent as companies are competing for the same market with greater demands on engineers or alternatively with increasing demands by the public on the environment (e.g. Three Mile Island, Bhopal, etc.). Appendix B presents ethical, moral, and legal obligations, codes of conduct by engineering institutions. It also presents heuristics (i.e. ways of assisting to discover what is best) and identification strategies that are helpful in solving ethical problems. In the final analysis all heuristics are fallible; however, they provide guidelines toward the solution of a problem as their characteristics are as follows [1]:

- A heuristic does not guarantee a solution.
- It may contradict other heuristics.
- It can reduce the time to solve a problem.
- Its acceptance depends on the immediate context instead of on an absolute standard.

## American Society of Mechanical Engineers (ASME)

### Ethics

ASME requires ethical practice by each of its members and has adopted the following Code of Ethics of Engineers as referenced in the ASME Constitution, Article C2.1.1.

### Code of Ethics of Engineers

#### The Fundamental Principles

Engineers uphold and advance the integrity, honor and dignity of the engineering profession by:

- I. Using their knowledge and skill for the enhancement of human welfare;
- II. Being honest and impartial, and serving with fidelity the public, their employers and clients; and
- III. Striving to increase the competence and prestige of the engineering profession.

#### The Fundamental Canons

1. Engineers shall hold paramount the safety, health and welfare of the public in the performance of their professional duties.
2. Engineers shall perform services only in the areas of their competence.
3. Engineers shall continue their professional development throughout their careers and shall provide opportunities for the professional and ethical development of those engineers under their supervision.
4. Engineers shall act in professional matters for each employer or client as faithful agents or trustees, and shall avoid conflicts of interest or the appearance of conflicts of interest.
5. Engineers shall build their professional reputation on the merit of their services and shall not compete unfairly with others.
6. Engineers shall associate only with reputable persons or organizations.
7. Engineers shall issue public statements only in an objective and truthful manner.
8. Engineers shall consider environmental impact in the performance of their professional duties.

### The ASME Criteria for the Interpretation of the Canons

The ASME criteria for interpretation of the Canons are guidelines and represent the objectives toward which members of the engineering profession should strive. They are principles, which an engineer can reference in specific situations. In addition, they provide interpretive guidance to the ASME Board on Professional Practice and Ethics on the Code of Ethics of Engineers.

1. Engineers shall hold paramount the safety, health and welfare of the public in the performance of their professional duties.
  - a. Engineers shall recognize that the lives, safety, health and welfare of the general public are dependent upon engineering judgments, decisions and practices incorporated into structures, machines, products, processes and devices.
  - b. Engineers shall not approve or seal plans and/or specifications that are not of a design safe to the public health and welfare and in conformity with accepted engineering standards.
  - c. Whenever the Engineers' professional judgments are overruled under circumstances where the safety, health, and welfare of the public are endangered, the Engineers shall inform their clients and/or employers of the possible consequences.
    - (i) Engineers shall endeavor to provide data such as published standards, test codes, and quality control procedures that will enable the users to understand safe use during life expectancy associated with the designs, products, or systems for which they are responsible.
    - (ii) Engineers shall conduct reviews of the safety and reliability of the designs, products, or systems for which they are responsible before giving their approval to the plans for the design.
    - (iii) Whenever Engineers observe conditions, directly related to their employment, which they believe will endanger public safety or health, they shall inform the proper authority of the situation.
  - d. If Engineers have knowledge of or reason to believe that another person or firm may be in violation of any of the provisions of these Canons, they shall present such information to the proper authority in writing and shall cooperate with the proper authority in furnishing such further information or assistance as may be required.
2. Engineers shall perform services only in areas of their competence.
  - a. Engineers shall undertake to perform engineering assignments only when qualified by education and/or experience in the specific technical field of engineering involved.
  - b. Engineers may accept an assignment requiring education and/or experience outside of their own fields of competence, but their services shall be restricted to other phases of the project in which they are qualified. All other phases of such project shall be performed by qualified associates, consultants, or employees.
3. Engineers shall continue their professional development throughout their careers, and should provide opportunities for the professional and ethical development of those engineers under their supervision.

4. Engineers shall act in professional matters for each employer or client as faithful agents or trustees, and shall avoid conflicts of interest or the appearance of conflicts of interest.
- a. Engineers shall avoid all known conflicts of interest with their employers or clients and shall promptly inform their employers or clients of any business association, interests, or circumstances which could influence their judgment or the quality of their services.
  - b. Engineers shall not undertake any assignments which would knowingly create a potential conflict of interest between themselves and their clients or their employers.
  - c. Engineers shall not accept compensation, financial or otherwise, from more than one party for services on the same project, or for services pertaining to the same project, unless the circumstances are fully disclosed to, and agreed to, by all interested parties.
  - d. Engineers shall not solicit or accept financial or other valuable considerations, for specifying products or material or equipment suppliers, without disclosure to their clients or employers.
  - e. Engineers shall not solicit or accept gratuities, directly or indirectly, from contractors, their agents, or other parties dealing with their clients or employers in connection with work for which they are responsible. Where official public policy or employers' policies tolerate acceptance of modest gratuities or gifts, engineers shall avoid a conflict of interest by complying with appropriate policies and shall avoid the appearance of a conflict of interest.
  - f. When in public service as members, advisors, or employees of a governmental body or department, Engineers shall not participate in considerations or actions with respect to services provided by them or their organization(s) in private or product engineering practice.
  - g. Engineers shall not solicit an engineering contract from a governmental body or other entity on which a principal, officer, or employee of their organization serves as a member without disclosing their relationship and removing themselves from any activity of the body which concerns their organization.
  - h. Engineers working on codes, standards or governmental sanctioned rules and specifications shall exercise careful judgment in their determinations to ensure a balanced viewpoint, and avoid a conflict of interest.
  - i. When, as a result of their studies, Engineers believe a project(s) will not be successful, they shall so advise their employer or client.
  - j. Engineers shall treat information coming to them in the course of their assignments as confidential, and shall not use such information as a means of making personal profit if such action is adverse to the interests of their clients, their employers or the public.
    - (i) They will not disclose confidential information concerning the business affairs or technical processes of any present or former employer or client or bidder under evaluation, without his consent, unless required by law or court order.
    - (ii) They shall not reveal confidential information or finding of any commission or board of which they are members unless required by law or court order.
    - (iii) Designs supplied to Engineers by clients shall not be duplicated by the Engineers for others without the express permission of the client(s).
  - k. Engineers shall act with fairness and justice to all parties when administering a construction (or other) contract.
  - l. Before undertaking work for others in which Engineers may make improvements, plans, designs, inventions, or other records which may justify seeking copyrights, patents, or proprietary rights, Engineers shall enter into positive agreements regarding the rights of respective parties.
5. Engineers shall build their professional reputation on the merit of their services and shall not compete unfairly with others.
- a. Engineers shall negotiate contracts for professional services on the basis of demonstrated competence and qualifications for the type of professional service required.
  - b. Engineers shall not request, propose, or accept professional commissions on a contingent basis if, under the circumstances, their professional judgments may be compromised.
  - c. Engineers shall not falsify or permit misrepresentation of their, or their associates, academic or professional qualification. They shall not misrepresent or exaggerate their degrees or responsibility in or for the subject matter of prior assignments. Brochures or other presentations used to solicit personal employment shall not misrepresent pertinent facts concerning employers, employees, associates, joint ventures, or their accomplishments.
  - d. Engineers shall prepare articles for the lay or technical press which are only factual.
    - (i) Technical Communications for publication (theses, articles, papers, reports, etc) which are based on research involving more than one individual (including students and supervising faculty, industrial supervisor/researcher or other co-workers) must recognize all significant contributors. Co authors listed on proposed and accepted publications should have entered the joint authorship arrangement by mutual consent prior to submittal of the document for publication and should have received written permission to use any unpublished work of others which serves as the major basis or key component of the publication.
    - (ii) Technical Communications should adhere to clearly defined and appropriately disseminated guidelines on authorship. These guidelines should be promulgated and publicized in corporate, university or other employer policies and should take cognizance of professional technical society recommendations on ethical practice.
    - (iii) Plagiarism, the act of substantially using another's ideas or written materials without due credit, is unethical.
  - e. Engineers shall not maliciously or falsely, directly or indirectly, injure the professional reputation, prospects, practice or employment of another engineer, nor shall they indiscriminately criticize another's work.
  - f. Engineers shall not use equipment, supplies, laboratory or office facilities of their employers to carry on outside private practice without consent.
  - 6. Engineers shall associate only with reputable persons or organizations.
    - a. Engineers shall not knowingly associate with or permit the use of their names or firm names in business ventures by any person or firm which they know, or have reason to believe, are engaging in business or professional practices of a fraudulent or dishonest nature.

- b.** Engineers shall not use association with non-engineers, corporations, or partnerships to disguise unethical acts.
- 7. Engineers shall issue public statements only in an objective and truthful manner.
  - a. Engineers shall endeavor to extend public knowledge, and to prevent misunderstandings of the achievements of engineering.
  - b. Engineers shall be completely objective and truthful in all professional reports, statements or testimony. They shall include all relevant and pertinent information in such reports, statements or testimony.
  - c. Engineers, when serving as expert or technical witnesses before any court, commission, or other tribunal, shall express an engineering opinion only when it is founded on their adequate knowledge of the facts in issue, their background of technical competence in the subject matter, and their belief in the accuracy and propriety of their testimony.
  - d. Engineers shall issue no statements, criticisms, or arguments on engineering matters which are inspired or paid for by an interested party, or parties, unless they preface their comments by identifying themselves, by disclosing the identities of the party or parties on whose behalf they are speaking, and by revealing the existence of any financial interest they may have in matters under discussion.
  - e. Engineers shall be truthful in explaining their work and merit, and shall avoid any act tending to promote their own interest at the expense of the integrity and honor of the profession or another individual.
- 8. Engineers shall consider environmental impact in the performance of their professional duties.
  - a. Engineers shall concern themselves with the impact of their plans and designs on the environment. When the impact is a clear threat to health or safety of the public, then the guidelines for the Canon revert to those of Canon 1.
- 9. Engineers accepting membership in The American Society of Mechanical Engineers by this action agree to abide by this Society Policy on Ethics and procedures for its implementation.

Responsibility: Council on Member Affairs/Board on Professional Practice and Ethics

Adopted: March 7, 1976  
 Revised: December 9, 1976  
           December 7, 1979  
           November 19, 1982  
           June 15, 1984  
           (editorial changes 7/84)  
           June 16, 1988  
           September 12, 1991  
           September 11, 1994  
           June 10, 1998  
           September 21, 2002

#### American Institute of Chemical Engineers (AIChE)

##### **Code of Ethics** (Revised January 17, 2003)

Members of the American Institute of Chemical Engineers shall uphold and advance the integrity, honor and dignity of the Engineering profession by: being honest and impartial and serving with fidelity their employers, their clients, and the public; striving to increase the competence and prestige of the engineering profession; and using their knowledge and skill for the enhancement of human welfare. To achieve these goals, members shall

1. Hold paramount the safety, health and welfare of the public and protect the environment in performance of their professional duties.

- 2. Formally advise their employers or clients (and consider further disclosure, if warranted) if they perceive that a consequence of their duties will adversely affect the present or future health or safety of their colleagues or the public.
- 3. Accept responsibility for their actions, seek and heed critical review of their work and offer objective criticism of the work of others.
- 4. Issue statements or present information only in an objective and truthful manner.
- 5. Act in professional matters for each employer or client as faithful agents or trustees, avoiding conflicts of interest and never breaching confidentiality.
- 6. Treat fairly and respectfully all colleagues and co-workers, recognizing their unique contributions and capabilities.
- 7. Perform professional services only in areas of their competence.
- 8. Build their professional development throughout their careers, and provide opportunities for the professional development of those under their supervision.
- 9. Never tolerate harassment.  
        Conduct themselves in a fair, honorable and respectful manner.

#### National Society of Professional Engineers

##### **Code of Ethics for Engineers**

###### **Preamble**

Engineering is an important and learned profession. As members of this profession, engineers are expected to exhibit the highest standards of honesty and integrity. Engineering has a direct and vital impact on the quality of life for all people. Accordingly, the services provided by engineers require honest, impartiality, fairness and equity, and must be dedicated to the protection of the public health, safety, and welfare. Engineers must perform under a standard of professional behavior that requires adherence to the highest principles of ethical conduct.

###### **I. Fundamental Canons**

Engineers, in the fulfillment of their professional duties, shall:

- 1. Hold paramount the safety, health and welfare of the public.
- 2. Perform services only in areas of their competence.
- 3. Issue public statements only in an objective and truthful manner.
- 4. Act for each employer or client as faithful agents or trustees.
- 5. Avoid deceptive acts.
- 6. Conduct themselves honorably, responsibly, ethically, and lawfully so as to enhance the honor, reputation, and usefulness of the profession.

###### **II. Rules of Practice**

- 1. Engineers shall hold paramount the safety, health, and welfare of the public.
  - a. If engineers' judgment is overruled under circumstances that endanger life or property, they shall notify their employer or client and such other authority as may be appropriate.
  - b. Engineers shall approve only those engineering documents that are in conformity with applicable standards.
  - c. Engineers shall not reveal facts, data or information without the prior consent of the client or employer except as authorized or required by law or this Code.
  - d. Engineers shall not permit the use of their names or associate in business ventures with any person or firm that they believe is engaged in fraudulent or dishonest enterprise.

- e. Engineers shall not aid or abet the unlawful practice of engineering by a person or firm.
- f. Engineers having knowledge of any alleged violation of this Code shall report thereon to appropriate professional bodies and, when relevant, also to public authorities, and cooperate with the proper authorities in furnishing such information or assistance as may be required.
- 2. Engineers shall perform services only in the areas of their competence.
  - a. Engineers shall undertake assignments only when qualified by education or experience in the specific technical fields involved.
  - b. Engineers shall not affix their signatures to any plans or documents dealing with subject matter in which they lack competence, nor to any plan or document not prepared under their direction and control.
  - c. Engineers may accept assignments and assume responsibility for coordination of an entire project, provided that each technical segment is signed and sealed only by the qualified engineers who prepared the segment.
- 3. Engineers shall issue public statements only in an objective and truthful manner.
  - a. Engineers shall be objective and truthful in professional reports, statements, or testimony. They shall include all relevant and pertinent information in such reports, statements, or testimony, which should bear the date indicating when it was current.
  - b. Engineers may express publicly technical opinions that are founded upon knowledge of the facts and competence in the subject matter.
  - c. Engineers shall issue no statements, criticisms, or arguments on technical matters that are inspired or paid for by interested parties, unless they have prefaced their comments by explicitly identifying the interested parties on whose behalf they are speaking, and by revealing the existence of any interest the engineers may have in the matters.
- 4. Engineers shall act for each employer or client as faithful agents or trustees.
  - a. Engineers shall disclose all known or potential conflicts of interest that could influence or appear to influence their judgment or the quality of their services.
  - b. Engineers shall not accept compensation, financial or otherwise, from more than one party for services on the same project, or for services pertaining to the same project, unless the circumstances are fully disclosed and agreed to by all interested parties.
  - c. Engineers shall not solicit or accept financial or other valuable consideration, directly or indirectly, from outside agents in connection with the work for which they are responsible.
  - d. Engineers in public service as members, advisors, or employees of a governmental or quasi-governmental body or department shall not participate in decisions with respect to services solicited or provided by them or their organizations in private or public engineering practice.
  - e. Engineers shall not solicit or accept a contract from a governmental body on which a principal or officer of their organization serves as a member.
- 5. Engineers shall avoid deceptive acts.
  - a. Engineers shall not falsify their qualifications or permit misrepresentation of their or their associates' qualification. They shall not misrepresent or exaggerate their responsibility in or for the subject matter of prior assignments. Brochures or other presentations incident to the

solicitation of employment shall not misrepresent pertinent facts concerning employers, employees, associates, joint ventures, or past accomplishments.

- b. Engineers shall not offer, give, solicit, or receive, either directly or indirectly, any contribution to influence the award of a contract by public authority, or which may be reasonably construed by the public as having the effect or intent of influencing the awarding of a contract. They shall not offer any gift or other valuable consideration in order to secure work. They shall not pay a commission, percentage, or brokerage fee in order to secure work, except to a bona fide employee or bona fide established commercial or marketing agencies retained by them.

### III. Professional Obligations

- 1. Engineers shall be guided in all their relations by the highest standards of honesty and integrity.
  - a. Engineers shall acknowledge their errors and shall not distort or alter the facts.
  - b. Engineers shall advise their clients or employers when they believe a project will not be successful.
  - c. Engineers shall not accept outside employment to the detriment of their regular work or interest. Before accepting any outside engineering employment, they will notify their employers.
  - d. Engineers shall not attempt to attract an engineer from another employer by false or misleading pretenses.
  - e. Engineers shall not promote their own interest at the expense of the dignity and integrity of the profession.
- 2. Engineers shall at all times strive to serve the public interest.
  - a. Engineers shall seek opportunities to participate in civic affairs; career guidance for youths; and work for the advancement of the safety, health, and well-being of their community.
  - b. Engineers shall not complete, sign, or seal plans and/or specifications that are not in conformity with applicable engineering standards. If the client or employer insists on such unprofessional conduct, they shall notify the proper authorities and withdraw from further service on the project.
  - c. Engineers shall endeavor to extend public knowledge and appreciation of engineering and its achievements.
- 3. Engineers shall avoid all conduct or practice that deceives the public.
  - a. Engineers shall avoid the use of statements containing a material misrepresentation of fact or omitting a material fact.
  - b. Consistent with the foregoing, engineers may advertise for recruitment of personnel.
  - c. Consistent with the foregoing, engineers may prepare articles for the lay or technical press, but such articles shall not imply credit to the author for work performed by others.
- 4. Engineers shall not disclose, without consent, confidential information concerning the business affairs or technical processes of any present or former client or employer, or public body on which they serve.
  - a. Engineers shall not, without the consent of all interested parties, promote or arrange for new employment or practice in connection with a specific project for which the engineer has gained particular and specialized knowledge.
  - b. Engineers shall not without the consent of all interested parties, participate in or represent an adversary interest in connection with a specific project or proceeding in which the engineer has gained particular specialized knowledge on behalf of a former client or employer.

5. Engineers shall not be influenced in their professional duties by conflicting interests.
  - a. Engineers shall not accept financial or other considerations, including free engineering designs, from material or equipment suppliers for specifying their product.
  - b. Engineers shall not accept commissions or allowances, directly or indirectly, from contractors or other parties dealing with clients or employers of the engineer in connection with work for which the engineer is responsible.
6. Engineers shall not attempt to obtain employment or advancement or professional engagements by untruthfully criticizing other engineers, or by other improper or questionable methods.
  - a. Engineers in salaried positions shall accept part-time engineering work only to the extent consistent with policies of the employer and in accordance with ethical considerations.
  - b. Engineers shall not, without consent, use equipment, supplies, laboratory, or office facilities of an employer to carry on outside private practice.
7. Engineers shall not attempt to injure, maliciously or falsely, directly or indirectly, the professional reputation, prospects, practice, or employment of other engineers. Engineers who believe others are guilty of unethical or illegal practice shall present such information to the proper authority for action.
  - a. Engineers in private practice shall not review the work of another engineer for the same client, except with the knowledge of such engineer, or unless the connection of such engineer with the work has been terminated.
  - b. Engineers in government, industrial, or educational employ are entitled to review and evaluate the work of other engineers when so required by their employment duties.
  - c. Engineers in sales or industrial employ are entitled to make engineering comparisons of represented products with products of other suppliers.
8. Engineers shall accept personal responsibility for their professional activities, provided, however, that engineers may seek indemnification for services arising out of their practice for other than gross negligence, where the engineer's interests cannot otherwise be protected.
  - a. Engineers shall conform with state registration laws in the practice of engineering.
  - b. Engineers shall not use association with nonengineer, a corporation, or partnership as a "cloak" for unethical acts.
9. Engineers shall give credit for engineering work to those to whom credit is due, and will recognize the proprietary interests of others.
  - a. Engineers shall, whenever possible, name the person or persons who may be individually responsible for designs, inventions, writings, or other accomplishments.
  - b. Engineers using designs supplied by a client recognize that the designs remain the property of the client and may not be duplicated by the engineer for others without express permission.
  - c. Engineers, before undertaking work for others in connection with which the engineer may make improvements, plans, designs, inventions, or other records that may justify copyrights or patents, should enter into a positive agreement regarding ownership.
  - d. Engineers' design, data, records, and notes referring exclusively to an employer's work are the employer's property. The employer should indemnify the engineer for use of the information for any purpose other than the original purpose.

- e. Engineers shall continue their professional development throughout their careers and should keep current in their specialty fields by engaging in professional practice, participating in continuing education course, reading in the technical literature, and attending professional meetings and seminars.

As revised January 2003

"By order of the United States District Court for the District of Columbia, former Section 11 (c) of the NSPE Code of Ethics prohibiting competitive bidding, and all policy statements, opinions, rulings or other guidelines interpreting its scope, have been rescinded as unlawfully interfering with the legal right of engineers, protected under the antitrust laws, to provide price information to prospective clients; accordingly, nothing contained in the NSPE Code of Ethics, policy statements, opinions, rulings or other guidelines prohibits the submission of price quotations or competitive bids for engineering services at any time or in any amount."

#### **Statement by NSPE Executive Committee**

In order to correct misunderstandings which have been indicated in some instances since the issuances of the Supreme Court decision and the entry of the Final Judgment, it is noted that in its decision of April 25, 1978, the Supreme Court of the United States declared: "The Sherman Act does not require competitive bidding."

It is further noted that as made clear in the Supreme Court decision:

1. Engineers and firms may individually refuse to bid for engineering services.
2. Clients are not required to seek bids for engineering services.
3. Federal, state, and local laws governing procedures to procure engineering services are not affected, and remain in full force and effect.
4. State societies and local chapters are free to actively and aggressively seek legislation for professional selection and negotiation procedures by public agencies.
5. State registration board rules of professional conduct, including rules prohibiting competitive bidding for engineering services, are not affected and remain in full force and effect. State registration boards with authority to adopt rules of professional conduct may adopt rules governing procedures to obtain engineering services.
6. As noted by the Supreme Court, "nothing in the judgment prevents NSPE and its members for attempting to influence governmental action . . ."

#### **Institution of Chemical Engineers (IChemE)**

##### **Rules of professional conduct**

Issue III: 7th December 2001

The Rules and Regulations embodied in this document are made and published as required by By-law 13 of the By-laws of The Institution of Chemical Engineers. They must be read and interpreted in accordance with By-law 1 of that Institution.

#### **RULES OF PROFESSIONAL CONDUCT**

1. A code of professional conduct and related Rules designed to cover broad ethical principles must necessarily be drawn up in general terms. These Rules, published by the Council, indicate the manner in which members are required by the Council to conduct themselves in most situations. This conduct must be of

a high standard: members must discharge their duties competently, with reasonable skill, and diligence, and shall comply with the standards of behavior, integrity, competence and professional judgement which other members might reasonably expect of them.

For situations not specifically encompassed by these Rules, members are required to order their conduct in accordance with the principle that, in any conflict between a member's personal interests and the fair and proper interests of the community, the member's duty to the community should prevail.

In order to fulfil their duty under these Rules, members shall give due attention to any Guidance Notes that may be published from time to time by the Council in accordance with Rule 2 below.

2. The Council may from time to time publish Guidance Notes to members containing advice as to specific conduct which is to be regarded as proper or improper as the case may be.
3. Members when discharging their professional duties shall act with integrity, in the public interest, and to exercise all reasonable professional skill and care to:
  - a. Prevent avoidable danger to health or safety.
  - b. Prevent avoidable adverse impact on the environment.
  - c. (i) Maintain their competence.  
 (ii) Undertake only professional tasks for which they are competent.  
 (iii) Disclose relevant limitations of competence.
  - d. (i) Accept appropriate responsibility for work carried out under their supervision.  
 (ii) Treat subordinates fairly and without bias.  
 (iii) Encourage others to advance their learning and competence.
  - e. (i) Avoid where possible real or perceived conflict of interests.  
 (ii) Advise affected parties when such conflicts arise.
  - f. Observe the proper duties of confidentiality owed to appropriate parties.
  - g. Reject bribery and any other corrupt practices.
  - h. Assess relevant risks and liability, and, if appropriate, hold professional indemnity insurance.
  - i. Notify the Institution if convicted of a criminal offence or upon becoming bankrupt or disqualified as a Company Director.
  - j. Notify the Institution of any significant violation of the Institution's Code of Conduct by another member.
  - k. Be mindful at all times of the dignity of the profession in their personal conduct.
4. Members may be deemed guilty of improper professional conduct if:
  - a. in the opinion of the Council their conduct fails to conform with the By-laws of the Institution (including the Code of Conduct contained in By-law 12(b)) or of these Rules or with any advice published thereunder; or
  - b. they are convicted by a competent tribunal of any criminal offence, or have in any other proceedings to which they are party been found by a Court or other competent tribunal guilty of conduct which (in either such case) in the opinion of the Council renders them unfit to be members; or
  - c. they are, in the opinion of the Council, precluded from performing their professional duties, in a manner which is consistent with the standards of their profession, as a result of having been adjudicated bankrupt or of making a composition with their creditors; or
  - d. in the opinion of the Council their conduct in other respects renders them unfit to be members of the Institution or is injurious to the Institution.

The following are heuristics that can be used effectively in engineering ethics [2]:

1. Use a traditional problem-solving strategy for solving ethical problems.
2. Consider the possibility that inexperienced people can be right.
3. Debrief people fully before assuming facts about their actions.
4. Consider what will happen if a specific decision is based on a false assumption.
5. Be honest and open. This is especially important when dealing with those who are predisposed to distrust you.
6. Be concerned about the welfare of your company.
7. Be concerned about the welfare of your employees.
8. Do not let people make ethical decisions for you.
9. Acquire all the information you can about the situation. The problem may be more serious than it first appears, or there may be no real problem at all.
10. Acknowledge the concerns of others, whether you share their concerns or not.
11. Remember that one is only as ethical as one can afford to be. One can enhance one's ability to afford to do the right thing by knowing the consequences and by balancing the responsibilities that one accepts with the rest of one's life.

Woods proposed the McMaster five-step strategy [3] in analyzing standard problem solving strategy [4] as follows:

**Define.** Was the problem well defined? Was the real problem that now things were more uncertain than before? There are other possible definitions of the problem, but, if one defines the problem as how to deal with obviously flawed data, one may miss the entire point. The problem would be thereby unnecessarily over-constrained. What is the real problem? If the characters had defined the problem better, would they reach a different conclusion? Would it have been a better conclusion?

**Explore.** Explore the "real problem" by creating a rich perspective of the problem. During the Explore stage, see it from many different view points. Be willing to spend at least half the total available time defining the problem. Ask many "what if?" questions. Identify the real problem, by asking a series of "Why?" questions to generalize the situation and to see the problem in the context of a "system."

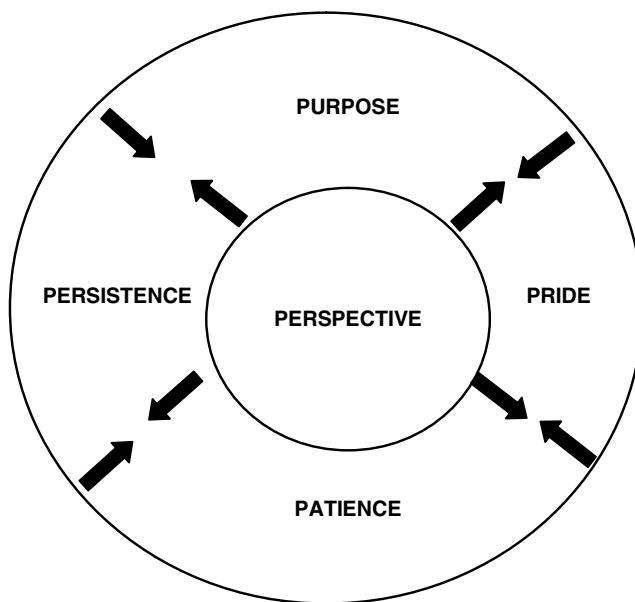
**Plan.** Use the criteria to select a sequence of diagnostic actions. Sometimes several actions can be combined. However, usually it is best to wait for the results from the first action before we recycle back to the Explore stage and relook our hypotheses. The criteria in selecting an action include the following: Will the action provide background information to ensure the problem is understood in context or is the action to test a hypothesis? Will the action produce results that give the accuracy needed? Is the action simple? inexpensive? safe?

**Do It.** Carry out the first action in the plan. Check. Monitor.

**Look back.** Compare the results obtained with the hypotheses. Look back at the process used. Self-assess. Return to previous stages of *Engage, Explore, Plan* and continue.

Blanchard and Peale [5] proposed a set of guidelines that helps to sort out the issues on ethical solution and to consider the following checklist of questions:

1. Is it legal? Will I be violating either civil law or company policy?
2. Is it balanced? Is it fair to all concerned in the short term as well as the long term? Does it promote win-win relationships?
3. How will it make me feel about myself? Will it make me proud? Would I feel good if my decisions were published in the newspaper? Would I feel good if my family knew about it?



**Figure B-1** The interrelationships among the five P's [4].

If the answer to the first question could be interpreted as "NO, it is not legal," then there is no need to proceed to the second and third Ethics Check list questions. However, if the solution is indeed legal and does not violate the company policy, then the second question raises the flag that a decision that greatly benefits one individual or company will eventually come back to haunt that individual or company. The last question is directed to prompt our sense of fairness and to ensure that our self-esteem is not diminished through an unethical decision.

In addition to the checklist, Blanchard and Peale discussed the five P's that can be considered in analyzing the solution: Purpose, Pride, Patience, Persistence, and Perspective.

### The Five P's

*Purpose:* What is the objective for which you are striving? Are you comfortable with that as your purpose? Does your purpose hold up when you look at yourself in the mirror?

*Pride:* Can you take pride in the solution you have developed? Is there any false pride or self-doubt involved?

*Patience:* Have you taken the time to think through all the ramifications of your solution?

*Persistence:* Are you sticking to your guns and not being dissuaded by other demands? Have you given up too soon on finding a solution that is fair and balanced to all concerned?

*Perspective:* Have you taken the time to focus inside yourself to be sure everything fits with your ideals and beliefs? How does the solution fit into the "Big Picture"?

Figure B-1 shows the interrelationships among the five P's. Perspective is the fifth P, the hub around which the other P's rotate. Part of Perspective is the inner guidance that is awakened from the other P's that helps us see things more clearly [4].

The various codes of conduct in this appendix should enable the engineers to make the correct decisions which can be identified by the use of a heuristic. The correct decision is one that is [2]

1. Consistent with the engineer's moral principles.
2. Consistent with the general accepted codes of engineering conduct.
3. Consistent with obligations that the engineer has accepted.
4. Consistent with the law.
5. Consistent with the applicable code of ethics.

Although it is possible that an individual decision may not be acceptable to another individual, ultimately the correct decision is one that individual can live with. The ability to make one's own ethical decisions is known as moral autonomy.

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## Appendix C

# PHYSICAL PROPERTIES OF LIQUIDS AND GASES

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### TABLES OF PHYSICAL PROPERTIES OF LIQUIDS AND GASES

C-1	Density of Liquids
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**TABLE C-1 Density of Liquids**

No.	Formula	Substance	$\rho_L = A B^{-(1-T/T_c)^n}$ ( $\rho_L$ —g/ml, $T$ —K)						
			A	B	n	$T_c$	$T_{\min}$	$T_{\max}$	$\rho_L$ at 25°C
1	C <sub>2</sub> H <sub>3</sub> C <sub>13</sub>	1,1,1-Trichloroethane	0.47476	0.27258	0.29333	545.00	242.75	545.00	1.330
2	C <sub>2</sub> H <sub>3</sub> C <sub>13</sub>	1,1,2-Trichloroethane	0.47455	0.25475	0.31000	602.00	236.50	602.00	1.435
3	C <sub>2</sub> H <sub>4</sub> C <sub>12</sub>	1,1-Dichloroethane	0.41231	0.26533	0.28700	523.00	176.19	523.00	1.168
4	C <sub>2</sub> H <sub>4</sub> C <sub>12</sub>	1,2-Dichloroethane	0.46501	0.28742	0.31041	561.00	237.49	561.00	1.246
5	C <sub>4</sub> H <sub>6</sub>	1,3-Butadiene	0.24597	0.27227	0.29074	425.37	164.25	425.37	0.615
6	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	1,4-Dioxane	0.37018	0.28130	0.30470	587.00	284.95	587.00	1.029
7	C <sub>4</sub> H <sub>10</sub> O	1-Butanol ( <i>n</i> -Butanol)	0.26891	0.26674	0.24570	562.93	183.85	562.93	0.806
8	C <sub>4</sub> H <sub>8</sub>	1-Butene	0.23224	0.26630	0.28530	419.59	87.80	419.59	0.588
9	C <sub>10</sub> H <sub>20</sub>	1-Decene	0.23981	0.25776	0.28562	617.05	206.89	617.05	0.737
10	C <sub>9</sub> H <sub>20</sub>	1-Nonane ( <i>n</i> -Nonane)	0.23364	0.25556	0.28571	595.65	219.63	595.65	0.715
11	C <sub>8</sub> H <sub>16</sub>	1-Octene	0.23682	0.25649	0.28571	566.60	171.45	566.60	0.711
12	C <sub>3</sub> H <sub>8</sub> O	1-Propanol ( <i>n</i> -Propanol)	0.27684	0.27200	0.24940	536.71	146.95	536.71	0.802
13	C <sub>2</sub> H <sub>4</sub> O	Acetaldehyde	0.28207	0.26004	0.27760	461.00	150.15	461.00	0.774
14	C <sub>2</sub> H <sub>4</sub> O <sub>2</sub>	Acetic acid	0.35182	0.26954	0.26843	592.71	289.81	592.71	1.043
15	C <sub>4</sub> H <sub>6</sub> O <sub>3</sub>	Acetic anhydride	0.33578	0.24080	0.26990	569.15	200.15	569.15	1.077
16	C <sub>3</sub> H <sub>6</sub> O	Acetone	0.27728	0.25760	0.29903	508.20	178.45	508.20	0.786
17	C <sub>3</sub> H <sub>4</sub> O <sub>2</sub>	Acrylic acid	0.34645	0.25822	0.30701	615.00	286.65	615.00	1.046
18	NH <sub>3</sub>	Ammonia	0.23689	0.25471	0.28870	405.65	195.41	405.65	0.602
19	C <sub>6</sub> H <sub>7</sub> N	Aniline	0.31190	0.25000	0.28571	699.00	267.13	699.00	1.018
20	C <sub>6</sub> H <sub>6</sub>	Benzene	0.30090	0.26770	0.28180	562.16	278.68	562.16	0.873
21	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Butyric acid	0.31132	0.26192	0.27997	628.00	267.95	628.00	0.953
22	CS <sub>2</sub>	Carbon disulfide	0.47589	0.28749	0.32260	552.00	161.58	552.00	1.256
23	CO <sub>2</sub>	Carbon dioxide	0.46382	0.26160	0.29030	304.19	216.58	304.19	0.713
24	CO	Carbon monoxide	0.29818	0.27655	0.29053	132.92	68.15	132.92	—
25	CC <sub>14</sub>	Carbon tetrachloride	0.56607	0.27663	0.29000	556.35	250.33	556.35	1.583
26	Cl <sub>2</sub>	Chlorine	0.56600	0.27315	0.28830	417.15	172.12	417.15	1.398
27	CHC <sub>13</sub>	Chloroform	0.49807	0.25274	0.28766	536.40	209.63	536.40	1.480
28	C <sub>6</sub> H <sub>12</sub>	Cyclohexane	0.27376	0.27408	0.28511	553.54	279.69	553.54	0.773
29	C <sub>6</sub> H <sub>12</sub> O	Cyclohexanol	0.29681	0.24340	0.28570	625.15	296.60	625.15	0.960
30	C <sub>3</sub> H <sub>6</sub>	Cyclopropane	0.25880	0.27400	0.28571	397.91	145.73	397.91	0.619
31	CH <sub>2</sub> C <sub>12</sub>	Dichloromethane	0.45965	0.25678	0.29020	510.00	178.01	510.00	1.318
32	C <sub>4</sub> H <sub>10</sub> O	Diethyl ether	0.27267	0.27608	0.29358	466.70	156.85	466.70	0.708
33	C <sub>5</sub> H <sub>10</sub> O	Diethyl ketone	0.25635	0.24291	0.27364	560.95	234.18	560.95	0.810
34	C <sub>2</sub> H <sub>7</sub> N	Dimethylamine	0.24110	0.26785	0.24800	437.65	180.96	437.65	0.650
35	C <sub>2</sub> H <sub>6</sub>	Ethane	0.20087	0.27330	0.28330	305.42	90.35	305.42	0.315
36	C <sub>2</sub> H <sub>6</sub> O	Ethanol	0.26570	0.26395	0.23670	516.25	159.05	516.25	0.787
37	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Ethyl acetate	0.30654	0.25856	0.27800	523.30	189.60	523.30	0.894
38	C <sub>2</sub> H <sub>5</sub> Cl	Ethyl chloride	0.32259	0.27464	0.23140	460.35	136.75	460.35	0.890
39	C <sub>8</sub> H <sub>10</sub>	Ethylbenzene	0.28889	0.26438	0.29210	617.17	178.20	617.17	0.865
40	C <sub>2</sub> H <sub>4</sub>	Ethylenes	0.21428	0.28061	0.28571	282.36	104.01	282.36	—
41	C <sub>2</sub> H <sub>6</sub> O <sub>2</sub>	Ethylene glycol	0.32503	0.25499	0.17200	645.00	260.15	645.00	1.110
42	C <sub>2</sub> H <sub>4</sub> O	Ethylene oxide	0.31402	0.26089	0.28253	469.15	161.45	469.15	0.862
43	F <sub>2</sub>	Fluorine	0.57092	0.28518	0.29000	144.31	53.48	144.31	—
44	C <sub>3</sub> H <sub>8</sub> O <sub>3</sub>	Glycerol	0.34908	0.24902	0.15410	723.00	291.33	723.00	1.257
45	H <sub>2</sub>	Hydrogen	0.03125	0.34730	0.27560	33.18	13.95	33.18	—
46	HCl	Hydrogen chloride	0.44134	0.26957	0.31870	324.65	159.97	324.65	0.796
47	CHN	Hydrogen cyanide	0.19501	0.18589	0.28206	456.65	259.91	465.65	0.680
48	H <sub>2</sub> O <sub>2</sub>	Hydrogen peroxide	0.43776	0.24982	0.28770	730.15	272.74	730.15	1.443
49	C <sub>4</sub> H <sub>10</sub>	<i>i</i> -Butane ( <i>iso</i> -Butane)	0.22281	0.27294	0.27301	408.14	113.54	408.14	0.552
50	CH <sub>4</sub>	Methane	0.15998	0.28810	0.27700	190.58	90.67	190.58	—
51	CH <sub>4</sub> O	Methanol	0.27197	0.27192	0.23310	512.58	175.47	512.58	0.787

**TABLE C-1—(continued)**

No.	Formula	Substance	$\rho_L = A B^{-(1 - T/T_c)^n}$						$(\rho_L - \text{g/ml}, T - \text{K})$
			A	B	n	$T_c$	$T_{\min}$	$T_{\max}$	
52	CH <sub>3</sub> Br	Methyl bromide	0.60859	0.26292	0.28030	467.00	179.55	467.00	1.662
53	CH <sub>3</sub> Cl	Methyl chloride	0.35821	0.26109	0.28690	416.25	175.45	416.25	0.913
54	C <sub>6</sub> H <sub>12</sub> O	Methyl isobutyl ketone	0.26654	0.25887	0.28571	571.40	189.15	571.40	0.796
55	CH <sub>5</sub> N	Methylamine	0.20168	0.21405	0.22750	430.05	179.69	430.05	0.655
56	C <sub>8</sub> H <sub>10</sub>	<i>m</i> -Xylene	0.27866	0.25925	0.27243	617.05	225.30	617.05	0.861
57	C <sub>10</sub> H <sub>8</sub>	Naphthalene	0.30619	0.25037	0.28300	748.35	353.43	748.35	—
58	C <sub>4</sub> H <sub>10</sub>	<i>n</i> -Butane	0.22827	0.27240	0.28630	425.18	134.86	425.18	0.573
59	C <sub>10</sub> H <sub>22</sub>	<i>n</i> -Decane	0.23276	0.25240	0.28570	618.45	243.49	618.45	0.728
60	C <sub>7</sub> H <sub>16</sub>	<i>n</i> -Heptane	0.23237	0.26020	0.27910	540.26	182.57	540.26	0.682
61	C <sub>6</sub> H <sub>14</sub>	<i>n</i> -Hexane	0.23242	0.26500	0.27810	507.43	177.84	507.43	0.656
62	C <sub>6</sub> H <sub>5</sub> NO <sub>2</sub>	Nitrobenzene	0.36140	0.24731	0.28570	719.00	278.91	719.00	1.199
63	N <sub>2</sub>	Nitrogen	0.31205	0.28479	0.29250	126.10	63.15	126.10	—
64	C <sub>9</sub> H <sub>20</sub>	<i>n</i> -Nonane	0.23364	0.25556	0.28571	595.65	219.63	595.65	0.715
65	C <sub>8</sub> H <sub>18</sub>	<i>n</i> -Octane	0.22807	0.25476	0.26940	568.83	216.38	568.83	0.699
66	C <sub>5</sub> H <sub>12</sub>	<i>n</i> -Pentane	0.23143	0.26923	0.28215	469.65	143.42	469.65	0.621
67	O <sub>2</sub>	Oxygen	0.43533	0.28772	0.29240	154.58	54.35	154.58	—
68	C <sub>8</sub> H <sub>10</sub>	<i>o</i> -Xylene	0.28381	0.26083	0.27410	630.37	247.98	630.37	0.876
69	C <sub>6</sub> H <sub>6</sub> O	Phenol	0.41476	0.32162	0.32120	694.25	314.06	694.25	—
70	C <sub>8</sub> H <sub>10</sub>	<i>p</i> -Xylene	0.27984	0.26003	0.27900	616.26	286.41	616.26	0.858
71	C <sub>5</sub> H <sub>5</sub> N	Pyridine	0.30752	0.24333	0.30450	619.95	231.53	619.95	0.979
72	C <sub>8</sub> H <sub>8</sub>	Styrene	0.29383	0.26315	0.28570	648.00	242.54	648.00	0.900
73	C <sub>7</sub> H <sub>8</sub>	Toluene	0.29999	0.27108	0.29889	591.79	178.18	591.79	0.865
74	C <sub>4</sub> H <sub>6</sub> O <sub>2</sub>	Vinyl acetate	0.31843	0.25803	0.28270	524.00	180.35	524.00	0.926
75	H <sub>2</sub> O	Water	0.34710	0.27400	0.28571	647.13	273.16	647.13	1.027

$\rho_L$  – density of liquid, g/ml  
 $A, B, n$ , and  $T_c$  – regression coefficients of chemical compound  
 $T$  – temperature, K  
 $T_{\min}$  – minimum temperature, K  
 $T_{\max}$  – maximum temperature, K

**TABLE C-2 Viscosity of Gas**

No.	Formula	Substance	$\mu_G = A + BT + CT^2$							
			A	B	C	T <sub>min</sub>	T <sub>max</sub>	$\mu_G$ at 25°C	$\mu_G$ at T <sub>min</sub>	$\mu_G$ at T <sub>max</sub>
1	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,1-Trichloroethane	-19.216	4.0308E-01	-6.8618E-05	347	997	-	112.39	314.45
2	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,2-Trichloroethane	-8.293	3.3989E-01	-5.3678E-05	387	987	-	115.21	274.89
3	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,1-Dichloroethane	-12.991	4.0085E-01	-1.1779E-04	320	472	-	103.22	149.97
4	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,2-Dichloroethane	1.025	3.1792E-01	-4.1853E-05	322	561	-	99.06	166.21
5	C <sub>4</sub> H <sub>6</sub>	1,3-Butadiene	10.256	2.6833E-01	-4.1148E-05	250	650	86.60	74.77	167.28
6	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	1,4-Dioxane	-16701	3.4988E-01	-5.3736E-05	374	994	-	106.64	277.99
7	C <sub>4</sub> H <sub>10</sub> O	1-Butanol ( <i>n</i> -Butanol)	-11.144	2.8790E-01	-5.6275E-05	391	1000	-	92.82	220.48
8	C <sub>4</sub> H <sub>8</sub>	1-Butene	-9.143	3.1562E-01	-8.4164E-05	175	800	77.48	43.51	189.49
9	C <sub>10</sub> H <sub>20</sub>	1-Decene	4.015	1.8083E-01	-3.8216E-05	444	1000	-	76.77	146.63
10	C <sub>9</sub> H <sub>20</sub>	1-Nonane ( <i>n</i> -Nonane)	-6.802	1.8688E-01	3.4929E-07	273	773	48.95	44.24	137.86
11	C <sub>8</sub> H <sub>16</sub>	1-Octene	2.722	2.0327E-01	-4.3879E-05	394	994	-	76.00	161.42
12	C <sub>3</sub> H <sub>8</sub> O	1-Propanol ( <i>n</i> -Propanol)	-14.894	3.2171E-01	-5.8021E-05	200	1000	75.87	47.13	248.80
13	C <sub>2</sub> H <sub>4</sub> O	Acetaldehyde	0.069	3.0246E-01	-4.2372E-05	294	1000	86.48	85.33	260.16
14	C <sub>2</sub> H <sub>4</sub> O <sub>2</sub>	Acetic acid	-28.660	2.3510E-01	2.2087E-04	366	523	-	86.97	154.71
15	C <sub>4</sub> H <sub>6</sub> O <sub>3</sub>	Acetic anhydride	-1.485	2.8869E-01	-2.3391E-05	295	993	82.51	81.64	262.12
16	C <sub>3</sub> H <sub>6</sub> O	Acetone	-4.055	2.6655E-01	-5.6936E-06	300	650	-	75.40	166.80
17	C <sub>3</sub> H <sub>4</sub> O <sub>2</sub>	Acrylic acid	-6.532	3.0600E-01	-4.6620E-05	287	1000	80.56	77.45	252.85
18	NH <sub>3</sub>	Ammonia	-7.874	3.6700E-01	-4.4700E-06	195	1000	101.28	63.61	355.10
19	C <sub>6</sub> H <sub>7</sub> N	Aniline	-6.918	2.5935E-01	-3.4348E-05	458	1000	-	104.66	218.08
20	C <sub>6</sub> H <sub>6</sub>	Benzene	-0.151	2.5706E-01	-8.9797E-06	287	628	75.69	72.89	157.74
21	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Butyric acid	-5.781	2.6159E-01	-3.4903E-05	268	1000	69.11	61.82	220.91
22	CS <sub>2</sub>	Carbon disulfide	-7.700	3.6594E-01	-2.5416E-05	273	583	99.15	90.31	197.00
23	CO <sub>2</sub>	Carbon dioxide	11.811	4.9838E-01	-1.0851E-04	195	1500	150.76	104.87	515.23
24	CO	Carbon monoxide	23.811	5.3944E-01	-1.5411E-04	68	1250	170.95	59.78	457.31
25	CCl <sub>4</sub>	Carbon tetrachloride	-7.745	3.9481E-01	-1.1150E-04	280	800	100.06	94.06	236.74
26	CHCl <sub>3</sub>	Chloroform	-4.392	3.7309E-01	-5.1696E-05	250	700	102.25	85.65	231.44
27	Cl <sub>2</sub>	Chlorine	-3.571	4.8700E-01	-8.5300E-05	200	1000	134.03	90.41	398.09
28	C <sub>6</sub> H <sub>12</sub>	Cyclohexane	1.190	2.4542E-01	-3.8334E-05	315	600	-	74.69	134.64
29	C <sub>6</sub> H <sub>12</sub> O	Cyclohexanol	-13.542	2.9086E-01	-3.9472E-05	400	1000	-	96.49	237.85
30	C <sub>3</sub> H <sub>6</sub>	Cyclopropane	-9.521	3.7037E-01	-1.3138E-04	240	446	89.23	71.80	129.53
31	CH <sub>2</sub> Cl <sub>2</sub>	Dichloromethane	-20.372	4.3745E-01	-7.7549E-05	273	993	103.16	93.27	337.55
32	C <sub>4</sub> H <sub>10</sub> O	Diethyl ether	-7.932	3.0235E-01	-7.3858E-05	200	1000	75.65	49.58	220.56
33	C <sub>5</sub> H <sub>10</sub> O	Diethyl ketone	-3.593	2.4308E-01	-3.7613E-05	234	1000	65.54	51.23	201.87
34	C <sub>2</sub> H <sub>7</sub> N	Dimethylamine	-9.275	2.8958E-01	-1.3875E-06	250	450	76.94	63.03	120.75
35	C <sub>2</sub> H <sub>6</sub>	Ethane	0.514	3.3449E-01	-7.1071E-05	150	1000	93.92	49.09	263.93
36	C <sub>2</sub> H <sub>6</sub> O	Ethanol	1.499	3.0741E-01	-4.4479E-05	200	1000	89.20	61.20	264.43
37	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Ethyl acetate	-9.259	3.0725E-01	-7.1069E-05	190	1000	76.03	46.55	226.92
38	C <sub>2</sub> H <sub>5</sub> Cl	Ethyl chloride	0.458	3.2827E-01	-1.2467E-05	213	523	97.22	69.81	168.73
39	C <sub>8</sub> H <sub>10</sub>	Ethylbenzene	-4.267	2.4735E-01	-5.4264E-05	409	1000	-	87.82	188.82
40	C <sub>2</sub> H <sub>4</sub>	Ethylene	-3.985	3.8726E-01	-1.1227E-04	150	1000	101.50	51.58	271.00
41	C <sub>2</sub> H <sub>6</sub> O <sub>2</sub>	Ethylene glycol	-7.178	3.1246E-01	-4.4028E-05	260	1000	82.07	71.09	261.25
42	C <sub>2</sub> H <sub>4</sub> O	Ethylene oxide	-12.180	3.7672E-01	-7.7599E-05	250	1000	93.24	77.15	286.94
43	F <sub>2</sub>	Fluorine	-0.811	8.9800E-01	-3.9600E-04	90	500	231.59	76.76	348.97
44	C <sub>3</sub> H <sub>8</sub> O <sub>3</sub>	Glycerol	-23.119	2.8879E-01	-3.4277E-05	563	993	-	128.61	229.85
45	H <sub>2</sub>	Hydrogen	27.758	2.1200E-01	-3.2800E-05	150	1500	88.03	58.81	271.76
46	HCl	Hydrogen chloride	-9.118	5.5500E-01	-1.1100E-04	200	1000	146.44	97.42	434.55
47	CHN	Hydrogen cyanide	-8.486	9.0368E-02	7.9146E-05	300	425	-	25.75	44.22
48	H <sub>2</sub> O <sub>2</sub>	Hydrogen peroxide	8.039	2.7000E-01	8.2900E-05	373	600	-	120.24	199.81
49	C <sub>4</sub> H <sub>10</sub>	<i>i</i> -Butane ( <i>iso</i> -Butane)	-4.731	2.9131E-01	-8.0995E-05	150	1000	74.92	37.14	205.58
50	CH <sub>4</sub>	Methane	3.844	4.0112E-01	-1.4303E-04	91	850	110.72	39.15	241.46
51	CH <sub>3</sub> OH	Methanol	-14.236	3.8935E-01	-6.2762E-05	240	1000	96.27	75.59	312.35

**TABLE C-2—(continued)**

No.	Formula	Substance	$\mu_G = A + BT + CT^2$								
			A	B	C	$T_{\min}$	$T_{\max}$	$\mu_G$ at 25°C	$\mu_G$ at $T_{\min}$	$\mu_G$ at $T_{\max}$	
52	CH <sub>3</sub> Br	Methyl bromide	-27.740	5.5901E-01	-7.0942E-05	260	440	132.62	112.81	204.49	
53	CH <sub>3</sub> Cl	Methyl chloride	-1.374	3.8627E-01	-4.8650E-05	230	700	109.47	84.90	245.18	
54	C <sub>6</sub> H <sub>12</sub> O	Methyl isobutyl ketone	-3.237	2.3310E-01	-3.5612E-05	189	1000	63.10	39.55	194.25	
55	CH <sub>5</sub> N	Methylamine	-5.334	3.4181E-01	-7.4297E-05	267	1000	89.97	80.63	262.18	
56	C <sub>8</sub> H <sub>10</sub>	<i>m</i> -Xylene	-21.620	2.7820E-01	-6.0531E-05	250	1000	55.94	44.15	196.05	
57	C <sub>10</sub> H <sub>8</sub>	Naphthalene	-16.789	2.5406E-01	-3.5495E-05	353	1000	—	68.47	201.78	
58	C <sub>4</sub> H <sub>10</sub>	<i>n</i> -Butane	-4.946	2.9001E-01	-6.9665E-05	150	1200	75.33	36.99	242.75	
59	C <sub>10</sub> H <sub>22</sub>	<i>n</i> -Decane	-7.297	1.8506E-01	-4.8008E-06	287	783	47.45	43.78	134.66	
60	C <sub>7</sub> H <sub>16</sub>	<i>n</i> -Heptane	-10.378	2.4401E-01	-5.4003E-05	338	700	—	65.93	133.97	
61	C <sub>6</sub> H <sub>14</sub>	<i>n</i> -Hexane	-8.222	2.6229E-01	-5.7366E-05	300	1000	—	65.30	196.70	
62	C <sub>6</sub> H <sub>5</sub> NO <sub>2</sub>	Nitrobenzene	-16.569	2.9184E-01	-2.5523E-05	450	1000	—	109.59	249.75	
63	N <sub>2</sub>	Nitrogen	42.606	4.7500E-01	-9.8800E-05	150	1500	175.52	111.67	533.12	
64	C <sub>9</sub> H <sub>20</sub>	<i>n</i> -Nonane	-6.802	1.8688E-01	3.4929E-07	273	773	48.95	44.24	137.86	
65	C <sub>8</sub> H <sub>18</sub>	<i>n</i> -Octane	3.940	1.6640E-01	1.4470E-05	374	670	—	68.20	121.92	
66	C <sub>5</sub> H <sub>12</sub>	<i>n</i> -Pentane	-3.202	2.6746E-01	-6.6178E-05	303	900	—	71.76	183.91	
67	O <sub>2</sub>	Oxygen	44.224	5.6200E-01	-1.1300E-04	150	1500	201.85	126.04	633.08	
68	C <sub>8</sub> H <sub>10</sub>	<i>o</i> -Xylene	-19.763	2.8022E-01	-5.9293E-05	250	1000	58.51	46.59	201.16	
69	C <sub>6</sub> H <sub>6</sub> O	Phenol	-7.185	2.7179E-01	-3.6205E-05	455	1000	—	108.98	228.40	
70	C <sub>3</sub> H <sub>8</sub>	Propane	-5.462	3.2722E-01	-1.0672E-04	193	750	82.61	53.72	179.92	
71	C <sub>8</sub> H <sub>10</sub>	<i>p</i> -Xylene	-17.226	2.5098E-01	-2.8232E-05	286	1000	55.09	52.25	205.52	
72	C <sub>5</sub> H <sub>5</sub> N	Pyridine	-5.739	2.7135E-01	-1.7202E-05	369	998	—	92.05	247.93	
73	C <sub>8</sub> H <sub>8</sub>	Styrene	-10.035	2.5191E-01	-3.7932E-05	243	1000	61.70	48.94	203.94	
74	SO <sub>2</sub>	Sulfur dioxide	-11.103	5.0200E-01	-1.0800E-04	200	1000	129.10	85.07	383.30	
75	C <sub>7</sub> H <sub>8</sub>	Toluene	1.787	2.3566E-01	-9.3508E-06	275	600	71.22	65.89	139.82	
76	C <sub>4</sub> H <sub>6</sub> O <sub>2</sub>	Vinyl acetate	-7.462	3.0466E-01	-5.7544E-05	346	1000	—	91.06	239.65	
77	H <sub>2</sub> O	Water	-36.826	4.2900E-01	-1.6200E-05	280	1073	89.68	82.07	404.97	

$\mu_G$  – viscosity of gas,  $\mu$ P

A, B, and C – regression coefficients for chemical compound

T – temperature, K

$T_{\min}$  – minimum temperature, K

$T_{\max}$  – maximum temperature, K

**TABLE C-3 Viscosity of Liquids**

$\log_{10}\mu_{\text{liq}} = A + B/T + CT + CT^2$										
No.	Formula	Substance	A	B	C	D	T <sub>min</sub>	T <sub>max</sub>	$\mu_{\text{liq}}$ at 25°C	$\mu_{\text{liq}}$ at T <sub>max</sub>
1	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,1-Trichloroethane	-3.9096	7.0709E+02	7.5847E-03	-9.1662E-06	243	545	0.810	0.063
2	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,2-Trichloroethane	-3.2716	6.8810E+02	4.8932E-03	-5.4671E-06	237	602	1.021	0.069
3	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,1-Dichloroethane	-3.8388	5.9046E+02	8.0953E-03	-9.9210E-06	176	523	0.471	0.065
4	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,2-Dichloroethane	-0.1656	2.7576E+02	-3.3493E-03	1.4093E-06	245	561	0.769	0.078
5	C <sub>4</sub> H <sub>6</sub>	1,3-Butadiene	0.3772	7.9658E+01	-5.8889E-03	2.9221E-06	250	425	0.141	0.039
6	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	1,4-Dioxane	-7.5724	1.3813E+03	1.3556E-02	-1.1464E-05	288	587	1.211	0.061
7	C <sub>4</sub> H <sub>10</sub> O	1-Butanol ( <i>n</i> -Butanol)	-5.3970	1.3256E+03	6.2223E-03	-5.5062E-06	250	563	2.599	0.052
8	C <sub>4</sub> H <sub>8</sub>	1-Butene	-4.9218	4.9503E+02	1.4390E-02	-2.0853E-05	160	420	0.150	0.042
9	C <sub>10</sub> H <sub>20</sub>	1-Decene	-6.8845	1.1003E+03	1.4341E-02	-1.3520E-05	273	617	0.758	0.040
10	C <sub>9</sub> H <sub>20</sub>	1-Nonane ( <i>n</i> -Nonane)	-6.0742	9.6861E+02	1.2677E-02	-1.2675E-05	220	596	0.672	0.040
11	C <sub>8</sub> H <sub>16</sub>	1-Octene	-5.6209	8.1305E+02	1.2523E-02	-1.3384E-05	250	567	0.447	0.041
12	C <sub>3</sub> H <sub>8</sub> O	1-Propanol ( <i>n</i> -Propanol)	-3.7702	9.9151E+02	4.0836E-03	-5.4586E-06	220	537	1.939	0.05
13	C <sub>2</sub> H <sub>4</sub> O	Acetaldehyde	-6.6171	6.8123E+02	1.9979E-02	-2.5563E-05	260	461	0.225	0.043
14	C <sub>2</sub> H <sub>4</sub> O <sub>2</sub>	Acetic acid	-3.8937	7.8482E+02	6.6650E-03	-7.5606E-06	290	593	1.132	0.053
15	C <sub>4</sub> H <sub>6</sub> O <sub>3</sub>	Acetic anhydride	-17.3580	2.3611E+03	4.2734E-02	-3.8202E-05	265	569	0.806	0.055
16	C <sub>3</sub> H <sub>6</sub> O	Acetone	-7.2126	9.0305E+02	1.8385E-02	-2.0353E-05	223	508	0.308	0.045
17	C <sub>3</sub> H <sub>4</sub> O <sub>2</sub>	Acrylic acid	-15.9215	2.4408E+03	3.4383E-02	-2.7677E-05	293	615	1.138	0.053
18	NH <sub>3</sub>	Ammonia	-8.5910	8.7640E+02	2.6810E-02	-3.6120E-05	195	406	0.135	0.032
19	C <sub>6</sub> H <sub>7</sub> N	Aniline	-13.8625	2.5109E+03	2.5681E-02	-1.8281E-05	268	699	3.898	0.056
20	C <sub>6</sub> H <sub>6</sub>	Benzene	-7.4005	1.1815E+03	1.4888E-02	-1.3713E-05	285	562	0.606	0.055
21	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Butyric acid	-7.9846	1.3636E+03	1.6315E-02	-1.4511E-05	268	628	1.457	0.051
22	CS <sub>2</sub>	Carbon disulfide	-9.1108	1.1216E+03	2.3216E-02	-2.2648E-05	235	552	0.363	0.068
23	CO <sub>2</sub>	Carbon dioxide	-19.4921	1.5948E+03	7.9274E-02	-1.2025E-04	219	304	0.064	0.055
24	CO	Carbon monoxide	-1.1224	5.7858E+01	-4.9174E-03	8.2233E-06	69	133	—	0.064
25	CCl <sub>4</sub>	Carbon tetrachloride	-6.4564	1.0379E+03	1.4021E-02	-1.4107E-05	265	556	0.893	0.070
26	Cl <sub>2</sub>	Chlorine	-0.7681	1.5140E+02	-8.0650E-04	4.0750E-07	172	417	0.343	0.214
27	CHCl <sub>3</sub>	Chloroform	-47.831	6.9902E+02	1.0929E-02	-1.2244E-05	210	536	0.539	0.073
28	C <sub>6</sub> H <sub>12</sub>	Cyclohexane	4.7423	-2.5322E+02	-1.6927E-02	1.2472E-05	285	554	0.901	0.054
29	C <sub>6</sub> H <sub>12</sub> O	Cyclohexanol	-5.3792	1.8793E+03	1.7011E-03	1.0187E-07	303	625	—	0.054
30	C <sub>3</sub> H <sub>6</sub>	Cyclopropane	-3.2541	3.2192E+02	9.9766E-03	-1.8191E-05	146	398	0.152	0.044
31	CH <sub>2</sub> Cl <sub>2</sub>	Dichloromethane	-5.1043	6.8653E+02	1.2459E-02	-1.4540E-05	208	510	0.417	0.065
32	C <sub>4</sub> H <sub>10</sub> O	Diethyl ether	-8.5060	1.0020E+03	2.2753E-02	-2.5780E-05	233	467	0.222	0.044
33	C <sub>5</sub> H <sub>10</sub> O	Diethyl ketone	-9.2905	1.2716E+03	2.1925E-02	-2.1036E-05	274	561	0.438	0.045
34	C <sub>2</sub> H <sub>7</sub> N	Dimethylamine	-11.5558	1.2126E+03	3.4999E-02	-4.1253E-05	240	438	0.190	0.042
35	C <sub>2</sub> H <sub>6</sub>	Ethane	-4.2694	2.8954E+02	1.7111E-02	-3.6092E-05	98	305	0.039	0.035
36	C <sub>2</sub> H <sub>6</sub> O	Ethanol	-6.4406	1.1176E+03	1.3721E-02	-1.5465E-05	240	516	1.057	0.049
37	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Ethyl acetate	-3.6861	5.5228E+02	8.0018E-03	-1.0439E-05	220	523	0.421	0.050
38	C <sub>2</sub> H <sub>5</sub> Cl	Ethyl chloride	-4.4279	5.1891E+02	1.2035E-02	-1.6620E-05	150	460	0.265	0.052
39	C <sub>8</sub> H <sub>10</sub>	Ethylbenzene	-5.2585	8.3065E+02	1.0784E-02	-1.0618E-05	210	617	0.629	0.050
40	C <sub>2</sub> H <sub>4</sub>	Ethylene	-4.5611	3.0811E+02	1.8030E-02	-3.8145E-05	105	282	—	0.038
41	C <sub>2</sub> H <sub>6</sub> O <sub>2</sub>	Ethylene glycol	-16.9728	3.1886E+03	3.2537E-02	-2.4480E-05	261	645	17.645	0.059
42	C <sub>2</sub> H <sub>4</sub> O	Ethylene oxide	-5.7794	6.7020E+02	1.5686E-02	-1.9462E-05	190	469	0.260	0.053
43	F <sub>2</sub>	Fluorine	-1.5760	8.5630E+01	-4.0730E-04	-2.7250E-06	54	145	—	0.079
44	C <sub>3</sub> H <sub>8</sub> O <sub>3</sub>	Glycerol	-18.2152	4.2305E+03	2.8705E-02	-1.8648E-05	293	723	749.338	0.044
45	H <sub>2</sub>	Hydrogen	-7.0154	4.0791E+01	2.3714E-01	-4.0830E-03	14	33	—	0.004
46	HCl	Hydrogen chloride	-1.5150	1.9460E+02	3.0670E-03	-1.3760E-05	159	325	0.067	0.042
47	CHN	Hydrogen cyanide	-12.0812	1.3183E+03	3.5234E-02	-4.0185E-05	260	457	0.188	0.033
48	H <sub>2</sub> O <sub>2</sub>	Hydrogen peroxide	-1.6150	5.0380E+02	3.5010E-04	-1.1680E-06	273	728	1.189	0.052
49	C <sub>4</sub> H <sub>10</sub>	<i>i</i> -Butane ( <i>iso</i> -Butane)	-13.4207	1.3131E+03	4.4329E-02	-5.5793E-05	190	408	0.174	0.039
50	CH <sub>4</sub>	Methane	-7.3801	3.1925E+02	4.7934E-02	-1.4120E-04	91	191	—	0.020

**TABLE C-3—(continued)**

$\log_{10}\mu_{\text{liq}} = A + B/T + CT + CT^2$										
No.	Formula	Substance	A	B	C	D	T <sub>min</sub>	T <sub>max</sub>	$\mu_{\text{liq}}$ at 25°C	$\mu_{\text{liq}}$ at T <sub>max</sub>
51	CH <sub>3</sub> Br	Methyl bromide	-9.5533	1.0306E+03	2.8322E-02	-3.1920E-05	193	467	0.324	0.083
52	CH <sub>3</sub> Cl	Methyl chloride	-7.3473	8.5395E+02	1.9485E-02	-2.3484E-05	249	416	0.173	0.056
53	C <sub>6</sub> H <sub>12</sub> O	Methyl isobutyl ketone	-3.0570	5.0050E+02	6.5038E-03	-8.8243E-06	246	571	0.598	0.045
54	CH <sub>5</sub> N	Methylamine	-9.4670	9.8286E+02	2.8918E-02	-3.5672E-05	180	430	0.191	0.045
55	C <sub>8</sub> H <sub>10</sub>	<i>m</i> -Xylene	-6.0517	9.2460E+02	1.2583E-02	-1.1850E-05	225	617	0.559	0.050
56	C <sub>10</sub> H <sub>8</sub>	Naphthalene	-10.3716	1.8572E+03	1.9320E-02	-1.4012E-05	353	748	—	0.053
57	C <sub>4</sub> H <sub>10</sub>	<i>n</i> -Butane	-6.8590	6.7393E+02	2.1973E-02	-3.0686E-05	180	425	0.168	0.033
58	C <sub>10</sub> H <sub>22</sub>	<i>n</i> -Decane	-6.0716	1.0177E+03	1.2247E-02	-1.1892E-05	243	618	0.863	0.040
59	C <sub>7</sub> H <sub>16</sub>	<i>n</i> -Heptane	-5.7782	8.0587E+02	1.3355E-02	-1.4794E-05	183	540	0.390	0.041
60	C <sub>6</sub> H <sub>14</sub>	<i>n</i> -Hexane	-5.0715	6.5536E+02	1.2349E-02	-1.5042E-05	178	507	0.296	0.041
61	C <sub>6</sub> H <sub>5</sub> NO <sub>2</sub>	Nitrobenzene	-7.7710	1.4019E+03	1.4653E-02	-1.1512E-05	273	719	1.890	0.058
62	N <sub>2</sub>	Nitrogen	-15.6104	4.6505E+02	1.6259E-01	-6.3353E-04	63	125	—	0.034
63	C <sub>9</sub> H <sub>20</sub>	<i>n</i> -Nonane	-6.0742	9.6861E+02	1.2677E-02	-1.2675E-05	220	596	0.672	0.040
64	C <sub>8</sub> H <sub>18</sub>	<i>n</i> -Octane	-5.9245	8.8809E+02	1.2955E-02	-1.3596E-05	216	569	0.511	0.040
65	C <sub>5</sub> H <sub>12</sub>	<i>n</i> -Pentane	-7.1711	7.4736E+02	2.1697E-02	-2.7176E-05	143	470	0.245	0.041
66	O <sub>2</sub>	Oxygen	-5.0957	1.7983E+02	3.9779E-02	-1.4664E-04	54	150	—	0.059
67	C <sub>8</sub> H <sub>10</sub>	<i>o</i> -Xylene	-7.8805	1.2500E+03	1.6116E-02	-1.3993E-05	268	630	0.747	0.050
68	C <sub>6</sub> H <sub>6</sub> O	Phenol	1.5349	4.2620E+02	-9.1577E-03	6.2322E-06	318	694	—	0.062
69	C <sub>8</sub> H <sub>10</sub>	<i>p</i> -Xylene	-9.4655	1.4400E+03	1.9910E-02	-1.6994E-05	288	616	0.616	0.049
70	C <sub>5</sub> H <sub>5</sub> N	Pyridine	-6.8100	1.1496E+03	1.3229E-02	-1.1661E-05	232	620	0.898	0.058
71	C <sub>8</sub> H <sub>8</sub>	Styrene	-8.0291	1.2666E+03	1.6127E-02	-1.3475E-05	243	648	0.675	0.052
72	SO <sub>2</sub>	Sulfur dioxide	-2.6700	4.0670E+02	6.1440E-03	-1.2540E-05	200	431	0.257	0.039
73	C <sub>7</sub> H <sub>8</sub>	Toluene	-5.1649	8.1068E+02	1.0454E-02	-1.0488E-05	200	592	0.548	0.052
74	C <sub>4</sub> H <sub>6</sub> O <sub>2</sub>	Vinyl acetate	-9.0671	1.1863E+03	2.2663E-02	-2.3208E-05	250	524	0.403	0.050
75	H <sub>2</sub> O	Water	-10.2158	1.7925E+03	1.7730E-02	-1.2631E-05	273	643	0.911	0.056

 $\mu_{\text{liq}}$  – viscosity of gas,  $\mu\text{P}$ 

A, B, and C – regression coefficients for chemical compound

T – temperature, K

T<sub>min</sub> – minimum temperature, KT<sub>max</sub> – maximum temperature, K

**TABLE C-4 Heat Capacity of Gas**

No.	Formula	Substance	$C_p = A + BT + CT^2 + DT^3 + ET^4$			$(C_p - \text{joule}/(\text{mol K}), T - \text{K})$			
			A	B	C	D	E	$T_{\min}$	$T_{\max}$
1	C <sub>2</sub> H <sub>3</sub> C <sub>13</sub>	1,1,1-Trichloroethane	18.674	3.3443E-01	-3.4963E-04	1.8764E-07	-4.0744E-11	100	1500
2	C <sub>2</sub> H <sub>3</sub> C <sub>13</sub>	1,1,2-Trichloroethane	28.881	2.4893E-01	-1.7639E-04	5.2632E-08	-3.5668E-12	200	1500
3	C <sub>2</sub> H <sub>4</sub> C <sub>12</sub>	1,1-Dichloroethane	15.730	2.6124E-01	-2.1489E-04	9.5761E-08	-1.8004E-11	200	1500
4	C <sub>2</sub> H <sub>4</sub> C <sub>12</sub>	1,2-Dichloroethane	37.275	1.4362E-01	1.0378E-05	-7.8305E-08	2.8872E-11	200	1500
5	C <sub>4</sub> H <sub>6</sub>	1,3-Butadiene	18.835	2.0473E-01	6.2485E-05	-1.7148E-07	6.0858E-11	100	1500
6	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	1,4-Dioxane	-46.223	5.7263E-01	-3.8800E-04	1.1392E-07	-9.0669E-12	298	1500
7	C <sub>4</sub> H <sub>10</sub> O	1-Butanol ( <i>n</i> -Butanol)	8.157	4.1032E-01	-2.2645E-04	6.0372E-08	-6.2802E-12	200	2980
8	C <sub>4</sub> H <sub>8</sub>	1-Butene	24.915	2.0648E-01	5.9828E-05	-1.4166E-07	4.7053E-11	200	1500
9	C <sub>10</sub> H <sub>20</sub>	1-Decene	121.553	2.0974E-01	7.8760E-04	-8.6982E-07	2.6033E-10	150	1500
10	C <sub>9</sub> H <sub>10</sub>	1-Nonane ( <i>n</i> -Nonane)	29.687	6.6821E-01	-9.6492E-05	-2.0014E-07	8.2200E-11	200	1500
11	C <sub>8</sub> H <sub>16</sub>	1-Octene	56.266	4.0665E-01	1.5805E-04	-3.2277E-07	1.0600E-10	200	1500
12	C <sub>3</sub> H <sub>8</sub> O	1-Propanol ( <i>n</i> -Propanol)	31.507	2.3082E-01	-7.8983E-05	6.3696E-09	8.6908E-13	100	2980
13	C <sub>2</sub> H <sub>4</sub> O	Acetaldehyde	34.140	4.0020E-02	1.5634E-04	-1.6445E-07	4.7248E-11	100	1500
14	C <sub>2</sub> H <sub>4</sub> O <sub>2</sub>	Acetic acid	34.850	3.7626E-02	2.8311E-04	-3.0767E-07	9.2646E-11	50	1500
15	C <sub>4</sub> H <sub>6</sub> O <sub>3</sub>	Acetic anhydride	9.500	3.4425E-01	-8.6736E-05	-7.6769E-08	3.6721E-11	200	1500
16	C <sub>3</sub> H <sub>6</sub> O	Acetone	35.918	9.3896E-02	1.8730E-04	-2.1643E-07	6.3174E-11	100	1500
17	C <sub>3</sub> H <sub>4</sub> O <sub>2</sub>	Acrylic acid	7.755	2.9386E-01	-2.0878E-04	7.1591E-08	-9.0960E-12	250	1500
18	NH <sub>3</sub>	Ammonia	33.573	-1.2581E-02	8.8906E-05	-7.1783E-08	1.8569E-11	100	1500
19	C <sub>6</sub> H <sub>7</sub> N	Aniline	-22.062	5.7313E-01	-4.5651E-04	1.8410E-07	-2.9867E-11	200	1500
20	C <sub>6</sub> H <sub>6</sub>	Benzene	-31.368	4.7460E-01	-3.1137E-04	8.5237E-08	-5.0524E-12	200	1500
21	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Butyric acid	14.368	3.9591E-01	-1.8906E-04	-7.6462E-09	2.0812E-11	298	1200
22	CS <sub>2</sub>	Carbon disulfide	20.461	1.2299E-01	-1.6184E-04	1.0199E-07	-2.4444E-11	100	1500
23	CO <sub>2</sub>	Carbon dioxide	27.437	4.2315E-02	-1.9555E-05	3.9968E-09	-2.9872E-13	50	5000
24	CO	Carbon monoxide	29.556	-6.5807E-03	2.0130E-05	-1.2227E-08	2.2617E-12	60	1500
25	CCl <sub>4</sub>	Carbon tetrachloride	19.816	3.3311E-01	-5.0511E-04	3.4057E-07	-8.4249E-11	100	1500
26	Cl <sub>2</sub>	Chlorine	27.213	3.0426E-02	-3.3353E-05	1.5961E-08	-2.7021E-12	50	1500
27	CHCl <sub>3</sub>	Chloroform	22.487	1.9823E-01	-2.1676E-04	1.1636E-07	-2.4555E-11	100	1500
28	C <sub>6</sub> H <sub>12</sub>	Cyclohexane	13.783	2.0742E-01	5.3682E-04	-6.3012E-07	1.8988E-10	100	1500
29	C <sub>6</sub> H <sub>12</sub> O	Cyclohexanol	17.124	3.3700E-01	2.8176E-04	-4.2713E-07	1.3215E-10	200	1500
30	C <sub>3</sub> H <sub>6</sub>	Cyclopropane	21.172	6.3106E-02	2.9197E-04	-3.2708E-07	9.9730E-11	100	1500
31	CH <sub>2</sub> Cl <sub>2</sub>	Dichloromethane	26.694	8.3984E-02	8.9712E-06	-5.0924E-08	1.8726E-11	100	1500
32	C <sub>4</sub> H <sub>10</sub> O	Diethyl ether	35.979	2.8444E-01	-1.2673E-06	-1.0128E-07	3.4529E-11	200	1500
33	C <sub>5</sub> H <sub>10</sub> O	Diethyl ketone	49.800	2.6897E-01	5.0669E-05	-1.5227E-07	4.9510E-11	200	1500
34	C <sub>2</sub> H <sub>7</sub> N	Dimethylamine	30.638	1.0737E-01	1.5824E-04	-1.9418E-07	5.8509E-11	200	1500
35	C <sub>2</sub> H <sub>6</sub>	Ethane	28.146	4.3447E-02	1.8946E-04	-1.9082E-07	5.3349E-11	100	1500
36	C <sub>2</sub> H <sub>6</sub> O	Ethanol	27.091	1.1055E-01	1.0957E-04	-1.5046E-07	4.6601E-11	100	1500
37	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Ethyl acetate	69.848	8.2338E-02	3.7159E-04	-4.1129E-07	1.2369E-10	200	1500
38	C <sub>2</sub> H <sub>5</sub> Cl	Ethyl chloride	35.946	5.2294E-02	2.0321E-04	-2.2795E-07	6.9123E-11	100	1500
39	C <sub>8</sub> H <sub>10</sub>	Ethylbenzene	-20.527	5.9578E-01	-3.0849E-04	3.5621E-08	1.2409E-11	200	1500
40	C <sub>2</sub> H <sub>4</sub>	Ethylene	32.083	-1.4831E-02	2.4774E-04	-2.3766E-07	6.8274E-11	60	1500
41	C <sub>2</sub> H <sub>6</sub> O <sub>2</sub>	Ethylene glycol	48.218	1.9073E-01	-6.6117E-08	-1.8834E-08	1.2555E-11	200	1500
42	C <sub>2</sub> H <sub>4</sub> O	Ethylene oxide	30.827	-7.6041E-03	3.2347E-04	-3.2747E-07	9.7271E-11	50	1500
43	F <sub>2</sub>	Fluorine	27.408	1.2928E-02	7.0701E-06	-1.6302E-08	5.9789E-12	100	1500
44	C <sub>3</sub> H <sub>8</sub> O <sub>3</sub>	Glycerol	9.656	4.2826E-01	-2.6797E-04	3.1794E-08	2.7745E-11	298	1200
45	H <sub>2</sub>	Hydrogen	25.399	2.0178E-02	-3.8549E-05	3.1880E-08	-8.7585E-12	250	1500
46	HCl	Hydrogen chloride	29.244	-1.2615E-03	1.1210E-06	4.9676E-09	-2.4963E-12	50	1500
47	CHN	Hydrogen cyanide	25.766	3.7969E-02	-1.2416E-05	-3.2240E-09	2.2610E-12	100	1500
48	H <sub>2</sub> O <sub>2</sub>	Hydrogen peroxide	36.181	8.2657E-03	6.6420E-05	-6.9944E-08	2.0951E-11	100	1500
49	C <sub>4</sub> H <sub>10</sub>	<i>i</i> -Butane ( <i>iso</i> -Butane)	6.772	3.1447E-01	-1.0271E-04	-3.6849E-08	2.0429E-11	200	1500
50	CH <sub>4</sub>	Methane	34.942	-3.9957E-02	1.9184E-04	-1.5303E-07	3.9321E-11	50	1500
51	CH <sub>4</sub> O	Methanol	40.046	-3.8287E-02	2.4529E-04	-2.1679E-07	5.9909E-11	100	1500

**TABLE C-4—(continued)**

$C_p = A + BT + CT^2 + DT^3 + ET^4$					$(C_p\text{-joule}/(\text{mol K}), T\text{-K})$				
No.	Formula	Substance	A	B	C	D	E	$T_{\min}$	$T_{\max}$
52	CH <sub>3</sub> Br	Methyl bromide	29.146	2.4374E-02	1.0655E-04	-1.1324E-07	3.3241E-11	100	1500
53	C <sub>6</sub> H <sub>12</sub> O	Methyl isobutyl ketone	2.404	5.8495E-01	-3.7647E-04	1.2418E-07	-1.7051E-11	298	1500
54	CH <sub>5</sub> N	Methylamine	40.039	-1.5108E-02	2.5012E-04	-2.3336E-07	6.5582E-11	100	1500
55	C <sub>8</sub> H <sub>10</sub>	<i>m</i> -Xylene	-16.725	5.6424E-01	-2.6465E-04	1.3381E-08	1.5869E-11	200	1500
56	C <sub>10</sub> H <sub>8</sub>	Naphthalene	67.099	4.3239E-02	9.1740E-04	-1.0019E-06	3.0896E-10	50	1500
57	C <sub>4</sub> H <sub>10</sub>	<i>n</i> -Butane	20.056	2.8153E-01	-1.3143E-05	-9.4571E-08	3.4149E-11	200	1500
58	C <sub>10</sub> H <sub>22</sub>	<i>n</i> -Decane	31.780	7.4489E-01	-1.0945E-04	-2.2668E-07	9.3458E-11	200	1500
59	C <sub>7</sub> H <sub>16</sub>	<i>n</i> -Heptane	26.984	5.0387E-01	-4.4748E-05	-1.6835E-07	6.5183E-11	200	1500
60	C <sub>6</sub> H <sub>14</sub>	<i>n</i> -Hexane	25.924	4.1927E-01	-1.2491E-05	-1.55916E-07	5.8784E-11	200	1500
61	C <sub>6</sub> H <sub>5</sub> NO <sub>2</sub>	Nitrobenzene	-16.202	5.6182E-01	-3.9302E-04	1.0043E-07	-1.2252E-12	200	1500
62	N <sub>2</sub>	Nitrogen	29.342	-3.5395E-03	1.0076E-05	-4.3116E-09	2.5935E-13	50	1500
63	C <sub>9</sub> H <sub>20</sub>	<i>n</i> -Nonane	29.687	6.6821E-01	-9.6492E-05	-2.0014E-07	8.2200E-11	200	1500
64	C <sub>8</sub> H <sub>18</sub>	<i>n</i> -Octane	29.053	5.8016E-01	-5.7103E-05	-1.9548E-07	7.6614E-11	200	1500
65	C <sub>5</sub> H <sub>12</sub>	<i>n</i> -Pentane	26.671	3.2324E-01	4.2820E-05	-1.6639E-07	5.6036E-11	200	1500
66	O <sub>2</sub>	Oxygen	29.526	-8.8999E-03	3.8083E-05	-3.2629E-08	8.8607E-12	50	1500
67	C <sub>8</sub> H <sub>10</sub>	<i>o</i> -Xylene	0.182	5.1344E-01	-2.0212E-04	-2.1615E-08	2.3212E-11	200	1500
68	C <sub>6</sub> H <sub>6</sub> O	Phenol	4.408	3.6338E-01	-6.0417E-05	-1.2794E-07	5.5287E-11	100	1500
69	C <sub>8</sub> H <sub>10</sub>	<i>p</i> -Xylene	-17.360	5.6470E-01	-2.6293E-04	1.1217E-08	1.6544E-11	200	1500
70	C <sub>5</sub> H <sub>5</sub> N	Pyridine	23.262	1.1251E-01	3.7351E-04	-4.5402E-07	1.4286E-10	50	1500
71	C <sub>8</sub> H <sub>8</sub>	Styrene	71.201	5.4767E-02	6.4793E-04	-6.9875E-07	2.1232E-10	100	1500
72	SO <sub>2</sub>	Sulfur dioxide	29.637	3.4735E-02	9.2903E-06	-2.9885E-08	1.0937E-11	100	1500
73	C <sub>7</sub> H <sub>8</sub>	Toluene	-24.097	5.2187E-01	-2.9827E-04	6.1220E-08	1.2576E-12	200	1500
74	C <sub>4</sub> H <sub>6</sub> O <sub>2</sub>	Vinyl acetate	27.664	2.3366E-01	6.2106E-05	-1.6972E-07	5.7917E-11	100	1500
75	H <sub>2</sub> O	Water	33.933	-8.4186E-03	2.9906E-05	-1.7825E-08	3.6934E-12	100	1500

**$C_p$  – heat capacity of ideal gas, joule/(mol K)**

**$A, B, C, D$ , and  $E$  – regression coefficients for chemical compound**

**$T$  – temperature, K**

**$T_{\min}$  – minimum temperature, K**

**$T_{\max}$  – maximum temperature, K**

**TABLE C-5 Heat Capacity of Liquids**

$C_p = A + BT + CT^2 + DT^3$							$T_{\min}$	$T_{\max}$	$C_p$ at 25°C
No.	Formula	Substance	A	B	C	D			
1	$C_2H_3Cl_3$	1,1,1-Trichloroethane	11.142	1.0501E+00	-3.0826E-03	3.5983E-06	244	491	145.56
2	$C_2H_3Cl_3$	1,1,2-Trichloroethane	34.934	8.5054E-01	-2.3306E-03	2.6455E-06	238	542	151.46
3	$C_2H_4Cl_2$	1,1-Dichloroethane	57.325	5.6014E-01	-1.8136E-03	2.5617E-06	177	471	131.00
4	$C_2H_4Cl_2$	1,2-Dichloroethane	26.310	7.7555E-01	-2.2271E-03	2.6107E-06	238	505	128.77
5	$C_4H_6$	1,3-Butadiene	34.680	7.3205E-01	-2.8426E-03	4.6035E-06	165	383	122.26
6	$C_4H_8O_2$	1,4-Dioxane	-20.729	1.2913E+00	-3.5408E-03	3.5408E-06	286	528	153.23
7	$C_4H_{10}O$	1-Butanol ( <i>n</i> -Butanol)	83.877	5.6628E-01	-1.7208E-03	2.2780E-06	185	507	160.12
8	$C_4H_8$	1-Butene	74.597	3.3434E-04	-1.3914E-03	3.0241E-06	89	378	130.74
9	$C_{10}H_{20}$	1-Decene	137.962	1.1934E+00	-3.2863E-03	3.9390E-06	208	555	306.05
10	$C_9H_{20}$	1-Nonane ( <i>n</i> -Nonane)	98.040	1.3538E+00	-3.8058E-03	4.4991E-06	221	536	282.60
11	$C_8H_{16}$	1-Octene	119.984	8.3332E-01	-2.5321E-03	3.4745E-06	172	510	235.43
12	$C_3H_6O$	1-Propanol ( <i>n</i> -Propanol)	88.080	4.0224E-01	-1.3032E-03	1.9677E-06	148	483	144.32
13	$C_2H_4O$	Acetaldehyde	45.056	4.4853E-01	-1.6607E-03	2.7000E-06	151	415	102.72
14	$C_2H_4O_2$	Acetic acid	-18.944	1.0971E+00	2.8921E-03	2.9275E-06	291	533	128.66
15	$C_4H_6O_3$	Acetic anhydride	71.831	8.8879E-01	-2.6534E-03	3.3501E-06	201	512	189.75
16	$C_3H_6O$	Acetone	46.878	6.2652E-01	-2.0761E-03	2.9583E-06	179	457	127.53
17	$C_3H_4O_2$	Acrylic acid	-18.242	1.2106E+00	-3.1160E-03	3.1409E-06	241	617	123.05
18	$NH_3$	Ammonia	-182.157	3.3618E+00	-1.4398E-02	2.0371E-05	195	385	80.16
19	$C_6H_7N$	Aniline	63.288	9.8960E-01	-2.3583E-03	2.3296E-06	268	629	210.44
20	$C_6H_6$	Benzene	-31.663	1.3043E+00	-3.6078E-03	3.8243E-06	280	506	137.87
21	$C_4H_8O_2$	Butyric acid	28.210	1.1040E+00	-2.8523E-03	2.9528E-06	269	565	182.09
22	$CS_2$	Carbon disulfide	39.938	2.3565E-01	-7.2098E-04	1.0443E-06	163	497	73.79
23	$CO_2$	Carbon dioxide	-338.956	5.2796E+00	-2.3279E-02	3.5980E-05	218	274	-
24	CO	Carbon monoxide	-19.312	2.5072E+00	-2.8970E-02	1.2745E-04	69	120	-
25	$CCl_4$	Carbon tetrachloride	9.671	9.3363E-01	-2.6768E-03	3.0425E-06	251	501	130.72
26	$Cl_2$	Chlorine	127.601	-6.0215E-01	1.5776E-03	-5.3099E-07	172	396	74.23
27	$CHCl_3$	Chloroform	28.296	6.5897E-01	-2.0353E-03	2.5901E-06	211	483	112.49
28	$C_6H_{12}$	Cyclohexane	-44.417	1.6016E+00	-4.4676E-03	4.7582E-06	281	498	162.07
29	$C_6H_{12}O$	Cyclohexanol	-47.321	1.9131E+00	-4.8388E-03	4.7281E-06	298	563	218.25
30	$C_3H_6$	Cyclopropane	30.543	5.0198E-01	-2.1040E-03	3.7444E-06	147	358	92.42
31	$CH_2Cl_2$	Dichloromethane	38.941	4.9008E-01	-1.6224E-03	2.3069E-06	179	459	101.98
32	$C_4H_{10}O$	Diethyl ether	75.939	7.7335E-01	-2.7936E-03	4.4383E-06	158	420	175.81
33	$C_5H_{10}O$	Diethyl ketone	26.231	1.2822E+00	-3.7449E-03	4.3816E-06	235	505	191.76
34	$C_2H_7N$	Dimethylamine	36.962	9.5817E-01	-3.5846E-03	5.3990E-06	182	394	147.08
35	$C_2H_6$	Ethane	38.332	4.1006E-01	-2.3024E-03	5.9347E-06	91	275	-
36	$C_2H_6O$	Ethanol	59.342	3.6358E-01	-1.2164E-03	1.8030E-06	160	465	107.40
37	$C_4H_8O_2$	Ethyl acetate	62.832	8.4097E-01	-2.6998E-03	3.6631E-06	191	471	170.66
38	$C_2H_5Cl$	Ethyl chloride	60.180	3.4553E-01	-1.2983E-03	2.1963E-06	138	414	106.00
39	$C_8H_{10}$	Ethylbenzene	102.11	5.5959E-01	-1.5609E-03	2.0149E-06	179	555	183.60
40	$C_2H_4$	Ethylene	25.597	5.7078E-01	-3.3620E-03	8.4120E-06	105	254	-
41	$C_2H_6O_2$	Ethylene glycol	75.878	6.4182E-01	-1.6493E-03	1.6937E-06	261	581	165.52
42	$C_2H_4O$	Ethylene oxide	35.720	4.2908E-01	-1.5473E-03	2.4070E-06	162	422	89.90
43	$F_2$	Fluorine	83.829	-7.8518E-01	5.2305E-03	4.6617E-06	53	137	-
44	$C_3H_8O_3$	Glycerol	132.145	8.6007E-01	-1.9745E-03	1.8068E-06	292	651	260.94
45	$H_2$	Hydrogen	50.607	-6.1136E+00	3.0930E-01	-4.1480E-03	14	32	-
46	HCl	Hydrogen chloride	73.993	-1.2946E-01	-7.8980E-05	2.6409E-06	165	308	98.37
47	CHN	Hydrogen cyanide	-123.155	1.7769E+00	-5.8083E-03	6.9129E-06	261	411	73.52
48	$H_2O_2$	Hydrogen peroxide	-15.248	6.7693E-01	-1.4948E-03	1.2018E-06	273	694	85.55
49	$C_4H_{10}$	<i>i</i> -Butane ( <i>iso</i> -Butane)	71.791	4.8472E-01	-2.0519E-03	4.0634E-06	115	367	141.61
50	$CH_4$	Methane	-0.018	1.1982E+00	-9.8722E-03	3.1670E-05	92	172	-
51	$CH_3OH$	Methanol	40.152	3.1046E-01	-1.0291E-03	1.4598E-06	176	461	79.93

**TABLE C-5—(continued)**

$C_p = A + BT + CT^2 + DT^3$							$C_p$ at 25°C		
No.	Formula	Substance	A	B	C	D	$T_{\min}$	$T_{\max}$	
52	CH <sub>3</sub> Br	Methyl bromide	25.042	4.9312E-01	-1.7627E-03	2.5993E-06	181	420	85.16
53	CH <sub>3</sub> Cl	Methyl chloride	11.381	6.2328E-01	-2.4353E-03	3.8333E-06	176	375	82.33
54	C <sub>6</sub> H <sub>12</sub> O	Methyl isobutyl ketone	96.184	8.5227E-01	-2.5379E-03	3.3066E-06	190	514	212.42
55	CH <sub>5</sub> N	Methylamine	13.565	9.0836E-01	-3.4881E-03	5.2770E-06	181	3887	114.19
56	C <sub>8</sub> H <sub>10</sub>	<i>m</i> -Xylene	70.916	8.0450E-01	-2.1885E-03	2.5061E-06	226	555	182.66
57	C <sub>10</sub> H <sub>8</sub>	Naphthalene	-30.842	1.5362E+00	-3.2492E-03	2.6568E-06	354	674	—
58	C <sub>4</sub> H <sub>10</sub>	<i>n</i> -Butane	62.873	5.8913E-01	-2.3588E-03	4.2257E-06	136	383	140.84
59	C <sub>10</sub> H <sub>22</sub>	<i>n</i> -Decane	79.741	1.6926E+00	-4.5287E-03	4.9769E-06	244	557	313.73
60	C <sub>7</sub> H <sub>16</sub>	<i>n</i> -Heptane	101.121	9.7739E-01	-3.0712E-03	4.1844E-06	184	486	230.42
61	C <sub>6</sub> H <sub>14</sub>	<i>n</i> -Hexane	78.848	8.8729E-01	-2.9482E-03	4.1999E-06	179	457	192.63
62	C <sub>6</sub> H <sub>5</sub> NO <sub>2</sub>	Nitrobenzene	51.773	9.1277E-01	-2.1098E-03	2.0093E-06	280	647	189.62
63	N <sub>2</sub>	Nitrogen	76.452	-3.5226E-01	-2.6690E-03	5.0057E-05	64	120	—
64	C <sub>9</sub> H <sub>20</sub>	<i>n</i> -Nonane	98.040	1.3538E+00	-3.8058E-03	4.4991E-06	221	536	282.60
65	C <sub>8</sub> H <sub>18</sub>	<i>n</i> -Octane	82.736	1.3043E+00	-3.8254E-03	4.6459E-06	217	512	254.71
66	C <sub>5</sub> H <sub>12</sub>	<i>n</i> -Pentane	80.641	6.2195E-01	-2.2682E-03	3.7423E-06	144	423	163.64
67	O <sub>2</sub>	Oxygen	46.432	3.9506E-01	-7.0522E-03	3.9897E-05	54	147	—
68	C <sub>8</sub> H <sub>10</sub>	<i>o</i> -Xylene	56.460	9.4926E-01	-2.4902E-03	2.6838E-06	249	567	189.25
69	C <sub>6</sub> H <sub>6</sub> O	Phenol	38.622	1.0983E+00	-2.4897E-03	2.2802E-06	315	625	—
70	C <sub>8</sub> H <sub>10</sub>	<i>p</i> -Xylene	-11.035	1.5158E+00	-3.9039E-03	3.9193E-06	287	555	197.75
71	C <sub>5</sub> H <sub>5</sub> N	Pyridine	37.150	6.9497E-01	-1.8749E-03	2.1188E-06	233	558	133.85
72	C <sub>8</sub> H <sub>8</sub>	Styrene	66.737	8.4051E-01	-2.1615E-03	2.3324E-06	244	583	187.00
73	SO <sub>2</sub>	Sulfur dioxide	203.445	-1.0537E+00	2.6113E-03	-1.0697E-06	198	409	93.06
74	C <sub>7</sub> H <sub>8</sub>	Toluene	83.703	5.1666E-01	-1.4910E-03	1.9725E-06	179	533	157.49
75	C <sub>4</sub> H <sub>6</sub> O <sub>2</sub>	Vinyl acetate	63.910	7.0656E-01	-2.2832E-03	3.1788E-06	181	472	155.86
76	H <sub>2</sub> O	Water	92.053	-3.9953E-02	-2.1103E-04	5.3469E-07	273	615	75.55

*C<sub>p</sub>* – heat capacity of liquid, J/(mol K)  
*A, B, C, and D* – regression coefficients for chemical compound  
*T* – temperature, K  
*T<sub>min</sub>* – minimum temperature, K  
*T<sub>max</sub>* – maximum temperature, K

**TABLE C-6 Thermal Conductivity of Gas**

No.	Formula	Substance	$K_{\text{gas}} = A + BT + CT^2$			$(K_{\text{gas}} - \text{W}/(\text{m K}), T - \text{K})$				
			A	B	C	$T_{\text{min}}$	$T_{\text{max}}$	$K_{\text{gas}}$ at 25°C	$K_{\text{gas}}$ at $T_{\text{min}}$	$K_{\text{gas}}$ at $T_{\text{max}}$
1	$\text{C}_2\text{H}_3\text{Cl}_3$	1,1,1-Trichloroethane	-0.00070	3.9347E-06	3.3468E-09	347	997	—	0.00107	0.00655
2	$\text{C}_2\text{H}_3\text{Cl}_3$	1,1,2-Trichloroethane	-0.00609	3.2555E-05	3.0529E-08	387	997	—	0.01108	0.05674
3	$\text{C}_2\text{H}_4\text{Cl}_2$	1,1-Dichloroethane	-0.00739	4.4752E-05	3.3174E-08	330	990	—	0.01099	0.06943
4	$\text{C}_2\text{H}_4\text{Cl}_2$	1,2-Dichloroethane	-0.00682	4.0081E-05	3.1925E-08	357	997	—	0.01156	0.06487
5	$\text{C}_4\text{H}_6$	1,3-Butadiene	-0.00085	7.1537E-06	1.6202E-07	250	850	0.01569	0.01106	0.12229
6	$\text{C}_4\text{H}_8\text{O}_2$	1,4-Dioxane	-0.01642	8.0095E-05	1.7582E-08	285	1000	0.00902	0.00784	0.08126
7	$\text{C}_4\text{H}_{10}\text{O}$	1-Butanol ( <i>n</i> -Butanol)	0.01783	-4.8291E-05	1.6334E-07	371	713	—	0.02240	0.06644
8	$\text{C}_4\text{H}_8$	1-Butene	-0.00293	3.0205E-05	1.0192E-07	225	800	0.01514	0.00903	0.08646
9	$\text{C}_{10}\text{H}_{20}$	1-Decene	-0.01118	6.2751E-05	2.8360E-08	444	1000	—	0.02227	0.07993
10	$\text{C}_9\text{H}_{20}$	1-Nonane ( <i>n</i> -Nonane)	-0.00655	3.2637E-05	7.7150E-08	449	678	—	0.02366	0.05104
11	$\text{C}_8\text{H}_{16}$	1-Octene	-0.00463	2.8829E-05	8.4175E-08	394	500	—	0.01980	0.03083
12	$\text{C}_3\text{H}_8\text{O}$	1-Propanol ( <i>n</i> -Propanol)	-0.00333	2.8691E-05	1.0222E-07	372	720	—	0.02149	0.07032
13	$\text{C}_2\text{H}_4\text{O}$	Acetaldehyde	-0.00181	2.1187E-05	8.0192E-08	200	700	0.01164	0.00564	0.05231
14	$\text{C}_2\text{H}_4\text{O}_2$	Acetic acid	0.00234	-6.5956E-06	1.1569E-07	295	687	0.01066	0.01046	0.05241
15	$\text{C}_4\text{H}_6\text{O}_3$	Acetic anhydride	-0.00846	5.2818E-05	1.7355E-08	413	993	—	0.01631	0.06110
16	$\text{C}_3\text{H}_6\text{O}$	Acetone	-0.00084	8.7475E-06	1.0678E-07	273	572	0.01126	0.00951	0.03910
17	$\text{C}_3\text{H}_4\text{O}_2$	Acrylic acid	-0.00889	6.0453E-05	1.2049E-08	414	1000	—	0.01820	0.06361
18	$\text{NH}_3$	Ammonia	0.00457	2.3239E-05	1.4810E-07	200	700	0.02466	0.01514	0.09341
19	$\text{C}_6\text{H}_5\text{N}$	Aniline	-0.01796	8.3464E-05	1.5022E-09	458	1000	—	0.02058	0.06701
20	$\text{C}_6\text{H}_6$	Benzene	-0.00565	3.4493E-05	6.9298E-08	325	700	—	0.01288	0.05246
21	$\text{C}_4\text{H}_8\text{O}_2$	Butyric acid	0.12421	-3.6238E-04	3.7750E-07	523	707	—	0.03794	0.05670
22	$\text{CS}_2$	Carbon disulfide	-0.00239	3.2574E-05	2.1193E-10	273	999	0.00734	0.00652	0.03036
23	$\text{CO}_2$	Carbon dioxide	-0.01200	1.0208E-04	-2.2403E-08	195	1500	0.01644	0.00705	0.09071
24	CO	Carbon monoxide	0.00158	8.2511E-05	-1.9081E-08	70	1250	0.02448	0.00726	0.07490
25	$\text{CC}_{14}$	Carbon tetrachloride	-0.00070	2.2065E-05	6.7913E-09	255	556	0.00648	0.00537	0.01367
26	$\text{Cl}_{12}$	Chlorine	-0.00194	3.8300E-05	-6.3523E-09	200	1000	0.00891	0.00547	0.03001
27	$\text{CHC}_{13}$	Chloroform	-0.00019	2.2269E-05	1.2257E-09	273	573	0.00656	0.00598	0.01297
28	$\text{C}_6\text{H}_{12}$	Cyclohexane	-0.00159	-1.7494E-07	1.4588E-07	325	650	—	0.01376	0.05993
29	$\text{C}_6\text{H}_{12}\text{O}$	Cyclohexanol	-0.02696	1.1162E-04	1.2497E-08	434	1000	—	0.02384	0.09716
30	$\text{C}_3\text{H}_6$	Cyclopropane	-0.00430	3.7668E-05	1.0953E-07	240	1000	0.01667	0.01105	0.14290
31	$\text{CH}_2\text{Cl}_2$	Dichloromethane	-0.00122	1.8440E-05	3.5385E-08	250	500	0.00742	0.00560	0.01685
32	$\text{C}_4\text{H}_{10}\text{O}$	Diethyl ether	-0.00032	1.6530E-05	1.1709E-07	200	600	0.01502	0.00767	0.05175
33	$\text{C}_5\text{H}_{10}\text{O}$	Diethyl ketone	0.00004	-2.7408E-07	1.2105E-07	273	1000	0.01072	0.00899	0.12082
34	$\text{C}_2\text{H}_7\text{N}$	Dimethylamine	-0.01559	8.3374E-05	6.3822E-08	273	990	0.01494	0.01193	0.12950
35	$\text{C}_2\text{H}_6$	Ethane	-0.01936	1.2547E-04	3.8298E-08	225	825	0.02145	0.01081	0.11022
36	$\text{C}_2\text{H}_5\text{O}$	Ethanol	-0.00556	4.3620E-05	8.5033E-08	351	991	—	0.02023	0.12118
37	$\text{C}_4\text{H}_8\text{O}_2$	Ethyl acetate	0.00207	-4.8558E-06	1.1222E-07	273	1000	0.01060	0.00911	0.10943
38	$\text{C}_2\text{H}_5\text{Cl}$	Ethyl chloride	-0.00291	3.1284E-05	5.5316E-08	273	773	0.01133	0.00975	0.05433
39	$\text{C}_8\text{H}_{10}$	Ethylbenzene	-0.00797	4.0572E-05	6.7289E-08	400	825	—	0.01903	0.07130
40	$\text{C}_2\text{H}_4$	Ethylene	-0.00123	3.6219E-05	1.2459E-07	150	750	0.02064	0.00701	0.09602
41	$\text{C}_2\text{H}_6\text{O}_2$	Ethylene glycol	-0.01076	7.9631E-05	5.7243E-09	470	990	—	0.02793	0.07369
42	$\text{C}_2\text{H}_4\text{O}$	Ethylene oxide	0.01612	-7.3460E-05	2.1215E-07	273	673	0.01308	0.01188	0.06277
43	$\text{F}_2$	Fluorine	-0.00076	9.5961E-05	-2.1800E-08	70	700	0.02591	0.00585	0.05573
44	$\text{C}_3\text{H}_8\text{O}_3$	Glycerol	-0.00842	4.4977E-05	2.4839E-08	563	993	—	0.02478	0.06073
45	$\text{H}_2$	Hydrogen	0.03951	4.5918E-04	-6.4933E-08	150	1500	0.17064	0.10693	0.58218
46	$\text{HCl}$	Hydrogen chloride	0.00119	4.4775E-05	2.0997E-10	159	1000	0.01456	0.00831	0.04617
47	CHN	Hydrogen cyanide	-0.00136	3.7886E-05	2.6807E-08	260	360	0.01232	0.01030	0.01575
48	$\text{H}_2\text{O}_2$	Hydrogen peroxide	-0.00858	8.6933E-05	-6.2970E-09	275	1200	0.01678	0.01485	0.08667
49	$\text{C}_4\text{H}_{10}$	<i>i</i> -Butane ( <i>iso</i> -Butane)	-0.00115	1.4943E-05	1.4921E-07	261	673	0.01657	0.01291	0.07649
50	$\text{CH}_4$	Methane	-0.00935	1.4028E-04	3.3180E-08	97	1400	0.03542	0.00457	0.25207

**TABLE C-6—(continued)**

$K_{\text{gas}} = A + BT + CT^2$						$(K_{\text{gas}} - W/(m \text{ K}), T - \text{K})$				
No.	Formula	Substance	A	B	C	$T_{\text{min}}$	$T_{\text{max}}$	$K_{\text{gas}}$ at $25^\circ \text{C}$	$K_{\text{gas}}$ at $T_{\text{min}}$	$K_{\text{gas}}$ at $T_{\text{max}}$
51	CH <sub>3</sub> Br	Methyl bromide	-0.00421	2.7182E-05	3.9758E-08	273	825	0.00743	0.00617	0.04528
52	CH <sub>3</sub> Cl	Methyl chloride	-0.00185	2.0296E-05	7.3234E-08	213	750	0.01071	0.00580	0.05457
53	C <sub>6</sub> H <sub>12</sub> O	Methyl isobutyl ketone	0.00209	-7.7952E-06	1.2993E-07	390	1000	—	0.01881	0.12422
54	CH <sub>5</sub> N	Methylamine	-0.01136	8.0400E-05	5.8607E-08	267	800	0.01782	0.01428	0.09047
55	C <sub>8</sub> H <sub>10</sub>	<i>m</i> -Xylene	-0.00375	2.9995E-05	7.4603E-08	400	825	—	0.02018	0.07177
56	C <sub>10</sub> H <sub>8</sub>	Naphthalene	-0.02306	9.2610E-05	4.4577E-10	353	1259	—	0.00969	0.09424
57	C <sub>4</sub> H <sub>10</sub>	<i>n</i> -Butane	-0.00182	1.9396E-05	1.3818E-07	225	675	0.01625	0.00954	0.07423
58	C <sub>10</sub> H <sub>22</sub>	<i>n</i> -Decane	-0.00113	8.1090E-06	9.6092E-08	470	700	—	0.02391	0.05163
59	C <sub>7</sub> H <sub>16</sub>	<i>n</i> -Heptane	-0.00172	1.6565E-05	1.0525E-07	250	750	0.01257	0.00900	0.06991
60	C <sub>6</sub> H <sub>14</sub>	<i>n</i> -Hexane	-0.00200	7.7788E-06	1.3824E-07	290	480	0.01261	0.01188	0.03358
61	C <sub>6</sub> H <sub>5</sub> NO <sub>2</sub>	Nitrobenzene	—	—	—	—	—	—	—	—
62	N <sub>2</sub>	Nitrogen	0.00309	7.5930E-05	-1.1014E-08	78	1500	0.02475	0.00895	0.09220
63	C <sub>9</sub> H <sub>20</sub>	<i>n</i> -Nonane	-0.00655	3.2637E-05	7.7150E-08	449	678	—	0.02366	0.05104
64	C <sub>8</sub> H <sub>18</sub>	<i>n</i> -Octane	-0.00213	1.8456E-05	9.4775E-08	300	800	—	0.01194	0.07329
65	C <sub>5</sub> H <sub>12</sub>	<i>n</i> -Pentane	-0.00137	1.8081E-05	1.2136E-07	225	480	0.01481	0.00884	0.03527
66	O <sub>2</sub>	Oxygen	0.00121	8.6157E-05	-1.3346E-08	80	1500	0.02571	0.00802	0.10042
67	C <sub>8</sub> H <sub>10</sub>	<i>o</i> -Xylene	-0.00979	7.4087E-05	1.8418E-08	400	825	—	0.02279	0.06387
68	C <sub>6</sub> H <sub>6</sub> O	Phenol	-0.00552	4.4952E-05	3.9900E-08	455	995	—	0.02319	0.07871
69	C <sub>8</sub> H <sub>10</sub>	<i>p</i> -Xylene	-0.00870	4.7349E-05	5.8829E-08	400	825	—	0.01965	0.07040
70	C <sub>3</sub> H <sub>8</sub>	Propane	-0.00869	6.6409E-05	7.8760E-08	233	773	0.01811	0.01106	0.08971
71	C <sub>5</sub> H <sub>5</sub> N	Pyridine	-0.00469	1.6151E-05	9.4006E-08	388	998	—	0.01573	0.10506
72	C <sub>8</sub> H <sub>8</sub>	Styrene	-0.00712	4.5538E-05	3.9529E-08	273	973	0.00997	0.00826	0.07461
73	SO <sub>2</sub>	Sulfur dioxide	-0.00394	4.4847E-05	2.1066E-09	198	1000	0.00962	0.00502	0.04301
74	C <sub>7</sub> H <sub>8</sub>	Toluene	-0.00776	4.4905E-05	6.4514E-08	350	800	—	0.01586	0.06945
75	C <sub>4</sub> H <sub>6</sub> O <sub>2</sub>	Vinyl acetate	-0.00846	5.8704E-08	1.7678E-08	346	996	—	0.01397	0.06755
76	H <sub>2</sub> O	Water	0.00053	4.7093E-05	4.9551E-08	275	1073	0.01898	0.01723	0.10811

**TABLE C-7a Thermal Conductivity of Liquids**

No.	Formula	Substance	$\log_{10} k_{\text{liq}} = A + B [1 - T/C]^{27}$								
			A	B	C	$T_{\text{min}}$	$T_{\text{max}}$	$k_{\text{liq}}$ at 25°C	$k_{\text{liq}}$ at $T_{\text{min}}$	$k_{\text{liq}}$ at $T_{\text{max}}$	
1	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,1-Trichloroethane	-1.7352	0.9286	545.00	243	518	0.1012	0.1096	0.0184	
2	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,2-Trichloroethane	-2.6567	2.1686	602.00	237	572	0.1340	0.1601	0.0022	
3	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,1-Dichloroethane	-1.7265	0.9930	523.00	176	497	0.1132	0.1412	0.0188	
4	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,2-Dichloroethane	-1.6509	0.9701	561.00	237	533	0.1350	0.1476	0.0223	
5	C <sub>4</sub> H <sub>6</sub>	1,3-Butadiene	-1.6512	0.9899	425.37	164	404	0.1122	0.1592	0.0223	
6	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	1,4-Dioxane	-2.1607	1.6668	587.00	285	558	0.1586	0.1578	0.0069	
7	C <sub>4</sub> H <sub>10</sub> O	1-Butanol ( <i>n</i> -Butanol)	-1.3120	0.6190	562.93	184	535	0.1538	0.1725	0.0488	
8	C <sub>4</sub> H <sub>8</sub>	1-Butene	-1.6539	0.9786	419.59	88	399	0.1078	0.1809	0.0222	
9	C <sub>10</sub> H <sub>20</sub>	1-Decene	-1.7491	1.0443	617.05	207	586	0.1305	0.1489	0.0293	
10	C <sub>9</sub> H <sub>20</sub>	1-Nonane ( <i>n</i> -Nonane)	-1.7865	1.1033	595.65	220	566	0.1313	0.1486	0.0163	
11	C <sub>8</sub> H <sub>16</sub>	1-Octene	-1.6274	0.9136	566.60	171	538	0.1290	0.1555	0.0236	
12	C <sub>3</sub> H <sub>8</sub> O	1-Propanol ( <i>n</i> -Propanol)	-1.2131	0.5097	536.71	147	510	0.1553	0.1776	0.0612	
13	C <sub>2</sub> H <sub>4</sub> O	Acetaldehyde	-1.4826	0.9821	461.00	150	438	0.1766	0.2447	0.0329	
14	C <sub>2</sub> H <sub>4</sub> O <sub>2</sub>	Acetic acid	-1.2836	0.5893	592.71	290	563	0.1581	0.1569	0.0520	
15	C <sub>4</sub> H <sub>6</sub> O <sub>3</sub>	Acetic anhydride	-1.3593	0.7106	569.15	200	541	0.1643	0.1834	0.0437	
16	C <sub>3</sub> H <sub>6</sub> O	Acetone	-1.3857	0.7643	508.20	178	483	0.1615	0.1925	0.0411	
17	C <sub>3</sub> H <sub>4</sub> O <sub>2</sub>	Acrylic acid	-1.6101	0.9742	615.00	287	584	0.1570	0.1559	0.0245	
18	NH <sub>3</sub>	Ammonia	1.1606	-2.2840E-03	3.1245E-18	220	400	0.480	0.658	0.247	
19	C <sub>6</sub> H <sub>7</sub> N	Aniline	-1.3485	0.6888	699.00	267	664	0.1734	0.1763	0.0448	
20	C <sub>6</sub> H <sub>6</sub>	Benzene	-1.6846	1.0520	562.16	279	534	0.1456	0.1469	0.0207	
21	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Butyric acid	-1.3420	0.6161	628.00	268	597	0.1481	0.1505	0.0455	
22	CS <sub>2</sub>	Carbon disulfide	-1.2917	0.5809	552.00	162	524	0.1491	0.1702	0.0511	
23	CO <sub>2</sub>	Carbon dioxide	-1.3679	0.8092	304.19	217	289	—	0.1500	0.0429	
24	CO	Carbon monoxide	-1.7115	1.1359	132.59	68	126	—	0.1580	0.0194	
25	CCl <sub>4</sub>	Carbon tetrachloride	-1.8791	1.0875	556.35	250	529	0.0987	0.1063	0.0132	
26	Cl <sub>2</sub>	Chlorine	0.2246	-6.4000E-05	-7.8800E-07	172	410	0.135	0.190	0.066	
27	CHCl <sub>3</sub>	Chloroform	-1.5271	0.7577	536.40	210	510	0.1185	0.1330	0.0297	
28	C <sub>6</sub> H <sub>12</sub>	Cyclohexane	-1.6817	0.9649	553.54	280	526	0.1236	0.1244	0.0208	
29	C <sub>6</sub> H <sub>12</sub> O	Cyclohexanol	-1.3475	0.5719	625.15	297	594	0.1342	0.1323	0.0449	
30	C <sub>3</sub> H <sub>6</sub>	Cyclopropane	-1.5958	0.9876	397.91	146	378	0.1173	0.1833	0.0254	
31	CH <sub>2</sub> Cl <sub>2</sub>	Dichloromethane	-1.8069	1.2216	510.00	178	485	0.1392	0.1841	0.0156	
32	C <sub>4</sub> H <sub>10</sub> O	Diethyl ether	-1.5629	0.9357	466.70	157	443	0.1369	0.1832	0.0274	
33	C <sub>5</sub> H <sub>10</sub> O	Diethyl ketone	-1.5868	0.9252	560.95	234	533	0.1439	0.1576	0.0259	
34	C <sub>2</sub> H <sub>7</sub> N	Dimethylamine	-1.2557	0.6069	437.65	181	416	0.1521	0.1819	0.0555	
35	C <sub>2</sub> H <sub>6</sub>	Ethane	-1.3474	0.7003	305.42	90	290	—	0.1916	0.0449	
36	C <sub>2</sub> H <sub>6</sub> O	Ethanol	-1.3172	0.6987	516.25	159	490	0.1694	0.2030	0.0482	
37	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Ethyl acetate	-1.6938	1.0862	523.30	190	497	0.1445	0.1789	0.0202	
38	C <sub>2</sub> H <sub>5</sub> Cl	Ethyl chloride	-2.0001	1.4496	460.35	137	437	0.1191	0.2003	0.0100	
39	C <sub>8</sub> H <sub>10</sub>	Ethylbenzene	1.7498	1.0437	617.17	178	586	0.1302	0.1554	0.0178	
40	C <sub>2</sub> H <sub>4</sub>	Ethylene	-1.3314	0.8527	282.36	104	268	—	0.2568	0.0466	
41	C <sub>2</sub> H <sub>6</sub> O <sub>2</sub>	Ethylene glycol	-0.5918	—	645.00	260	613	0.2560	0.2560	0.2560	
42	C <sub>2</sub> H <sub>4</sub> O	Ethylene oxide	-1.4656	0.8777	469.15	161	446	0.1557	0.2026	0.0342	
43	F <sub>2</sub>	Fluorine	0.2758	-1.6297E-03	-3.7475E-18	53	130	—	0.189	0.064	
44	C <sub>3</sub> H <sub>8</sub> O <sub>3</sub>	Glycerol	-0.3550	-0.2097	723.00	293	550	0.2916	0.2994	0.4416	
45	H <sub>2</sub>	Hydrogen	-0.1433	2.3627E-02	-5.1480E-04	14	33	—	0.0866	0.0737	
46	HCl	Hydrogen chloride	0.8045	-2.1020E-03	-2.3238E-16	273	323	0.178	0.231	0.126	
47	CHN	Hydrogen cyanide	-1.4117	10.0351	456.65	260	434	0.2256	0.2429	0.0388	
48	H <sub>2</sub> O <sub>2</sub>	Hydrogen peroxide	0.4425	-1.8406E-04	-3.8824E-07	273	657	0.353	0.363	0.154	
49	C <sub>4</sub> H <sub>10</sub>	<i>i</i> -Butane ( <i>iso</i> -Butane)	-1.6862	0.9802	408.14	114	388	0.0972	0.1589	0.0206	
50	CH <sub>4</sub>	Methane	-1.0976	0.5387	190.58	91	181	—	0.2206	0.0799	

**TABLE C-7a—(continued)**

$\log_{10} k_{\text{liq}} = A + B[1 - T/C]^{27}$										
No.	Formula	Substance	A	B	C	$T_{\text{min}}$	$T_{\text{max}}$	$k_{\text{liq}}$ at 25°C	$k_{\text{liq}}$ at $T_{\text{min}}$	$k_{\text{liq}}$ at $T_{\text{max}}$
51	CH <sub>3</sub> Br	Methyl bromide	-1.7379	1.0082	467.00	180	444	0.1038	0.1352	0.0183
52	CH <sub>3</sub> Cl	Methyl chloride	-1.7528	1.3666	416.25	175	395	0.1593	0.2541	0.0177
53	C <sub>6</sub> H <sub>12</sub> O	Methyl isobutyl ketone	-1.6520	0.9961	571.40	189	543	0.1428	0.1696	0.0223
54	CH <sub>5</sub> N	Methylamine	-1.0947	0.5539	430.05	180	409	0.1997	0.2369	0.0804
55	C <sub>8</sub> H <sub>10</sub>	<i>m</i> -Xylene	-1.7286	1.0193	617.05	225	586	0.1305	0.1442	0.0187
56	C <sub>10</sub> H <sub>8</sub>	Naphthalene	-1.0304	0.1860	748.35	353	711	—	0.1326	0.0932
57	C <sub>4</sub> H <sub>10</sub>	<i>n</i> -Butane	-1.8929	1.2885	425.18	135	404	0.1046	0.1796	0.0128
58	C <sub>10</sub> H <sub>22</sub>	<i>n</i> -Decane	-1.7768	1.0839	618.45	243	588	0.1322	0.1425	0.0167
59	C <sub>7</sub> H <sub>16</sub>	<i>n</i> -Heptane	-1.8482	1.1843	540.26	183	513	0.1240	0.1570	0.0142
60	C <sub>6</sub> H <sub>14</sub>	<i>n</i> -Hexane	-1.8389	1.1860	507.43	178	482	0.1208	0.1588	0.0145
61	C <sub>6</sub> H <sub>5</sub> NO <sub>2</sub>	Nitrobenzene	-1.3942	0.6571	719.00	279	683	0.1478	0.1484	0.0403
62	N <sub>2</sub>	Nitrogen	0.2130	-4.2050E-04	-7.2951E-06	70	126	—	0.148	0.044
63	C <sub>9</sub> H <sub>20</sub>	<i>n</i> -Nonane	-1.7865	1.1033	595.65	220	566	0.1313	0.1486	0.0163
64	C <sub>8</sub> H <sub>18</sub>	<i>n</i> -Octane	-1.8388	1.1699	568.83	216	540	0.1281	0.1487	0.0145
65	C <sub>5</sub> H <sub>12</sub>	<i>n</i> -Pentane	-1.2287	0.5822	469.65	143	446	0.1480	0.1769	0.0591
66	O <sub>2</sub>	Oxygen	0.2320	-5.6357E-04	-3.8093E-06	60	155	—	0.184	0.053
67	C <sub>8</sub> H <sub>10</sub>	<i>o</i> -Xylene	-1.7372	1.0282	630.37	248	599	0.1315	0.1398	0.0183
68	C <sub>6</sub> H <sub>6</sub> O	Phenol	-1.1489	0.4091	694.25	314	660	—	0.1553	0.0710
69	C <sub>8</sub> H <sub>10</sub>	<i>p</i> -Xylene	-1.7354	1.0254	616.26	286	585	0.1299	0.1291	0.0184
70	C <sub>5</sub> H <sub>5</sub> N	Pyridine	-1.2083	0.5146	619.95	232	589	0.1653	0.1729	0.0619
71	C <sub>8</sub> H <sub>8</sub>	Styrene	-1.7023	1.0002	648.00	243	616	0.1369	0.1460	0.0198
72	SO <sub>2</sub>	Sulfur dioxide	0.3822	-6.2540E-04	-5.6891E-19	200	400	0.196	0.257	0.132
73	C <sub>7</sub> H <sub>8</sub>	Toluene	-1.6735	0.9773	591.79	178	562	0.1338	0.1596	0.0212
74	C <sub>4</sub> H <sub>6</sub> O <sub>2</sub>	Vinyl acetate	-1.7519	1.1895	524.00	180	498	0.1525	0.1970	0.0177
75	H <sub>2</sub> O	Water	-0.2758	4.6120E-03	-5.5391E-06	273	633	0.607	0.570	0.424

*k*<sub>liq</sub> – thermal conductivity of liquid, W/(m K)

*A*, *B*, and *C* – regression coefficients for chemical compound

*T* – temperature, K

*T*<sub>min</sub> – minimum temperature, K

*T*<sub>max</sub> – maximum temperature, K

**TABLE C-7b Thermal Conductivity of Liquids and Solids (Inorganic)**

No.	Formula	Substance	$k = A + BT + CT^2$								
			A	B	C	$T_{\min}$	$T_{\max}$	Phase	$k$ at $25^\circ\text{C}$	$k$ at $T_{\min}$	$k$ at $T_{\max}$
1	Ag	Silver	438.2178	-2.2947E-02	-3.5429E-05	200	1200	solid	428.227	432.211	359.664
2	Al	Aluminum	228.2103	5.7999E-02	-8.6806E-05	200	934	solid	237.786	236.338	206.658
3	As	Arsenic	122.7520	-3.4038E-01	3.3153E-04	200	500	solid	50.739	67.937	35.445
4	Au	Gold	335.4544	-5.8253E-02	-7.4773E-06	200	1300	solid	317.422	323.505	247.089
5	Ba	Barium	26.5122	-5.3067E-02	8.6615E-05	150	300	solid	18.390	20.501	18.387
6	Br <sub>2</sub>	Bromine	0.1325	1.0331E-04	-4.5514E-07	266	584	liquid	0.123	0.128	0.038
7	Cd	Cadmium	101.2521	-5.5398E-03	-2.7578E-05	200	594	solid	97.149	99.041	88.231
8	Co	Cobalt	150.4892	-1.9030E-01	8.9428E-05	200	1200	solid	101.701	116.006	50.906
9	Cu	Copper	426.2970	-8.3932E-02	9.3782E-06	200	1300	solid	402.106	409.886	333.035
10	HCl	Hydrogen chloride	0.8045	-2.1020E-03	-2.3238E-16	273	323	liquid	0.178	0.231	0.126
11	H <sub>2</sub> O	Water	-0.2758	4.6120E-03	-5.5391E-06	273	633	liquid	0.607	0.570	0.424
12	Hg	Mercury	0.9230	2.8887E-02	-1.5499E-05	234	1562	liquid	8.158	6.834	8.229
13	I <sub>2</sub>	Iodine	0.0965	1.7970E-04	-3.4020E-07	387	785	liquid	-	0.115	0.028
14	K	Potassium	69.3316	-4.7493E-02	9.4295E-06	350	1600	liquid	-	53.864	17.482
15	KBr	Potassium bromide	13.8899	-5.4091E-02	8.0015E-05	241	372	solid	4.875	5.501	4.841
16	KCl	Potassium chloride	39.3670	-1.9735E-01	2.8892E-04	120	360	solid	6.210	19.845	5.765
17	Li	Lithium	28.9331	3.5692E-02	-1.0087E-05	454	3677	liquid	-	43.058	23.793
18	Mg	Magnesium	165.8119	-3.7721E-02	1.6112E-05	200	900	solid	155.998	158.912	144.914
19	NH <sub>3</sub>	Ammonia	1.1606	-2.2840E-03	3.1245E-18	220	400	liquid	0.480	0.658	0.247
20	NH <sub>5</sub> S	Ammonium hydrogen sulfide	0.1661	6.3855E-04	-1.6809E-06	391	449	liquid	-	0.159	0.114
21	NO	Nitric oxide	0.1878	1.0293E-03	-9.4300E-06	110	176	liquid	-	0.187	0.077
22	Na	Sodium	98.8908	-4.3158E-02	5.2876E-06	371	2316	liquid	-	83.607	27.299
23	NaBr	Sodium bromide	0.3616	-3.3985E-05	-7.3045E-09	1020	3858	liquid	-	0.319	0.122
24	NaCl	Sodium chloride	51.6119	-2.9610E-01	4.7053E-04	80	380	solid	5.157	30.935	7.038
25	NaOH	Sodium hydroxide	-3.2252	4.0045E-03	5.0633E-06	592	592	solid	-	0.920	0.920
26	Ni	Nickel	144.4370	-2.2793E-01	1.6434E-04	250	800	solid	91.088	97.726	67.271
27	Pb	Lead	39.3335	-1.3469E-02	5.1500E-07	200	600	solid	35.363	36.660	31.438
28	S	Sulfur	0.0262	4.1738E-04	-3.2471E-07	388	1182	liquid	-	0.139	0.066
29	Sb	Antimony	39.6013	-6.0319E-02	3.9401E-05	200	900	solid	25.120	29.114	17.229
30	SiO <sub>2</sub>	Silicon oxide	2.1744	-3.7847E-03	4.6353E-06	273	1400	solid	1.458	1.487	5.961
31	Sn	Tin	92.3073	-1.1537E-01	1.0000E-04	200	500	solid	66.799	73.233	59.622
32	SnCl <sub>2</sub>	Stannous chloride	0.1616	-1.1786E-04	-1.7857E-07	260	340	liquid	0.111	0.119	0.101
33	SnCl <sub>4</sub>	Stannic chloride	0.1482	-6.7765E-05	-1.8857E-07	243	558	liquid	0.111	0.121	0.052
34	Zn	Zinc	125.5025	-3.0320E-02	-1.1471E-05	200	693	solid	115.443	118.980	98.982
35	ZnCl <sub>2</sub>	Zinc chloride	2.2616	-6.2681E-03	4.9822E-06	563	641	solid	-	0.312	0.291
36	ZnO	Zinc oxide	71.4550	-1.8504E-01	1.6306E-04	319	418	solid	-	29.020	22.599

*k* – thermal conductivity, W/(m K)*A, B, and C* – regression coefficients for chemical compound*T* – temperature, K*T<sub>min</sub>* – minimum temperature, K*T<sub>max</sub>* – maximum temperature, K

**TABLE C-8 Surface Tension of Organic Liquids**

$\sigma = A (1 - T/T_c)^n$								
No.	Formula	Substance	A	T <sub>c</sub>	n	T <sub>min</sub>	T <sub>max</sub>	$\sigma$ at 25°C
1	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,1-Trichloroethane	65.600	545.00	1.2170	242.75	545.00	25.02
2	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,2-Trichloroethane	77.200	602.00	1.2100	236.50	602.00	33.75
3	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,1-Dichloroethane	72.110	523.00	1.2530	176.19	523.00	25.04
4	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,2-Dichloroethane	80.140	561.00	1.2000	237.49	561.00	32.27
5	C <sub>4</sub> H <sub>6</sub>	1,3-Butadiene	47.682	425.37	1.0507	164.25	425.37	13.41
6	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	1,4-Dioxane	82.160	587.00	1.2899	284.95	587.00	32.92
7	C <sub>4</sub> H <sub>10</sub> O	1-Butanol ( <i>n</i> -Butanol)	64.526	562.93	1.2222	283.15	562.93	25.67
8	C <sub>4</sub> H <sub>8</sub>	1-Butene	56.000	419.59	1.2341	87.80	419.59	12.12
9	C <sub>10</sub> H <sub>20</sub>	1-Decene	54.880	617.05	1.2818	206.89	617.05	23.55
10	C <sub>9</sub> H <sub>20</sub>	1-Nonane ( <i>n</i> -Nonane)	55.400	595.65	1.3027	219.63	595.65	22.43
11	C <sub>8</sub> H <sub>16</sub>	1-Octene	54.040	566.60	1.2468	171.45	566.60	21.29
12	C <sub>3</sub> H <sub>8</sub> O	1-Propanol ( <i>n</i> -Propanol)	66.660	536.71	1.2222	283.15	536.71	24.74
13	C <sub>2</sub> H <sub>4</sub> O	Acetaldehyde	67.660	476.00	1.1940	150.15	461.00	19.53
14	C <sub>2</sub> H <sub>4</sub> O <sub>2</sub>	Acetic acid	57.050	592.71	1.0703	289.81	592.71	26.99
15	C <sub>4</sub> H <sub>6</sub> O <sub>3</sub>	Acetic anhydride	80.370	569.15	1.2420	200.15	569.15	31.98
16	C <sub>3</sub> H <sub>6</sub> O	Acetone	62.200	508.20	1.1240	178.45	508.20	23.04
17	C <sub>3</sub> H <sub>4</sub> O <sub>2</sub>	Acrylic acid	65.495	615.00	1.2549	286.65	615.00	28.50
18	C <sub>6</sub> H <sub>7</sub> N	Aniline	77.260	699.00	1.0800	267.13	699.00	42.38
19	C <sub>6</sub> H <sub>6</sub>	Benzene	71.950	562.16	1.2389	278.68	562.16	28.21
20	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Butyric acid	56.230	628.00	1.1930	267.95	628.00	26.08
21	CS <sub>2</sub>	Carbon disulfide	79.590	552.00	1.1909	161.11	552.00	19.72
22	CO <sub>2</sub>	Carbon dioxide	79.970	304.19	1.2617	216.58	304.19	0.57
23	CO	Carbon monoxide	27.959	132.92	1.1330	68.15	132.92	—
24	CCl <sub>4</sub>	Carbon tetrachloride	66.750	556.35	1.2140	250.33	556.35	26.29
25	Cl <sub>2</sub>	Chlorine	67.560	417.15	1.0850	172.12	417.15	17.32
26	CHCl <sub>3</sub>	Chloroform	69.284	536.40	1.1761	209.63	536.40	26.68
27	C <sub>6</sub> H <sub>12</sub>	Cyclohexane	66.097	553.54	1.2553	279.69	553.54	24.65
28	C <sub>6</sub> H <sub>12</sub> O	Cyclohexanol	65.400	625.15	1.0360	296.60	625.15	3.42
29	C <sub>3</sub> H <sub>6</sub>	Cyclopropane	74.254	397.91	1.2599	145.59	397.91	12.99
30	CH <sub>2</sub> Cl <sub>2</sub>	Dichloromethane	88.570	510.00	1.2800	178.01	510.00	28.77
31	C <sub>4</sub> H <sub>10</sub> O	Diethyl ether	57.356	466.70	1.2280	156.85	466.70	16.42
32	C <sub>5</sub> H <sub>10</sub> O	Diethyl ketone	54.700	560.95	1.0560	234.18	560.95	24.56
33	C <sub>2</sub> H <sub>7</sub> N	Dimethylamine	56.215	437.65	1.0807	180.96	437.65	16.34
34	C <sub>2</sub> H <sub>6</sub>	Ethane	48.984	305.42	1.2065	90.35	305.42	0.54
35	C <sub>2</sub> H <sub>6</sub> O	Ethanol	67.036	516.25	1.2222	273.15	516.25	23.39
36	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Ethyl acetate	59.870	523.30	1.1220	189.60	523.30	23.24
37	C <sub>2</sub> H <sub>5</sub> Cl	Ethyl chloride	57.652	460.35	1.0880	134.80	460.35	18.53
38	C <sub>8</sub> H <sub>10</sub>	Ethylbenzene	65.700	617.05	1.2780	225.30	617.05	28.26
39	C <sub>2</sub> H <sub>4</sub>	Ethylene	52.940	282.36	1.2784	103.97	282.36	—
40	C <sub>2</sub> H <sub>6</sub> O <sub>2</sub>	Ethylene glycol	106.491	645.00	1.2222	260.15	645.00	49.89
41	C <sub>2</sub> H <sub>4</sub> O	Ethylene oxide	74.730	469.15	1.1410	160.71	469.15	23.63
42	F <sub>2</sub>	Fluorine	40.040	144.31	1.2266	53.53	144.31	22.68
43	C <sub>3</sub> H <sub>8</sub> O <sub>3</sub>	Glycerol	124.793	723.00	1.2222	291.33	723.00	65.16
44	H <sub>2</sub>	Hydrogen	5.336	33.18	1.0622	13.95	33.18	2.99
45	HCl	Hydrogen chloride	85.200	324.65	1.2970	158.97	324.65	3.30
46	CHN	Hydrogen cyanide	52.256	456.65	1.0198	259.83	456.65	17.76
47	H <sub>2</sub> O <sub>2</sub>	Hydrogen peroxide	141.031	730.15	1.2222	272.72	730.15	74.26
48	C <sub>4</sub> H <sub>10</sub>	<i>i</i> -Butane ( <i>iso</i> -Butane)	52.165	408.14	1.2723	113.54	408.14	9.84
49	CH <sub>4</sub>	Methane	35.684	190.58	1.0920	90.67	190.58	—
50	CH <sub>4</sub> O	Methanol	68.329	512.58	1.2222	273.10	512.58	23.55
51	CH <sub>3</sub> Br	Methyl bromide	83.795	467.00	1.2222	179.47	467.00	24.17

**TABLE C-8—(continued)**

$\sigma = A (1 - T/T_c)^n$									
No.	Formula	Substance	A	T <sub>c</sub>	n	T <sub>min</sub>	T <sub>max</sub>	σ at 25°C	
52	CH <sub>3</sub> Cl	Methyl chloride	68.594	416.25	1.1966	175.43	416.25	15.19	
53	C <sub>6</sub> H <sub>12</sub> O	Methyl isobutyl ketone	57.130	571.40	1.2040	189.15	571.40	23.50	
54	CH <sub>5</sub> N	Methylamine	85.600	430.05	1.2556	179.69	430.05	19.41	
55	C <sub>8</sub> H <sub>10</sub>	<i>m</i> -Xylene	65.700	617.05	1.2780	225.30	617.05	28.26	
56	C <sub>10</sub> H <sub>8</sub>	Naphthalene	83.190	748.35	1.3896	353.43	748.35	—	
57	C <sub>4</sub> H <sub>10</sub>	<i>n</i> -Butane	52.660	425.18	1.2330	134.86	425.18	11.87	
58	C <sub>10</sub> H <sub>22</sub>	<i>n</i> -Decane	55.777	618.45	1.3198	243.49	618.45	23.41	
59	C <sub>7</sub> H <sub>16</sub>	<i>n</i> -Heptane	53.640	540.26	1.2431	182.56	540.26	19.78	
60	C <sub>6</sub> H <sub>14</sub>	<i>n</i> -Hexane	56.081	507.43	1.2843	177.84	507.43	17.98	
61	C <sub>6</sub> H <sub>5</sub> NO <sub>2</sub>	Nitrobenzene	79.440	719.00	1.1362	278.91	719.00	43.23	
62	C <sub>9</sub> H <sub>20</sub>	<i>n</i> -Nonane	55.400	595.65	1.3027	219.63	595.65	22.43	
63	C <sub>8</sub> H <sub>18</sub>	<i>n</i> -Octane	52.036	568.83	1.2168	216.38	568.83	21.08	
64	C <sub>5</sub> H <sub>12</sub>	<i>n</i> -Pentane	52.090	469.65	1.2054	143.42	469.65	15.47	
65	C <sub>8</sub> H <sub>10</sub>	<i>o</i> -Xylene	66.100	630.37	1.2544	247.98	630.37	29.60	
66	C <sub>6</sub> H <sub>6</sub> O	Phenol	74.500	694.25	1.0767	314.06	694.25	—	
67	C <sub>8</sub> H <sub>10</sub>	<i>p</i> -Xylene	64.850	616.26	1.2743	286.41	616.26	27.92	
68	C <sub>5</sub> H <sub>5</sub> N	Pyridine	81.500	619.95	1.2160	231.51	619.95	36.72	
69	C <sub>8</sub> H <sub>8</sub>	Styrene	68.178	648.00	1.2222	273.15	648.00	32.10	
70	C <sub>7</sub> H <sub>8</sub>	Toluene	66.850	591.79	1.2456	178.18	591.79	27.93	
71	C <sub>4</sub> H <sub>6</sub> O <sub>2</sub>	Vinyl acetate	68.685	524.00	1.2500	180.35	524.00	23.99	
72	H <sub>2</sub> O	Water	132.674	647.13	0.9550	273.15	647.13	73.56	

*σ* – surface tension, dyn/cm – surface tension, dyn/cm  
*A*, *T<sub>c</sub>*, and *n* – regression coefficients for chemical compound  
*T* – temperature, K  
*T<sub>min</sub>* – minimum temperature, K  
*T<sub>max</sub>* – maximum temperature, K

**TABLE C-9 Vapor Pressure**

No .	Formula	Substance	$\log_{10} P_v = A + B/T + C \log_{10} T + DT + ET^2$					
			A	B	C	D	E	$T_{\min}$
1	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,1-Trichloroethane	36.5468	-2.8421E+03	-1.0205E+01	-2.6369E-09	3.7075E-06	242.75
2	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,2-Trichloroethane	25.0845	-2.7368E+03	-5.9182E+00	2.5155E-01	1.1831E-06	236.50
3	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,1-Dichloroethane	33.3800	-2.6102E+03	-9.1336E+00	-2.8388E-11	3.7323E-06	176.19
4	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,2-Dichloroethane	48.4226	-3.1803E+03	-1.5370E+01	7.2935E-03	2.6844E-14	237.49
5	C <sub>4</sub> H <sub>6</sub>	1,3-Butadiene	30.0572	-1.9891E+03	-8.2922E+00	2.5664E-10	5.1334E-06	164.25
6	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	1,4-Dioxane	20.5761	-2.4658E+03	-4.3645E+00	-2.7053E-10	8.5235E-07	284.95
7	C <sub>4</sub> H <sub>10</sub> O	1-Butanol ( <i>n</i> -Butanol)	39.6673	-4.0017E+03	-1.0295E+01	-3.2572E-10	8.6672E-07	183.85
8	C <sub>4</sub> H <sub>8</sub>	1-Butene	27.3116	-1.9235E+03	-7.2064E+00	7.4852E-12	3.6481E-06	87.80
9	C <sub>10</sub> H <sub>20</sub>	1-Decene	2.2678	-3.1244E+03	5.4320E+00	-2.0137E-02	1.1221E-05	206.89
10	C <sub>9</sub> H <sub>20</sub>	1-Nonane ( <i>n</i> -Nonane)	8.8817	-2.8042E+03	1.5262E+00	-1.0464E-02	5.7972E-06	219.63
11	C <sub>8</sub> H <sub>16</sub>	1-Octene	56.1183	-3.7657E+03	-1.8006E+01	7.7387E-03	-1.3036E-13	171.45
12	C <sub>3</sub> H <sub>8</sub> O	1-Propanol ( <i>n</i> -Propanol)	31.5155	-3.4570E+03	-7.5235E+00	-4.2870E-11	1.3029E-07	146.95
13	C <sub>2</sub> H <sub>4</sub> O	Acetaldehyde	87.3702	-3.6822E+03	-3.1548E+01	2.0114E-02	5.5341E-13	150.15
14	C <sub>2</sub> H <sub>4</sub> O <sub>2</sub>	Acetic acid	28.3756	-2.9734E+03	-7.0320E+00	-1.5051E-09	2.1806E-06	289.81
15	C <sub>4</sub> H <sub>6</sub> O <sub>3</sub>	Acetic anhydride	43.5021	-3.8643E+03	-1.2162E+01	-2.1843E-09	3.3250E-06	200.15
16	C <sub>3</sub> H <sub>6</sub> O	Acetone	28.5884	-2.4690E+03	-7.3510E+00	2.8025E-10	2.7361E-06	178.45
17	C <sub>3</sub> H <sub>4</sub> O <sub>2</sub>	Acrylic acid	23.0607	-3.1347E+03	-4.8813E+00	4.3690E-04	-4.9161E-13	286.65
18	NH <sub>3</sub>	Ammonia	37.1575	-2.0277E+03	-1.1601E+01	7.4625E-03	-9.5811E-12	195.41
19	C <sub>6</sub> H <sub>7</sub> N	Aniline	124.3764	-7.1676E+03	-4.2763E+01	1.7336E-02	5.7138E-15	267.13
20	C <sub>6</sub> H <sub>6</sub>	Benzene	31.7718	-2.7254E+03	-8.4443E+00	-5.3534E-09	2.7187E-06	276.68
21	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Butyric acid	8.0847	-3.3219E+03	2.4312E+00	-1.1734E-02	5.7992E-06	267.95
22	CS <sub>2</sub>	Carbon disulfide	25.1475	-2.0439E+03	-6.7794E+00	3.4828E-03	-1.0105E-14	161.58
23	CO <sub>2</sub>	Carbon dioxide	35.0169	-1.5119E+03	-1.1334E+01	9.3368E-03	1.7136E-09	216.58
24	CO	Carbon monoxide	51.8145	-7.8824E+02	-2.2734E+01	5.1225E-02	6.1896E-11	68.15
25	CCl <sub>4</sub>	Carbon tetrachloride	31.9407	-2.6614E+03	-8.5763E+00	-6.7136E-10	2.9732E-06	250.33
26	Cl <sub>2</sub>	Chlorine	28.8659	-1.6745E+03	-8.5216E+00	5.3792E-03	-7.7867E-13	172.12
27	CHCl <sub>3</sub>	Chloroform	56.6178	-3.2462E+03	-1.8700E+01	9.5150E-03	1.553E-12	209.63
28	C <sub>6</sub> H <sub>12</sub>	Cyclohexane	48.5529	-3.0874E+03	-1.5521E+01	7.3830E-03	6.3563E-12	279.69
29	C <sub>6</sub> H <sub>12</sub> O	Cyclohexanol	49.9123	-4.8446E+03	-1.3711E+01	3.5451E-09	1.5932E-06	296.60
30	C <sub>3</sub> H <sub>6</sub>	Cyclopropane	37.8180	-1.8661E+03	-1.2278E+01	8.5721E-03	-2.9652E-13	145.59
31	CH <sub>2</sub> Cl <sub>2</sub>	Dichloromethane	32.5609	-2.5166E+03	-8.8015E+00	1.2934E-10	3.194E-06	178.01
32	C <sub>4</sub> H <sub>10</sub> O	Diethyl ether	41.7519	-2.7410E+03	-1.2270E+01	-3.1948E-10	5.9802E-06	156.85
33	C <sub>5</sub> H <sub>10</sub> O	Diethyl ketone	32.2560	-2.9431E+03	-8.5068E+00	-4.5720E-10	2.6177E-06	234.18
34	C <sub>2</sub> H <sub>7</sub> N	Dimethylamine	36.9182	-2.4965E+03	-1.0417E+01	-1.6287E-09	4.6496E-06	180.96
35	C <sub>2</sub> H <sub>6</sub>	Ethane	20.6973	-1.341E+03	-5.2514E+00	-9.8774E-11	6.7329E-06	90.35
36	C <sub>2</sub> H <sub>6</sub> O	Ethanol	23.8442	-2.8642E+03	-5.0474E+00	3.7448E-11	2.7361E-07	159.05
37	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Ethyl acetate	0.6955	-2.2498E+03	5.4643E+00	-1.9451E-02	1.2362E-05	189.60
38	C <sub>2</sub> H <sub>5</sub> Cl	Ethyl chloride	28.3448	-2.0788E+03	-7.5387E+00	-1.6384E-11	4.0550E-06	136.75
39	C <sub>8</sub> H <sub>10</sub>	Ethylbenzene	36.1998	-3.3402E+03	-9.7970E+00	-1.1467E-11	2.5758E-06	178.15
40	C <sub>2</sub> H <sub>4</sub>	Ethylene	18.7964	-9.9962E+02	-4.5788E+00	9.9746E-11	6.7880E-06	104.01
41	C <sub>2</sub> H <sub>6</sub> O <sub>2</sub>	Ethylene glycol	82.4062	-6.3472E+03	-2.5433E+01	-2.3732E-09	8.7467E-06	260.15
42	C <sub>2</sub> H <sub>4</sub> O	Ethylene oxide	39.9235	-2.3595E+03	-1.2517E+01	6.9835E-03	-1.1101E-13	160.71
43	F <sub>2</sub>	Fluorine	27.1409	-5.7201E+02	-1.0015E+01	2.1078E-02	8.9567E-13	53.48
44	C <sub>3</sub> H <sub>8</sub> O <sub>3</sub>	Glycerol	-62.7929	-3.6585E+03	3.4249E+01	-5.1940E-02	2.2830E-05	291.33
45	H <sub>2</sub>	Hydrogen	3.4132	-4.1316E+01	1.0947E+00	-6.6896E-10	1.4589E-04	13.95
46	HCl	Hydrogen chloride	43.5455	-1.6279E+03	-1.5214E+01	1.3783E-02	-1.4984E-11	158.97
47	CHN	Hydrogen cyanide	-57.5717	-3.5182E+02	2.9640E+01	-4.7820E-02	2.8550E-05	259.91
48	H <sub>2</sub> O <sub>2</sub>	Hydrogen peroxide	33.3222	-3.7350E+03	-8.3458E+00	-1.2351E-10	1.6917E-06	272.74
49	C <sub>4</sub> H <sub>10</sub>	<i>i</i> -Butane ( <i>iso</i> -Butane)	31.2541	-1.9532E+03	-8.8060E+00	8.9246E-11	5.7501E-06	113.54
50	CH <sub>4</sub>	Methane	14.6667	-5.7097E+02	-3.3373E+00	2.1999E-09	1.3096E-05	90.67
51	CH <sub>4</sub> O	Methanol	45.6171	-3.2447E+03	-1.3988E+01	6.6365E-03	-1.0507E-13	175.47

**TABLE C-9—(continued)**

$\log_{10} P_v = A + B/T + C \log_{10} T + DT + ET^2$									
No.	Formula	Substance	A	B	C	D	E	$T_{\min}$	$T_{\max}$
52	CH <sub>3</sub> Br	Methyl bromide	29.3988	-2.0406E+03	-7.9966E+00	-4.1899E-10	5.0174E-06	179.47	467.00
53	CH <sub>3</sub> Cl	Methyl chloride	25.7264	-1.7503E+03	-6.7151E+00	-1.2956E-09	4.4341E-06	175.43	416.25
54	C <sub>6</sub> H <sub>12</sub> O	Methyl isobutyl ketone	64.1919	-4.3577E+03	-1.9766E+01	-3.9997E-10	7.1020E-06	189.15	571.40
55	CH <sub>5</sub> N	Methylamine	30.5366	-2.2074E+03	-8.0919E+00	-2.7828E-11	3.5234E-06	179.69	430.05
56	C <sub>8</sub> H <sub>10</sub>	<i>m</i> -Xylene	34.6803	-3.2981E+03	-9.2570E+00	-4.3563E-10	2.4103E-06	225.30	617.05
57	C <sub>10</sub> H <sub>8</sub>	Naphthalene	34.9161	-3.9357E+03	-9.0648E+00	-2.0672E-09	1.5550E-06	353.43	748.35
58	C <sub>4</sub> H <sub>10</sub>	<i>n</i> -Butane	27.0441	-1.9049E+03	-7.1805E+05	-6.6845E-11	4.2190E-06	134.86	425.18
59	C <sub>10</sub> H <sub>22</sub>	<i>n</i> -Decane	26.5125	-3.3584E+03	-6.1174E+00	-3.3225E-10	4.8554E-07	243.49	618.45
60	C <sub>7</sub> H <sub>16</sub>	<i>n</i> -Heptane	65.0257	-3.8188E+03	-2.1684E+01	1.0387E-02	1.0206E-14	182.56	540.26
61	C <sub>6</sub> H <sub>14</sub>	<i>n</i> -Hexane	69.7378	-3.6278E+03	-2.3927E+01	1.2810E-02	-1.6844E-13	177.84	507.43
62	C <sub>6</sub> H <sub>5</sub> NO <sub>2</sub>	Nitrobenzene	-54.4937	-2.1123E+03	2.9321E+01	-4.4839E-02	2.0162E-05	278.91	719.00
63	N <sub>2</sub>	Nitrogen	23.8572	-4.7668E+02	-8.6689E+00	2.0128E-02	-2.4139E-11	63.15	126.10
64	C <sub>9</sub> H <sub>20</sub>	<i>n</i> -Nonane	8.8817	-2.8042E+03	1.5262E+00	-1.0464E-02	5.7972E-06	219.63	595.65
65	C <sub>8</sub> H <sub>18</sub>	<i>n</i> -Octane	29.0948	-3.0114E+03	-7.2653E+00	-2.2696E-11	1.4680E-06	216.38	568.83
66	C <sub>5</sub> H <sub>12</sub>	<i>n</i> -Pentane	33.3239	-2.4227E+03	-9.2354E+00	9.0199E-11	4.1050E-06	143.42	469.65
67	O <sub>2</sub>	Oxygen	20.6695	-5.2697E+02	-6.7062E+00	1.2926E-02	-9.8832E-13	54.35	154.58
68	C <sub>8</sub> H <sub>10</sub>	<i>o</i> -Xylene	37.2413	-3.4573E+03	-1.0126E+01	9.0676E-11	2.6123E-06	247.98	630.37
69	C <sub>6</sub> H <sub>6</sub> O	Phenol	23.5332	-3.4961E+03	-4.8990E+00	1.2160E-04	9.6537E-13	314.06	694.25
70	C <sub>8</sub> H <sub>10</sub>	<i>p</i> -Xylene	60.0531	-4.0159E+03	-1.9441E+01	8.2881E-03	-2.3647E-12	286.41	616.26
71	C <sub>5</sub> H <sub>5</sub> N	Pyridine	33.5541	-3.1318E+03	-8.8646E+00	7.1293E-12	2.2813E-06	231.51	619.95
72	C <sub>8</sub> H <sub>8</sub>	Styrene	55.8621	-4.0240E+03	-1.7609E+01	6.6842E-03	1.9438E-13	242.54	648.00
73	SO <sub>2</sub>	Sulfur dioxide	19.7418	-1.8132E+03	-4.1458E+00	-4.4284E-09	8.4918E-07	197.67	430.75
74	C <sub>7</sub> H <sub>8</sub>	Toluene	34.0775	-3.0379E+03	-9.1635E+00	1.0289E-11	2.7035E-06	178.18	591.79
75	C <sub>4</sub> H <sub>6</sub> O <sub>2</sub>	Vinyl acetate	12.7220	-2.1770E+03	-9.1458E-01	-4.5688E-03	2.9673E-06	180.35	524.00
76	H <sub>2</sub> O	Water	29.8605	-3.1522E+03	-7.3037E+00	2.4247E-09	1.8090E-06	273.16	647.13

 $P_v$  – vapor pressure, mmHg

A, B, C, D, and E – regression coefficients for chemical compound

T – temperature, K

 $T_{\min}$  – minimum temperature, K $T_{\max}$  – maximum temperature, K

**TABLE C-10 Enthalpy of Vaporization**

$\Delta H_v = A (1 - T/T_c)^n$									
No.	Formula	Substance	A	T <sub>c</sub>	n	T <sub>min</sub>	T <sub>max</sub>	T <sub>B</sub>	$\Delta H_v$ at T <sub>B</sub>
1	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,1-Trichloroethane	44.083	545.00	0.385	242.75	545.00	347.23	29.84
2	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,2-Trichloroethane	46.865	602.00	0.294	236.50	602.00	387.00	34.62
3	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,1-Dichloroethane	42.180	523.00	0.367	176.19	523.00	330.45	29.23
4	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,2-Dichloroethane	45.426	561.00	0.342	237.49	561.00	356.59	32.16
5	C <sub>4</sub> H <sub>6</sub>	1,3-Butadiene	35.170	425.37	0.448	200.00	425.37	266.74	22.48
6	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	1,4-Dioxane	36.047	587.00	0.049	340.00	587.00	374.47	34.30
7	C <sub>4</sub> H <sub>10</sub> O	1-Butanol ( <i>n</i> -Butanol)	63.024	562.93	0.318	220.00	562.93	390.81	43.24
8	C <sub>4</sub> H <sub>8</sub>	1-Butene	33.390	419.59	0.393	87.80	419.59	266.90	22.44
9	C <sub>10</sub> H <sub>20</sub>	1-Decene	65.577	617.05	0.402	206.89	617.05	443.75	39.36
10	C <sub>9</sub> H <sub>20</sub>	1-Nonane ( <i>n</i> -Nonane)	59.378	595.65	0.377	219.63	595.65	423.97	37.15
11	C <sub>8</sub> H <sub>16</sub>	1-Octene	55.443	566.60	0.401	171.45	566.60	394.44	34.39
12	C <sub>3</sub> H <sub>8</sub> O	1-Propanol ( <i>n</i> -Propanol)	70.179	536.71	0.451	146.95	536.71	370.35	41.38
13	C <sub>2</sub> H <sub>4</sub> O	Acetaldehyde	44.950	461.00	0.594	150.15	461.00	293.55	24.63
14	C <sub>2</sub> H <sub>4</sub> O <sub>2</sub>	Acetic acid	11.575	592.71	-0.650	289.81	391.05	391.05	23.33
15	C <sub>4</sub> H <sub>6</sub> O <sub>3</sub>	Acetic anhydride	58.520	569.15	0.280	200.15	569.15	411.78	40.83
16	C <sub>3</sub> H <sub>6</sub> O	Acetone	49.244	508.20	0.481	260.00	508.20	392.44	29.79
17	C <sub>3</sub> H <sub>4</sub> O <sub>2</sub>	Acrylic acid	62.977	615.00	0.351	286.65	615.00	414.15	42.52
18	NH <sub>3</sub>	Ammonia	31.523	405.65	0.364	195.41	405.65	239.72	22.77
19	C <sub>6</sub> H <sub>7</sub> N	Aniline	72.038	699.00	0.459	267.13	699.00	457.60	44.22
20	C <sub>6</sub> H <sub>6</sub>	Benzene	49.888	562.16	0.489	278.68	562.16	353.24	30.75
21	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Butyric acid	18.437	628.00	-0.757	320.00	436.42	436.42	45.29
22	CS <sub>2</sub>	Carbon disulfide	34.997	552.00	0.299	161.11	552.00	270.65	28.61
23	CO <sub>2</sub>	Carbon dioxide	15.326	304.19	0.227	216.58	304.19	194.67	—
24	CO	Carbon monoxide	8.003	132.92	0.318	68.15	132.92	81.70	5.91
25	CCl <sub>4</sub>	Carbon tetrachloride	37.890	556.35	0.241	250.33	556.35	349.79	29.84
26	Cl <sub>2</sub>	Chlorine	28.560	417.15	0.401	172.12	417.15	239.12	20.29
27	CHCl <sub>3</sub>	Chloroform	42.953	536.40	0.375	209.63	536.40	334.33	29.79
28	C <sub>6</sub> H <sub>12</sub>	Cyclohexane	49.060	553.54	0.486	279.69	553.54	353.87	29.89
29	C <sub>6</sub> H <sub>12</sub> O	Cyclohexanol	85.741	625.15	0.527	296.60	625.15	434.00	45.92
30	C <sub>3</sub> H <sub>6</sub>	Cyclopropane	28.060	397.91	0.361	145.59	397.91	240.37	20.08
31	CH <sub>2</sub> Cl <sub>2</sub>	Dichloromethane	41.910	510.00	0.410	178.01	510.00	312.90	28.38
32	C <sub>4</sub> H <sub>10</sub> O	Diethyl ether	41.972	466.70	0.407	156.85	466.70	307.58	27.09
33	C <sub>5</sub> H <sub>10</sub> O	Diethyl ketone	49.640	560.95	0.354	234.18	560.95	375.14	33.57
34	C <sub>2</sub> H <sub>7</sub> N	Dimethylamine	41.070	437.65	0.424	180.96	437.65	280.03	26.64
35	C <sub>2</sub> H <sub>6</sub>	Ethane	21.342	305.42	0.403	90.35	305.42	184.55	14.69
36	C <sub>2</sub> H <sub>6</sub> O	Ethanol	43.122	516.25	0.079	300.00	516.25	351.44	39.40
37	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Ethyl acetate	49.346	523.30	0.385	189.60	523.30	350.21	32.23
38	C <sub>2</sub> H <sub>5</sub> Cl	Ethyl chloride	35.233	460.35	0.365	134.80	460.35	285.42	24.75
39	C <sub>8</sub> H <sub>10</sub>	Ethylbenzene	54.788	617.17	0.388	178.15	617.17	409.35	35.91
40	C <sub>2</sub> H <sub>4</sub>	Ethylene	19.986	282.36	0.431	103.97	282.36	169.47	13.46
41	C <sub>2</sub> H <sub>6</sub> O <sub>2</sub>	Ethylene glycol	88.200	645.00	0.397	260.15	645.00	470.45	52.49
42	C <sub>2</sub> H <sub>4</sub> O	Ethylene oxide	36.474	469.15	0.377	160.71	469.15	283.85	25.70
43	F <sub>2</sub>	Fluorine	10.549	144.31	0.546	53.48	144.31	84.95	6.50
44	C <sub>3</sub> H <sub>8</sub> O <sub>3</sub>	Glycerol	104.153	723.00	0.301	291.33	723.00	563.15	66.13
45	H <sub>2</sub>	Hydrogen	0.659	33.18	0.380	13.95	33.18	20.39	0.46
46	HCl	Hydrogen chloride	30.540	324.65	0.647	158.97	324.65	188.15	17.43
47	CHN	Hydrogen cyanide	42.384	456.65	0.428	259.83	456.65	298.85	26.90
48	H <sub>2</sub> O <sub>2</sub>	Hydrogen peroxide	61.900	730.15	0.325	272.74	730.15	423.35	46.72
49	C <sub>4</sub> H <sub>10</sub>	<i>i</i> -Butane ( <i>iso</i> -Butane)	31.954	408.14	0.392	113.54	408.14	261.43	21.40
50	CH <sub>4</sub>	Methane	10.312	190.58	0.265	90.67	190.58	111.66	8.16
51	CH <sub>4</sub> O	Methanol	52.723	512.58	0.377	175.47	512.58	337.85	35.14

**TABLE C-10—(continued)**

$\Delta H_v = A (1 - T/T_c)^n$							$\Delta H_v$ at $T_B$		
No.	Formula	Substance	A	$T_c$	n	$T_{\min}$	$T_{\max}$	$T_B$	
52	CH <sub>3</sub> Br	Methyl bromide	32.009	467.00	0.308	179.47	467.00	276.71	24.28
53	CH <sub>3</sub> Cl	Methyl chloride	32.534	416.25	0.452	175.43	416.25	248.93	21.55
54	C <sub>6</sub> H <sub>12</sub> O	Methyl isobutyl ketone	57.680	571.40	0.416	189.15	571.40	389.65	35.82
55	CH <sub>5</sub> N	Methylamine	38.656	430.05	0.405	179.69	430.05	266.82	26.11
56	C <sub>8</sub> H <sub>10</sub>	<i>m</i> -Xylene	60.216	617.05	0.458	225.30	617.05	412.27	36.33
57	C <sub>10</sub> H <sub>8</sub>	Naphthalene	76.150	748.35	0.526	353.46	748.35	491.14	43.42
58	C <sub>4</sub> H <sub>10</sub>	<i>n</i> -Butane	33.020	425.18	0.377	134.86	425.18	272.65	22.44
59	C <sub>10</sub> H <sub>22</sub>	<i>n</i> -Decane	71.428	618.45	0.451	243.49	618.45	447.30	40.02
60	C <sub>7</sub> H <sub>16</sub>	<i>n</i> -Heptane	49.730	540.26	0.386	182.56	540.26	371.58	31.73
61	C <sub>6</sub> H <sub>14</sub>	<i>n</i> -Hexane	45.610	507.43	0.401	177.84	507.43	341.88	29.11
62	C <sub>6</sub> H <sub>5</sub> NO <sub>2</sub>	Nitrobenzene	67.414	719.00	0.380	278.91	719.00	483.95	44.08
63	N <sub>2</sub>	Nitrogen	9.430	126.10	0.533	63.15	126.10	77.35	5.68
64	C <sub>9</sub> H <sub>20</sub>	<i>n</i> -Nonane	59.378	595.65	0.377	219.63	595.65	423.97	37.15
65	C <sub>8</sub> H <sub>18</sub>	<i>n</i> -Octane	59.077	568.83	0.439	216.38	568.83	398.83	34.77
66	C <sub>5</sub> H <sub>12</sub>	<i>n</i> -Pentane	39.854	469.65	0.398	143.42	469.65	309.22	25.99
67	O <sub>2</sub>	Oxygen	8.040	154.58	0.201	54.35	154.58	90.17	6.74
68	C <sub>8</sub> H <sub>10</sub>	<i>o</i> -Xylene	55.606	630.37	0.375	247.98	630.37	417.58	37.00
69	C <sub>6</sub> H <sub>5</sub> O	Phenol	77.397	694.25	0.462	314.06	694.25	454.99	47.31
70	C <sub>8</sub> H <sub>10</sub>	<i>p</i> -Xylene	52.910	616.26	0.354	286.41	616.26	411.51	35.82
71	C <sub>5</sub> H <sub>5</sub> N	Pyridine	53.461	619.95	0.408	231.51	619.95	388.41	35.77
72	C <sub>8</sub> H <sub>8</sub>	Styrene	65.327	648.00	0.558	350.00	648.00	418.31	36.62
73	SO <sub>2</sub>	Sulfur dioxide	46.900	430.75	0.636	197.67	430.75	263.13	25.72
74	C <sub>7</sub> H <sub>8</sub>	Toluene	50.139	591.79	0.383	178.18	591.79	383.78	33.59
75	C <sub>4</sub> H <sub>6</sub> O <sub>2</sub>	Vinyl acetate	45.805	524.00	0.353	180.35	524.00	345.65	31.31
76	H <sub>2</sub> O	Water	52.053	647.13	0.321	273.16	647.13	373.15	39.50

$\Delta H_v$  – enthalpy of vaporization, kJ/mol  
 $A$ ,  $T_c$ , and  $n$  – regression coefficients of chemical compound  
 $T$  – temperature, K  
 $T_{\min}$  – minimum temperature, K  
 $T_{\max}$  – maximum temperature, K  
 $T_B$  – boiling temperature, K

**TABLE C-11 Enthalpy of Formation**

No.	Formula	Substance	$\Delta H_f^\circ = A + BT + CT^2$		(ΔHf° - kjoule/mol, T - K)					
			A	B	C	T <sub>min</sub>	T <sub>max</sub>	ΔHf° at 298 K	ΔHf° at 500 K	
1	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,1-Trichloroethane	-136.019	-2.6621E-02	1.7816E-05	298	1000	-142.30	-144.88	
2	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,2-Trichloroethane	-131.466	-2.9213E-02	1.7876E-05	298	1000	-138.49	-141.60	
3	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,1-Dichloroethane	-119.156	-4.3078E-02	2.2469E-05	298	1000	-129.91	-135.08	
4	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,2-Dichloroethane	-119.407	-4.0489E-02	1.9654E-05	298	1000	-129.70	-134.74	
5	C <sub>4</sub> H <sub>6</sub>	1,3-Butadiene	123.286	-5.1225E-02	2.3192E-05	298	1000	110.16	103.47	
6	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	1,4-Dioxane	-286.843	-1.1418E-01	6.2626E-05	298	1000	-315.06	-328.28	
7	C <sub>4</sub> H <sub>10</sub> O	1-Butanol ( <i>n</i> -Butanol)	-245.806	-1.1235E-01	5.3505E-05	298	1000	-274.43	-288.61	
8	C <sub>4</sub> H <sub>8</sub>	1-Butene	21.822	-8.5458E-02	3.8902E-05	298	1000	-0.13	-11.18	
9	C <sub>10</sub> H <sub>20</sub>	1-Decene	-72.195	-2.0530E-01	1.0175E-04	298	1000	-124.14	149.40	
10	C <sub>9</sub> H <sub>20</sub>	1-Nonane ( <i>n</i> -Nonane)	-175.883	-2.1036E-01	1.0501E-04	298	1000	-229.03	-254.91	
11	C <sub>8</sub> H <sub>16</sub>	1-Octene	-41.002	-1.6529E-01	8.0839E-05	298	1000	-82.93	-103.44	
12	C <sub>3</sub> H <sub>8</sub> O	1-Propanol ( <i>n</i> -Propanol)	-233.653	-9.2123E-02	4.2848E-05	298	1000	-257.53	-269.30	
13	C <sub>2</sub> H <sub>4</sub> O	Acetaldehyde	-154.122	-4.7166E-02	2.0279E-05	298	1000	-166.36	-172.64	
14	C <sub>2</sub> H <sub>4</sub> O <sub>2</sub>	Acetic acid	-422.584	-4.8354E-02	2.3337E-05	298	1000	-434.84	-440.93	
15	C <sub>4</sub> H <sub>6</sub> O <sub>3</sub>	Acetic anhydride	-554.715	-8.4124E-02	4.3618E-05	298	1000	-575.72	-585.87	
16	C <sub>3</sub> H <sub>6</sub> O	Acetone	-199.175	-7.1484E-02	3.2534E-05	298	1000	-217.57	-226.78	
17	C <sub>3</sub> H <sub>4</sub> O <sub>2</sub>	Acrylic acid	-325.038	-4.4058E-02	2.0926E-05	298	1000	-336.23	-341.84	
18	C <sub>6</sub> H <sub>7</sub> N	Aniline	105.261	-7.3513E-02	3.7553E-05	298	1000	86.86	77.89	
19	C <sub>6</sub> H <sub>6</sub>	Benzene	101.403	-7.2136E-02	3.2877E-05	298	1000	82.93	73.55	
20	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Butyric acid	-448.775	-8.5107E-02	4.2361E-05	298	1000	-470.28	-480.74	
21	CS <sub>2</sub>	Carbon disulfide	137.658	-8.1634E-02	4.2246E-05	389	717	117.07	107.40	
22	CO <sub>2</sub>	Carbon dioxide	-393.422	1.5913E-04	-1.3945E-06	298	1000	-393.51	-393.69	
23	CO	Carbon monoxide	-112.190	8.1182E-03	-8.0425E-06	298	1000	-110.54	-110.14	
24	CCl <sub>4</sub>	Carbon tetrachloride	-103.244	9.0605E-03	9.5332E-07	298	1000	-100.42	-98.48	
25	CHCl <sub>3</sub>	Chloroform	-98.906	-1.0149E-02	7.1513E-06	298	1000	-101.25	-102.19	
26	C <sub>6</sub> H <sub>12</sub>	Cyclohexane	-81.822	-1.6705E-01	9.2830E-05	298	1000	-123.14	-142.14	
27	C <sub>6</sub> H <sub>12</sub> O	Cyclohexanol	-255.672	-1.5846E-01	9.1123E-05	298	1000	-294.55	-312.12	
28	C <sub>3</sub> H <sub>6</sub>	Cyclopropane	71.797	-7.2889E-02	3.4947E-05	298	1000	53.30	44.09	
29	CH <sub>2</sub> Cl <sub>2</sub>	Dichloromethane	-88.943	-2.5399E-02	1.2304E-05	298	1000	-95.40	-98.57	
30	C <sub>4</sub> H <sub>10</sub> O	Diethyl ether	-223.739	-1.1173E-01	5.3459E-05	298	1000	-252.21	-266.24	
31	C <sub>5</sub> H <sub>10</sub> O	Diethyl ketone	-230.936	-1.1046E-01	5.2823E-05	298	1000	-259.20	-272.96	
32	C <sub>2</sub> H <sub>7</sub> N	Dimethylamine	2.793	-8.5680E-02	4.2790E-05	298	1000	-18.83	-29.35	
33	C <sub>2</sub> H <sub>6</sub>	Ethane	-66.735	-6.9337E-02	3.0379E-05	298	1000	-84.68	-93.81	
34	C <sub>2</sub> H <sub>6</sub> O	Ethanol	-216.961	-6.9572E-02	3.1744E-05	298	1000	-234.81	-243.81	
35	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Ethyl acetate	-420.199	-8.9885E-02	4.4497E-05	298	1000	-442.92	-454.02	
36	C <sub>2</sub> H <sub>5</sub> Cl	Ethyl chloride	-96.985	-5.7861E-02	2.7636E-05	298	1000	-111.71	-119.01	
37	C <sub>8</sub> H <sub>10</sub>	Ethylbenzene	58.099	-1.1129E-01	5.3183E-05	298	1000	29.79	15.75	
38	C <sub>2</sub> H <sub>4</sub>	Ethylene	63.053	-4.1076E-02	1.6598E-05	298	1000	52.30	46.66	
39	C <sub>2</sub> H <sub>6</sub> O <sub>2</sub>	Ethylene glycol	-377.811	-4.5844E-02	2.3144E-05	298	1000	-389.32	-394.95	
40	C <sub>2</sub> H <sub>4</sub> O	Ethylene oxide	-38.880	-5.4041E-02	2.5601E-05	298	1000	-52.63	-59.50	
41	C <sub>3</sub> H <sub>8</sub> O <sub>3</sub>	Glycerol	-559.438	-9.2185E-02	4.5003E-05	298	1000	-582.80	-594.28	
42	CHN	Hydrogen cyanide	135.847	-2.1831E-03	-4.3739E-07	298	1000	135.14	134.65	
43	C <sub>4</sub> H <sub>10</sub>	<i>i</i> -Butane ( <i>iso</i> -Butane)	-106.746	-1.0929E-01	5.2693E-05	298	1000	-134.52	-148.22	
44	CH <sub>4</sub>	Methane	-63.425	-4.3355E-02	1.7220E-05	298	1000	-74.85	-80.80	
45	CH <sub>4</sub> O	Methanol	-188.188	-4.9823E-02	2.0791E-05	298	1000	-201.17	-207.90	
46	CH <sub>3</sub> Br	Methyl bromide	-44.104	-3.5166E-02	1.5043E-05	333	1000	-37.66	-57.93	
47	CH <sub>3</sub> Cl	Methyl chloride	-76.576	-3.7541E-02	1.6128E-05	298	1000	-86.32	-91.31	
48	C <sub>6</sub> H <sub>12</sub> O	Methyl isobutyl ketone	-256.000	-1.2779E-01	6.2353E-05	298	1000	-288.49	-304.30	

**TABLE C-11—(continued)**

No.	Formula	Substance	$\Delta H_f^\circ = A + BT + CT^2$			$(\Delta H_f^\circ - \text{kjoule/mol}, T - \text{K})$			
			A	B	C	$T_{\min}$	$T_{\max}$	$\Delta H_f^\circ$ at 298 K	$\Delta H_f^\circ$ at 500 K
49	CH <sub>5</sub> N	Methylamine	-7.489	-6.0538E-02	2.7800E-05	298	1000	-23.01	-30.81
50	C <sub>8</sub> H <sub>10</sub>	<i>m</i> -Xylene	46.618	-1.1480E-01	5.3371E-05	298	1000	17.24	2.56
51	C <sub>10</sub> H <sub>8</sub>	Naphthalene	173.657	-8.9278E-02	4.2685E-05	298	1000	150.96	139.69
52	C <sub>4</sub> H <sub>10</sub>	<i>n</i> -Butane	-98.186	-1.0974E-01	5.2254E-05	298	1000	-126.15	-139.99
53	C <sub>10</sub> H <sub>22</sub>	<i>n</i> -Decane	-191.468	-2.3050E-01	1.1559E-04	298	1000	-249.66	-277.82
54	C <sub>7</sub> H <sub>16</sub>	<i>n</i> -Heptane	-144.670	-1.7028E-01	8.4057E-05	298	1000	-187.78	-208.79
55	C <sub>6</sub> H <sub>14</sub>	<i>n</i> -Hexane	-129.114	-1.5013E-01	7.3458E-05	298	1000	-167.19	-185.82
56	C <sub>6</sub> H <sub>3</sub> NO <sub>2</sub>	Nitrobenzene	—	—	—	—	—	67.60	—
57	C <sub>9</sub> H <sub>20</sub>	<i>n</i> -Nonane	-175.883	-2.1036E-01	1.0501E-04	298	1000	-229.03	-254.81
58	C <sub>8</sub> H <sub>18</sub>	<i>n</i> -Octane	-160.339	-1.9025E-01	9.4491E-05	298	1000	-208.45	-231.84
59	C <sub>5</sub> H <sub>12</sub>	<i>n</i> -Pentane	-113.399	-1.3001E-01	6.2902E-05	298	1000	-146.44	-162.68
60	C <sub>8</sub> H <sub>10</sub>	<i>o</i> -Xylene	46.756	-1.0824E-01	4.9928E-05	298	1000	19.00	5.12
61	C <sub>6</sub> H <sub>6</sub> O	Phenol	-80.956	-6.1053E-02	3.0058E-05	298	1000	-96.36	-103.97
62	C <sub>8</sub> H <sub>10</sub>	<i>p</i> -Xylene	47.806	-1.1644E-01	5.3672E-05	298	1000	17.95	3.01
63	C <sub>5</sub> H <sub>5</sub> N	Pyridine	156.938	-6.7082E-02	3.4853E-05	298	1000	140.16	132.11
64	C <sub>8</sub> H <sub>8</sub>	Styrene	167.879	-8.0354E-02	3.7418E-05	298	1000	147.36	137.06
65	C <sub>7</sub> H <sub>8</sub>	Toluene	74.320	-9.5998E-02	4.7011E-05	298	1000	50.00	38.07
66	C <sub>4</sub> H <sub>6</sub> O <sub>2</sub>	Vinyl acetate	-296.867	-7.5158E-02	3.9850E-05	298	1000	-315.70	-324.48

$\Delta H_f^\circ$ —enthalpy of formation of ideal gas, kjoule/mol  
*A*, *B*, and *C*—regression coefficients for chemical compound  
*T*—temperature, K  
*T*<sub>min</sub>—minimum temperature, K  
*T*<sub>max</sub>—maximum temperature, K

**TABLE C-12 Gibbs Energy of Formation**

$\Delta G_f^\circ = A + BT + CT^2$									
No.	Formula	Substance	A	B	C	T <sub>min</sub>	T <sub>max</sub>	$\Delta G_f^\circ$ at 298 K	$\Delta G_f^\circ$ at 500 K
1	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,1-Trichloroethane	-143.436	2.2407E-01	3.9301E-06	298	1000	-76.19	-30.42
2	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,2-Trichloroethane	-139.616	2.0630E-01	5.9155E-06	298	1000	-77.49	-34.99
3	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,1-Dichloroethane	-131.060	1.9006E-01	1.2939E-05	298	1000	-73.09	-32.79
4	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,2-Dichloroethane	-130.549	1.8562E-01	1.3775E-05	298	1000	-73.85	-34.29
5	C <sub>4</sub> H <sub>6</sub>	1,3-Butadiene	109.172	1.3296E-01	1.9003E-05	298	1000	150.67	180.40
6	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	1,4-Dioxane	-318.550	4.5161E-01	3.0870E-05	298	1000	-180.79	-85.03
7	C <sub>4</sub> H <sub>10</sub> O	1-Butanol ( <i>n</i> -Butanol)	-276.720	4.0989E-01	3.9100E-05	298	1000	-150.67	-62.00
8	C <sub>4</sub> H <sub>8</sub>	1-Butene	-1.692	2.3442E-01	3.1582E-05	298	1000	71.30	123.42
9	C <sub>10</sub> H <sub>20</sub>	1-Decene	-128.709	8.1530E-01	6.7574E-05	298	1000	121.04	295.84
10	C <sub>9</sub> H <sub>20</sub>	1-Nonane ( <i>n</i> -Nonane)	-233.826	8.4477E-01	6.8451E-05	298	1000	24.81	205.67
11	C <sub>8</sub> H <sub>16</sub>	1-Octene	-86.500	6.2135E-01	5.5457E-05	298	1000	104.22	238.04
12	C <sub>3</sub> H <sub>8</sub> O	1-Propanol ( <i>n</i> -Propanol)	-259.317	3.1232E-01	3.3063E-05	298	1000	-162.97	-94.89
13	C <sub>2</sub> H <sub>4</sub> O	Acetaldehyde	-167.052	1.0714E-01	1.8665E-05	298	1000	-133.30	-108.82
14	C <sub>2</sub> H <sub>4</sub> O <sub>2</sub>	Acetic acid	-435.963	1.9346E-01	1.6362E-05	298	1000	-376.69	-335.14
15	C <sub>4</sub> H <sub>6</sub> O <sub>3</sub>	Acetic anhydride	-578.076	3.3162E-01	2.5188E-05	298	1000	-476.68	-405.97
16	C <sub>3</sub> H <sub>6</sub> O	Acetone	-218.77	2.1177E-01	2.6619E-05	298	1000	-153.05	-106.24
17	C <sub>3</sub> H <sub>4</sub> O <sub>2</sub>	Acrylic acid	-337.271	1.6672E-01	1.5209E-05	298	1000	-286.06	-250.11
18	C <sub>6</sub> H <sub>7</sub> N	Aniline	84.822	2.6707E-01	2.2598E-05	298	1000	166.69	224.01
19	C <sub>6</sub> H <sub>6</sub>	Benzene	81.512	1.5282E-01	2.6522E-05	298	1000	129.66	164.55
20	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Butyric acid	-472.274	3.8416E-01	2.7623E-05	298	1000	-355.00	-273.29
21	CS <sub>2</sub>	Carbon disulfide	121.242	-1.9750E-01	5.0587E-05	389	717	66.90	35.14
22	CO <sub>2</sub>	Carbon dioxide	-393.360	-3.8212E-03	1.3322E-06	298	1000	-394.38	-394.94
23	CO	Carbon monoxide	-109.885	-9.2218E-02	1.4547E-06	298	1000	-137.28	-155.63
24	CCl <sub>4</sub>	Carbon tetrachloride	-100.838	1.4561E-01	-8.6766E-06	298	1000	-58.24	-30.20
25	CHCl <sub>3</sub>	Chloroform	-101.846	1.1137E-01	8.6302E-07	298	1000	-68.53	-45.95
26	C <sub>6</sub> H <sub>12</sub>	Cyclohexane	-127.917	5.2032E-01	4.4706E-05	298	1000	31.76	143.42
27	C <sub>6</sub> H <sub>12</sub> O	Cyclohexanol	-299.464	5.9552E-01	3.9211E-05	298	1000	-117.91	8.10
28	C <sub>3</sub> H <sub>6</sub>	Cyclopropane	51.643	1.6873E-01	2.4956E-05	298	1000	104.39	142.25
29	CH <sub>2</sub> Cl <sub>2</sub>	Dichloromethane	-95.965	8.8080E-02	8.5438E-06	298	1000	-68.87	-49.79
30	C <sub>4</sub> H <sub>10</sub> O	Diethyl ether	-254.382	4.3005E-01	3.8907E-05	298	1000	-122.34	-29.63
31	C <sub>5</sub> H <sub>10</sub> O	Diethyl ketone	-261.163	4.1025E-01	3.8997E-05	298	1000	-135.06	-46.39
32	C <sub>2</sub> H <sub>7</sub> N	Dimethylamine	-20.837	2.8875E-01	2.7736E-05	298	1000	67.99	130.47
33	C <sub>2</sub> H <sub>6</sub>	Ethane	-85.787	1.6858E-01	2.6853E-05	298	1000	-32.93	5.21
34	C <sub>2</sub> H <sub>6</sub> O	Ethanol	-236.103	2.1904E-01	2.5659E-05	298	1000	-168.28	-120.17
35	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Ethyl acetate	-444.940	3.8444E-01	2.9614E-05	298	1000	-327.40	-245.32
36	C <sub>2</sub> H <sub>5</sub> Cl	Ethyl chloride	-112.949	1.7104E-01	1.9992E-05	298	1000	-60.00	-22.43
37	C <sub>8</sub> H <sub>10</sub>	Ethylbenzene	27.421	3.3327E-01	3.8542E-05	298	1000	130.58	203.69
38	C <sub>2</sub> H <sub>4</sub>	Ethylene	51.752	4.9338E-02	1.7284E-05	298	1000	68.12	80.74
39	C <sub>2</sub> H <sub>6</sub> O <sub>2</sub>	Ethylene glycol	-390.502	2.8378E-01	1.4492E-05	298	1000	-304.47	-244.99
40	C <sub>2</sub> H <sub>4</sub> O	Ethylene oxide	-53.838	1.3048E-01	1.8741E-05	298	1000	-13.10	16.09
41	C <sub>3</sub> H <sub>8</sub> O <sub>3</sub>	Glycerol	-584.906	4.4735E-01	3.0773E-05	298	1000	-448.49	-353.54
42	CHN	Hydrogen cyanide	135.278	-3.6253E-02	2.3053E-06	298	1000	124.68	117.73
43	CH <sub>4</sub>	Methane	-75.262	7.5925E-02	1.8700E-05	298	1000	-50.84	-32.63
44	CH <sub>4</sub> O	Methanol	-201.860	1.2542E-01	2.0345E-05	298	1000	-162.51	-134.06
45	CH <sub>3</sub> Br	Methyl bromide	-55.241	8.0199E-02	1.1207E-05	333	1000	-28.16	-12.34
46	CH <sub>3</sub> Cl	Methyl chloride	-86.903	7.5722E-02	1.4823E-05	298	1000	-62.89	-45.34
47	C <sub>6</sub> H <sub>12</sub> O	Methyl isobutyl ketone	-291.134	5.0824E-01	4.3056E-05	298	1000	-135.36	-26.25
48	CH <sub>5</sub> N	Methylamine	-24.115	1.8179E-01	2.2182E-05	298	1000	32.26	72.32
49	C <sub>8</sub> H <sub>10</sub>	<i>m</i> -Xylene	15.063	3.3452E-01	4.1387E-05	298	1000	118.87	192.67

**TABLE C-12—(continued)**

$\Delta G_f^\circ = A + BT + CT^2$									
No.	Formula	Substance	A	B	C	$T_{\min}$	$T_{\max}$	$\Delta G_f^\circ$ at 298 K	$\Delta G_f^\circ$ at 500 K
50	C <sub>10</sub> H <sub>8</sub>	Naphthalene	148.988	2.4014E-01	3.0705E-05	298	1000	223.59	276.74
51	C <sub>4</sub> H <sub>10</sub>	<i>n</i> -Butane	-128.375	3.6047E-01	3.8256E-05	298	1000	-17.15	61.43
52	C <sub>10</sub> H <sub>22</sub>	<i>n</i> -Decane	-255.000	9.4201E-01	7.4254E-05	298	1000	33.22	234.57
53	C <sub>7</sub> H <sub>16</sub>	<i>n</i> -Heptane	-191.520	6.5052E-01	5.6444E-05	298	1000	7.99	147.85
54	C <sub>6</sub> H <sub>14</sub>	<i>n</i> -Hexane	-170.447	5.5417E-01	5.0303E-05	298	1000	-0.25	119.21
55	C <sub>6</sub> H <sub>5</sub> NO <sub>2</sub>	Nitrobenzene	-	-	-	-	-	158.00	-
56	C <sub>9</sub> H <sub>20</sub>	<i>n</i> -Nonane	-233.826	8.4477E-01	6.8451E-05	298	1000	24.81	205.67
57	C <sub>8</sub> H <sub>18</sub>	<i>n</i> -Octane	-212.692	7.4774E-01	6.2361E-05	298	1000	16.40	176.77
58	C <sub>5</sub> H <sub>12</sub>	<i>n</i> -Pentane	-149.141	4.5748E-01	4.4417E-05	298	1000	-8.37	90.70
59	C <sub>8</sub> H <sub>10</sub>	<i>o</i> -Xylene	17.048	3.3940E-01	3.9428E-05	298	1000	122.09	196.61
60	C <sub>6</sub> H <sub>6</sub> O	Phenol	-97.896	2.1140E-01	1.9980E-05	298	1000	-32.89	12.80
61	C <sub>8</sub> H <sub>10</sub>	<i>p</i> -Xylene	15.763	3.3952E-01	4.2301E-05	298	1000	121.13	196.10
62	C <sub>5</sub> H <sub>5</sub> N	Pyridine	138.330	1.6730E-01	2.0124E-05	298	1000	190.20	227.01
63	C <sub>8</sub> H <sub>8</sub>	Styrene	145.657	2.1917E-01	2.8490E-05	298	1000	213.80	262.36
64	C <sub>7</sub> H <sub>8</sub>	Toluene	47.813	2.3831E-01	3.1916E-05	298	1000	122.01	174.95

$\Delta G_f^\circ$  – Gibbs energy of formation of ideal gas, kJ/mol  
 $T$  – temperature, K  
 $T_{\min}$  – minimum temperature, K  
 $T_{\max}$  – maximum temperature, K

**TABLE C-13 Solubility of Salt Water**

$\log_{10} S = A + BX + CX^2$									
No.	Formula	Substance	$M_w$	$T_B$	A	B	C	S at X = 0	S at X = 34,472
1	C <sub>9</sub> H <sub>20</sub>	1-Nonane (Nonane)	128.257	423.81	-0.9136	-3.8600E-06	1.9300E-12	1.2200E-01	9.0290E-02
2	C <sub>6</sub> H <sub>6</sub>	Benzene	78.113	353.31	3.2443	-3.2714E-06	1.6357E-16	1.7550E+03	1.3540E+03
3	C <sub>6</sub> H <sub>12</sub>	Cyclohexane	84.161	353.90	1.7490	-4.5987E-06	2.2993E-12	5.6100E+01	3.9190E+01
4	C <sub>8</sub> H <sub>10</sub>	Ethylbenzene	106.167	409.17	2.2178	-5.0916E-06	2.5458E-12	1.6510E+02	1.1100E+02
5	C <sub>8</sub> H <sub>10</sub>	<i>m</i> -Xylene	106.67	412.22	2.2405	-6.3514E-06	3.1757E-12	1.7400E+02	1.0600E+02
6	C <sub>7</sub> H <sub>16</sub>	<i>n</i> -Heptane (Heptane)	100.203	371.56	0.3502	-3.8600E-06	1.9300E-12	2.2400E+00	1.6570E+00
7	C <sub>6</sub> H <sub>14</sub>	<i>n</i> -Hexane (Hexane)	86.177	341.93	0.9763	-3.8600E-06	1.9300E-12	9.4700E+00	7.0070E+00
8	C <sub>9</sub> H <sub>20</sub>	<i>n</i> -Nonane (Nonane)	128.257	423.81	-0.9136	-3.8600E-06	1.9300E-12	1.2200E-01	9.0290E-02
9	C <sub>8</sub> H <sub>18</sub>	<i>n</i> -Octane (Octane)	114.230	398.77	-0.3655	-3.8600E-06	1.9300E-12	4.3100E-01	3.1900E-01
10	C <sub>5</sub> H <sub>12</sub>	<i>n</i> -Pentane (Pentane)	72.150	309.22	1.5966	-4.5956E-06	2.2978E-12	3.9500E+01	2.7600E+01
11	C <sub>8</sub> H <sub>10</sub>	<i>o</i> -Xylene	106.167	417.46	2.3441	-6.8318E-06	3.4159E-12	2.2080E+02	1.2960E+02
12	C <sub>8</sub> H <sub>10</sub>	<i>p</i> -Xylene	106.67	411.44	2.3047	-7.6669E-06	3.8335E-12	2.0170E+02	1.1090E+02
13	C <sub>7</sub> H <sub>8</sub>	Toluene	92.140	383.73	2.7343	-3.8393E-06	1.9196E-12	5.4240E+02	4.0200E+02

$S$  – solubility in salt water at 25°C, parts per million by weight, ppm (wt)  
 $X$  – concentration of salt (NaCl) in water, parts per million by weight, ppm (wt)  
 $A$ ,  $B$ , and  $C$  – correlation constants for compound  
 $M_w$  – molecular weight of compound, g/mol  
 $T_B$  – boiling point of compound, K  
 $X = 0$  for water without salt  
 $X = 34,472$  for sea water

**TABLE C-14 Solubility of Organic Compounds in Water as a Function of Temperature**

$\log_{10} S = A + B/T + CT^2$									
No.	Formula	Substance	M <sub>w</sub>	T <sub>B</sub>	A	B	C	S at 25°C	S at 100°C
1	C <sub>9</sub> H <sub>20</sub>	1-Nonane (Nonane)	128.257	423.81	18.222	-11404.793	1.6993E+06	1.2200E-01	7.2870E-01
2	C <sub>6</sub> H <sub>6</sub>	Benzene	78.113	353.31	11.994	-5214.537	7.76966E+05	1.7550E+03	3.9780E+03
3	C <sub>6</sub> H <sub>12</sub>	Cyclohexane	84.161	353.90	6.403	-2773.806	4.13297E+05	5.6100E+01	8.6640E+01
4	C <sub>8</sub> H <sub>10</sub>	Ethylbenzene	106.167	409.17	13.365	-6643.672	9.89907E+05	1.6510E+02	4.6780E+02
5	C <sub>8</sub> H <sub>10</sub>	m-Xylene	106.167	412.22	14.743	-7451.440	1.11026E+06	1.7400E+02	5.5930E+02
6	C <sub>7</sub> H <sub>16</sub>	n-Heptane (Heptane)	100.203	371.56	12.811	-7426.389	1.10653E+06	2.2400E+00	7.1780E+00
7	C <sub>6</sub> H <sub>14</sub>	n-Hexane (Hexane)	86.177	341.93	10.992	-5969.484	8.89453E+05	9.4700E+00	2.4120E+01
8	C <sub>9</sub> H <sub>20</sub>	n-Nonane (Nonane)	128.257	423.81	18.222	-11404.793	1.69931E+06	1.2200E-01	7.2870E-01
9	C <sub>8</sub> H <sub>18</sub>	n-Octane (Octane)	114.230	398.77	16.865	-10269.143	1.53010E+06	4.3100E-01	2.1560E+00
10	C <sub>5</sub> H <sub>12</sub>	n-Pentane (Pentane)	72.150	309.22	9.036	-4433.922	6.60654E+05	3.9500E+01	7.9120E+01
11	C <sub>8</sub> H <sub>10</sub>	o-Xylene	106.167	417.46	13.112	-6417.607	9.56223E+05	2.2080E+02	6.0390E+02
12	C <sub>8</sub> H <sub>10</sub>	p-Xylene	106.167	411.44	13.731	-6810.190	1.01472E+06	2.0170E+02	5.8610E+02
13	C <sub>7</sub> H <sub>8</sub>	Toluene	92.140	383.73	15.471	-7591.270	1.13110E+06	5.4240E+02	1.7810E+03
14	C <sub>5</sub> H <sub>10</sub>	Cyclopentane	70.134	322.41	8.665	-3850.808	5.73770E+05	1.6000E+02	2.9240E+02
15	C <sub>5</sub> H <sub>12</sub>	2-Methylbutane (isopentane)	72.150	301.15	9.473	-4645.254	6.92143E+05	4.7800E+01	9.8870E+01
16	C <sub>6</sub> H <sub>12</sub>	Methylcyclopentane	84.161	344.97	12.430	-6436.630	9.59058E+05	4.2640E+01	1.1700E+02
17	C <sub>7</sub> H <sub>14</sub>	Ethylcyclopentane	98.188	376.60	14.360	-7974.533	1.18821E+06	9.5560E+00	3.3310E+01
18	C <sub>7</sub> H <sub>14</sub>	Methylcyclohexane	98.188	374.08	13.091	-7084.522	1.05559E+06	1.6000E+01	4.8570E+01
19	C <sub>9</sub> H <sub>12</sub>	Propylbenzene	120.194	430.17	11.722	-5962.906	8.88473E+05	5.2160E+01	1.3270E+02
20	C <sub>9</sub> H <sub>12</sub>	Cumene	120.194	426.31	13.052	-6766.120	1.00815E+06	5.0000E+01	1.4450E+02
21	C <sub>9</sub> H <sub>18</sub>	Butylcyclopentane	126.241	429.76	20.741	-12579.687	1.87437E+06	4.3080E-01	3.0920E+00
22	C <sub>9</sub> H <sub>18</sub>	Propylcyclohexane	126.241	429.90	18.885	-11356.483	1.69212E+06	6.7730E-01	4.0120E+00
23	C <sub>10</sub> H <sub>14</sub>	Butylbenzene	134.221	456.43	9.533	-5001.869	7.45278E+05	1.3820E+01	3.0270E+01
24	C <sub>10</sub> H <sub>14</sub>	m-Diethylbenzene	134.221	454.25	10.975	-5686.134	8.47234E+05	2.7230E+01	6.6290E+01
25	C <sub>10</sub> H <sub>14</sub>	o-Diethylbenzene	134.221	457.02	10.908	-5686.174	8.47240E+05	2.3320E+01	5.6810E+01
26	C <sub>10</sub> H <sub>14</sub>	p-Diethylbenzene	134.221	456.94	10.910	-5686.175	8.47240E+05	2.3420E+01	5.7070E+01
27	C <sub>10</sub> H <sub>22</sub>	Decane	142.284	446.86	17.396	-11133.010	1.65882E+06	5.2000E-02	2.9790E-01
28	C <sub>11</sub> H <sub>16</sub>	Pentylbenzene	148.247	478.62	9.504	-5316.406	7.92144E+05	3.8400E+00	8.8240E+00
29	C <sub>11</sub> H <sub>22</sub>	Pentylcyclohexane	154.295	476.87	22.316	-14034.923	2.09120E+06	5.8500E-02	5.2790E-01
30	C <sub>11</sub> H <sub>24</sub>	Undecane	156.311	468.70	28.051	-18123.096	2.70034E+06	4.4000E-03	7.5240E-02
31	C <sub>12</sub> H <sub>18</sub>	Hexylbenzene	162.274	499.26	11.076	-6596.944	9.82945E+05	1.0180E+00	2.8590E+00
32	C <sub>12</sub> H <sub>26</sub>	Dodecane	170.337	488.61	23.755	-15607.165	2.32547E+06	3.7000E-03	4.2720E-02
33	C <sub>13</sub> H <sub>28</sub>	Tridecane	184.364	507.77	22.937	-15279.183	2.27660E+06	2.0000E-03	2.1910E-02
34	C <sub>14</sub> H <sub>22</sub>	1-Phenyloctane	190.328	537.56	9.743	-6166.179	9.18761E+05	2.4970E-01	6.5570E-01
35	C <sub>14</sub> H <sub>30</sub>	Tetradecane	198.391	526.14	17.765	-12172.057	1.81364E+06	2.2000E-03	1.4810E-02
36	C <sub>15</sub> H <sub>32</sub>	Pentadecane	212.418	543.59	18.533	-12808.889	1.90852E+06	1.1000E-03	8.1890E-03
37	C <sub>16</sub> H <sub>34</sub>	Hexadecane	226.445	560.50	16.462	-11626.514	1.73235E+06	9.0000E-04	5.5670E-03
38	C <sub>17</sub> H <sub>36</sub>	Heptadecane	240.471	576.00	10.581	-8007.275	1.19308E+06	1.4000E-03	4.9080E-03

**TABLE C-15 Henry's Law Constant for Gases in Water as a Function of Temperature**

$\log_{10} H = A + B/T + C \log T + DT$											
No.	Formula	Substance	$M_w$	A	B	C	D	$T_{\min}$	$T_{\max}$	$H$ at $25^\circ C$	$H$ at $T_{\max}$
1	Ar	Argon	39.948	65.3235	-3.24690E+03	-2.01398E+01	0.00000E+00	273.15	348.15	39696.1	64077.9
2	CF <sub>4</sub>	Carbon tetrafluoride	88.005	1507.2737	-4.19725E+04	-5.99350E+02	4.09087E-01	273.15	323.15	259777.3	374833.0
3	CH <sub>4</sub>	Methane	16.043	146.8858	-5.76834E+03	-5.19144E+01	1.84936E-02	273.15	360.95	39240.1	64820.5
4	CO	Carbon monoxide	28.010	74.5962	-3.60330E+03	-2.33376E+01	0.00000E+00	273.15	353.15	57978.5	85034.9
5	COS	Carbonyl sulfide	60.076	96.0704	-5.22240E+03	-3.03658E+01	0.00000E+00	273.15	303.15	2607.5	3060.5
6	CO <sub>2</sub>	Carbon dioxide	44.010	69.4237	-3.79646E+03	-2.16694E+01	4.78857E-04	273.15	353.15	1635.2	4263.7
7	C <sub>2</sub> H <sub>2</sub>	Acetylene	26.038	67.9714	-3.54393E+03	-2.14030E+01	0.00000E+00	274.15	343.15	1332.6	2381.4
8	C <sub>2</sub> H <sub>3</sub> Cl	Vinyl chloride	62.499	61.1868	-3.43853E+03	-1.90984E+01	2.51906E-03	273.15	323.15	1403.3	2719.5
9	C <sub>2</sub> H <sub>4</sub>	Ethylene	28.054	107.7298	-6.33990E+03	-3.17169E+01	-1.32533E-02	283.15	360.95	10784.4	18502.9
10	C <sub>2</sub> H <sub>6</sub>	Ethane	30.070	108.9263	-5.51363E+03	-3.47413E+01	0.00000E+00	275.15	323.15	29402.4	48324.7
11	C <sub>3</sub> H <sub>6</sub>	Cyclopropane	42.081	-141.9730	5.87466E+03	5.09015E+01	0.00000E+00	298.15	361.15	4820.7	30453.0
12	C <sub>3</sub> H <sub>6</sub>	Propylene	42.081	-2570.0227	6.86740E+04	1.03384E+03	-7.19694E-01	283.15	360.95	8182.3	29043.9
13	C <sub>3</sub> H <sub>8</sub>	Propane	44.096	2874.1130	-8.56732E+04	-1.12809E+03	7.01580E-01	283.15	360.95	35420.1	90695.7
14	C <sub>4</sub> H <sub>8</sub>	1-Butene	56.107	12.9563	-2.78226E+03	0.00000E+00	0.00000E+00	311.15	378.15	—	399742.0
15	C <sub>4</sub> H <sub>8</sub>	Isobutene	56.107	103.9814	-5.54785E+03	-3.28909E+01	0.00000E+00	273.15	343.15	9714.4	26279.4
16	C <sub>4</sub> H <sub>10</sub>	n-Butane	58.123	121.8305	-6.34244E+03	-3.87599E+01	0.00000E+00	273.15	349.15	44563.3	125372.9
17	C <sub>4</sub> H <sub>10</sub>	Isobutane	58.123	161.2644	-7.94950E+03	-5.24651E+01	0.00000E+00	278.15	318.15	60280.3	94825.0
18	ClO <sub>2</sub>	Chloride dioxide	67.452	-24.6413	-6.21813E+01	1.07454E+01	0.00000E+00	283.15	333.15	54.814	190.01
19	Cl <sub>2</sub>	Chlorine	70.905	232.4396	-8.21980E+03	-8.69997E+01	4.42155E-02	283.15	353.15	599.8	1279.3
20	H <sub>2</sub>	Hydrogen	2.016	54.6946	-2.40098E+03	-1.68893E+01	0.00000E+00	273.15	345.15	70832.1	74664.1
21	H <sub>2</sub> S	Hydrogen sulfide	34.082	10.8191	-1.51009E+03	-3.99300E-01	-6.81842E-03	273.15	353.15	541.18	1311.5
22	He	Helium	4.003	46.0252	-1.84993E+03	-1.40094E+01	0.00000E+00	273.15	348.15	142950.4	126766.7
23	Kr	Krypton	83.800	77.5359	-3.95280E+03	-2.42207E+01	0.00000E+00	273.15	353.15	22161.3	42614.9
24	NO	Nitric oxide	30.006	30.2512	-2.42215E+03	-5.70490E+00	-1.19149E-02	273.15	353.15	28739.1	44559.0
25	N <sub>2</sub>	Nitrogen	28.013	78.8622	-3.74498E+03	-2.47981E+01	0.00000E+00	273.15	350.15	87143.1	118634.5
26	N <sub>2</sub> O	Nitrous oxide	44.013	68.8882	-3.85775E+03	-2.12530E+01	0.00000E+00	273.15	313.15	2291.4	3363.7
27	Ne	Neon	20.180	60.7869	-2.65134E+03	-1.89157E+01	0.00000E+00	273.15	348.15	12262.8	123620.2
28	O <sub>2</sub>	Oxygen	31.999	77.8881	-3.79901E+03	-2.44526E+01	0.00000E+00	273.15	348.15	43630.9	66575.2
29	PH <sub>3</sub>	Phosphine	33.998	67.5831	-3.57648E+03	-2.09165E+01	0.00000E+00	298.15	323.15	6776.6	10656.0
30	Rn	Radon	222.000	109.3341	-5.64696E+03	-3.50047E+01	0.00000E+00	273.15	323.15	5987.3	10449.8
31	SF <sub>6</sub>	Sulfur hexafluoride	146.056	191.8514	-9.19008E+03	-6.29116E+01	0.00000E+00	276.15	323.15	22752.1	348071.1
32	SO <sub>2</sub>	Sulfur dioxide	64.065	22.3423	-1.98711E+03	-5.68540E+00	0.00000E+00	283.15	323.15	40.674	84.355
33	Xe	Xenon	131.290	87.3918	-4.56921E+03	-2.74664E+01	0.00000E+00	273.15	348.15	12669.1	28466.9

*H* – Henry's Law constant (atm/mol fraction) for gas in water at pressure of 1 atm (101.325 kPa)

1. Henry's Law constant is applicable at very low concentration ( $x \ll 1$ ). Most compounds in the tabulation are at very low concentrations (range of 0.0001–0.00001).

2. A few compounds (e.g., ClO<sub>2</sub> and SO<sub>2</sub>) are not at very low concentrations. The tabulated data for these compounds should be considered rough values.

3. For those compounds that react with water, all chemical species of the gas and its reaction products are included.

**TABLE C-16 Solubility of Selected Gases in Water as a Function of Temperature**

No.	Formula	Substance	M <sub>w</sub> , g/mol	$\log x_{wt} = A + B/T + C \log T + DT$					x <sub>wt</sub> at 25°C	x <sub>wt</sub> at T <sub>max</sub>	
				A	B	C	D	T <sub>min</sub>	T <sub>max</sub>		
1	Ar	Argon	39.948	-106.3705	2.87724E+04	4.24672E+02	-3.02604E-01	273.15	348.15	5.4131E-05	2.1760E-05
2	CF <sub>4</sub>	Carbon tetrafluoride	88.005	-1923.2642	5.23082E+04	7.69721E+02	-5.41819E-01	273.15	323.15	1.8227E-05	1.1456E-05
3	CH <sub>4</sub>	Methane	16.043	-1388.5451	3.78016E+04	5.55218E+02	-3.91585E-01	273.15	360.95	2.1949E-05	6.0342E-06
4	CO	Carbon monoxide	28.010	-1316.0154	3.56367E+04	5.26642E+02	-3.73092E-01	273.15	353.15	2.5937E-05	9.9946E-06
5	COS	Carbonyl sulfide	60.076	-310.8791	1.03967E+04	1.19013E+02	-7.17370E-02	273.15	303.15	1.2373E-03	1.0429E-03
6	CO <sub>2</sub>	Carbon dioxide	44.010	-1307.3028	3.57336E+04	5.23648E+02	-3.72704E-01	273.15	353.15	1.4429E-03	3.1311E-04
7	C <sub>2</sub> H <sub>2</sub>	Acetylene	26.038	-904.6169	2.48670E+04	3.61545E+02	-2.56198E-01	273.15	343.15	1.0504E-03	4.2360E-04
8	C <sub>2</sub> H <sub>3</sub> Cl	Vinyl chloride	62.499	-466.7040	1.34792E+04	1.85229E+02	-1.32359E-01	273.15	323.15	2.3906E-03	1.1202E-03
9	C <sub>2</sub> H <sub>4</sub>	Ethylene	28.054	-2717.3688	7.56485E+04	1.08310E+03	-7.38856E-01	283.15	360.96	1.3646E-04	3.1695E-05
10	C <sub>2</sub> H <sub>6</sub>	Ethane	30.070	-546.9272	1.64179E+04	2.13803E+02	-1.38988E-01	275.15	323.15	5.5018E-05	3.0358E-05
11	C <sub>3</sub> H <sub>6</sub>	Cyclopropane	42.081	-3892.0392	1.04154E+05	1.56543E+03	-1.12081E+00	298.15	361.15	4.7881E-04	2.8627E-05
12	C <sub>3</sub> H <sub>6</sub>	Propylene	42.081	-39.4402	6.34551E+02	1.75424E+01	-3.24155E-02	283.15	360.95	2.6979E-04	3.0288E-05
13	C <sub>3</sub> H <sub>8</sub>	Propane	44.096	-5554.9049	1.56979E+05	2.20789E+03	-1.47267E+00	283.15	360.95	6.5123E-05	1.0153E-05
14	C <sub>4</sub> H <sub>8</sub>	1-Butene	56.107	-2991.5526	8.40296E+04	1.19372E+03	-8.29111E-01	311.15	348.15	—	2.0827E-05
15	C <sub>4</sub> H <sub>8</sub>	Isobutene	56.107	-919.4382	2.63056E+04	3.64670E+02	-2.50383E-01	273.15	343.15	3.1083E-04	8.2769E-05
16	C <sub>4</sub> H <sub>10</sub>	n-Butane	58.123	-1160.1487	3.29912E+04	4.60393E+02	-3.14847E-01	273.15	349.15	7.0137E-05	1.5785E-05
17	C <sub>4</sub> H <sub>10</sub>	Isobutane	58.123	-566.9083	1.80393E+04	2.18486E+02	-1.29161E-01	278.15	318.15	5.1857E-05	3.0814E-05
18	ClO <sub>2</sub>	Chloride dioxide	67.452	-569.1018	1.46852E+04	2.32419E+02	-1.89385E-01	283.15	333.15	6.2050E-02	1.5612E-02
19	Cl <sub>2</sub>	Chlorine	70.905	-1847.3759	5.05934E+04	7.39612E+02	-5.18667E-01	283.15	353.15	6.2695E-03	1.6680E-03
20	H <sub>2</sub>	Hydrogen	2.016	-938.5889	2.49279E+04	3.75683E+02	-2.69793E-01	273.15	345.15	1.5319E-06	1.0072E-06
21	H <sub>2</sub> S	Hydrogen sulfide	34.082	-1247.8348	3.34178E+02	5.01994E+02	-3.65161E-01	273.15	353.15	3.3698E-03	7.8792E-04
22	He	Helium	4.003	-1043.1419	2.73774E+04	4.18569E+02	-3.02623E-01	273.15	348.15	1.5063E-06	1.1022E-06
23	Kr	Krypton	83.800	-1318.1442	3.59766E+04	5.27392E+02	-3.73004E-01	273.15	353.15	2.0297E-04	5.9664E-05
24	NO	Nitric oxide	30.006	-1271.6006	3.44544E+04	5.08993E+02	-3.61166E-01	273.15	353.15	5.6051E-05	2.0432E-05
25	N <sub>2</sub>	Nitrogen	28.013	-1163.3807	3.16107E+04	4.64916E+02	-3.28050E-01	273.15	350.15	1.7282E-05	7.8317E-06
26	N <sub>2</sub> O	Nitrous oxide	44.013	-371.1999	1.12586E+04	1.45406E+02	-9.84292E-02	273.15	313.15	1.0319E-03	6.7314E-04
27	Ne	Neon	20.180	-1057.1943	2.81787E+04	4.23473E+02	-3.02621E-01	273.15	348.15	8.8522E-06	5.6978E-06
28	O <sub>2</sub>	Oxygen	31.999	-1074.0450	2.93249E+04	4.28990E+02	-3.02607E-01	273.15	348.15	3.9450E-05	1.6776E-05
29	PH <sub>3</sub>	Phosphine	33.998	-864.2313	2.41128E+04	3.44136E+02	-2.40666E-01	298.15	323.15	2.6973E-04	1.5551E-04
30	Rn	Radon	222.000	-514.8565	1.57032E+04	2.01357E+02	-1.29996E-01	273.15	323.15	1.9910E-03	1.0349E-03
31	SF <sub>6</sub>	Sulfur hexafluoride	146.056	-640.4119	2.03918E+04	2.46507E+02	-1.42242E-01	276.15	323.15	3.4534E-05	2.0471E-05
32	SO <sub>2</sub>	Sulfur dioxide	64.065	-359.9136	9.82234E+03	1.45582E+02	-1.15277E-01	283.15	323.15	7.8097E-02	3.5720E-02
33	Xe	Xenon	131.290	-1081.4609	3.00530E+04	4.31418E+02	-3.02221E-01	273.15	348.15	5.5714E-04	1.6095E-04

x<sub>wt</sub> – solubility (weight fraction) of compound in water at pressure of 1 atm (101.325 kPa)

1. Most compounds in the tabulation are at very low concentrations (range of 0.0001–0.000001). At very low concentrations, Henry's Law is applicable ( $p = Hx$  where  $x \ll 1$ ).

2. A few compounds (e.g., ClO<sub>2</sub> and SO<sub>2</sub>) are not at very low concentrations. The tabulation data for these compounds should be considered rough values.

3. For those compounds that react with water, all chemical species of the gas and its reaction products are included.

**TABLE C-17 Solubility of Sulfur Compounds in Water as a Function of Boiling Point for Mercaptans and Aromatics**

No.	Formula	Substance	Type	$T_B$ , K	S at 25°C, ppm (wt)	S at 25°C, ppm (mol)	Code
1	CH <sub>3</sub> S	Methyl mercaptan	mercaptan	279.11	2.3300E+04	8.8548E+03	1
2	C <sub>2</sub> H <sub>6</sub> S	Ethyl mercaptan	mercaptan	308.15	1.5000E+04	4.3962E+03	1
3	C <sub>3</sub> H <sub>8</sub> S	Propyl mercaptan	mercaptan	340.87	3.6899E+03	8.7535E+02	2
4	C <sub>3</sub> H <sub>8</sub> S	Isopropyl mercaptan	mercaptan	325.71	6.1810E+03	1.4691E+03	2
5	C <sub>4</sub> H <sub>10</sub> S	Butyl mercaptan	mercaptan	371.61	6.0000E+02	1.1992E+02	1
6	C <sub>4</sub> H <sub>10</sub> S	Isobutyl mercaptan	mercaptan	361.64	1.6210E+03	3.2425E+02	2
7	C <sub>4</sub> H <sub>10</sub> S	sec-Butyl mercaptan	mercaptan	358.13	1.8786E+03	3.7586E+02	2
8	C <sub>4</sub> H <sub>10</sub> S	tert-Butyl mercaptan	mercaptan	337.37	4.1849E+03	8.3883E+02	2
9	C <sub>5</sub> H <sub>12</sub> S	Penyl Mercaptan	mercaptan	399.79	2.7257E+02	4.7132E+01	2
10	C <sub>5</sub> H <sub>12</sub> S	2-Pentanethiol	mercaptan	385.55	5.4872E+02	9.4907E+01	2
11	C <sub>5</sub> H <sub>12</sub> S	3-Pentanethiol	mercaptan	387.05	5.1059E+02	8.8310E+01	2
12	C <sub>6</sub> H <sub>12</sub> S	2-Methyl-1-butanethiol	mercaptan	392.15	3.9851E+02	6.8919E+01	2
13	C <sub>6</sub> H <sub>12</sub> S	3-Methyl-1-butanethiol	mercaptan	391.50	4.1147E+02	7.1161E+01	2
14	C <sub>6</sub> H <sub>12</sub> S	2-Methyl-2-butanethiol	mercaptan	372.28	1.0171E+03	1.7599E+02	2
15	C <sub>6</sub> H <sub>12</sub> S	3-Methyl-2-butanethiol	mercaptan	382.91	6.2217E+02	1.0762E+02	2
16	C <sub>6</sub> H <sub>12</sub> S	2,2-Dimethyl-1-propanethiol	mercaptan	376.83	8.2658E+02	1.4300E+02	2
17	C <sub>6</sub> H <sub>14</sub> S	Hexyl mercaptan	mercaptan	425.81	7.0459E+01	1.0737E+01	2
18	C <sub>6</sub> H <sub>14</sub> S	2-Hexanethiol	mercaptan	412.05	1.4553E+02	2.2177E+01	2
19	C <sub>8</sub> H <sub>18</sub> S	Octyl mercaptan	mercaptan	472.19	5.6934E+00	7.0119E-01	2
20	C <sub>8</sub> H <sub>18</sub> S	2-Octanethiol	mercaptan	459.55	1.1319E+01	1.3940E+00	2
21	C <sub>8</sub> H <sub>18</sub> S	tert-Octyl mercaptan	mercaptan	429.00	5.9408E+01	7.3169E+00	2
22	C <sub>9</sub> H <sub>20</sub> S	Nonyl mercaptan	mercaptan	492.95	1.8771E+00	2.1095E-01	2
23	C <sub>9</sub> H <sub>20</sub> S	2-Nonanethiol	mercaptan	481.35	3.4756E+00	3.9059E-01	2
24	C <sub>10</sub> H <sub>22</sub> S	Decyl mercaptan	mercaptan	512.35	6.9284E-01	7.1599E-02	2
25	C <sub>10</sub> H <sub>22</sub> S	2-Decanethiol	mercaptan	501.45	1.2056E+00	1.2459E-01	2
26	C <sub>11</sub> H <sub>24</sub> S	1-Undecanethiol	mercaptan	530.55	2.8701E-01	2.7451E-02	2
27	C <sub>11</sub> H <sub>24</sub> S	2-Undecanethiol	mercaptan	520.45	4.6457E-01	4.4434E-02	2
28	C <sub>12</sub> H <sub>26</sub> S	1-Dodecanethiol	mercaptan	547.75	1.3292E-01	1.1832E-02	2
29	C <sub>12</sub> H <sub>26</sub> S	2-Dodecanethiol	mercaptan	538.35	2.0073E-01	1.7868E-02	2
30	C <sub>13</sub> H <sub>28</sub> S	1-Tridecanethiol	mercaptan	563.96	6.8950E-02	5.7399E-03	2
30	C <sub>13</sub> H <sub>28</sub> S	2-Tridecanethiol	mercaptan	555.15	9.7619E-02	8.1266E-03	2
31	C <sub>14</sub> H <sub>30</sub> S	1-Tetradecanethiol	mercaptan	579.36	3.9756E-02	3.1082E-03	2
32	C <sub>14</sub> H <sub>30</sub> S	2-Tetradecanethiol	mercaptan	571.15	5.2816E-02	4.1293E-03	2

**Code:** 1 – experimental data; 2 – estimate

**$T_B$  – boiling point, K**

**S – solubility in water, ppm (wt); ppm = parts per million**

**T – temperature, °C**

**TABLE C-18 Solubility of Naphthenes in Water**

No.	Formula	Substance	$M_w$	$\log_{10} S = A + B/T + C/T^2$					
				$T_B$	$A$	$B$	$C$	$S$ at 25°C	$S$ at 100°C
1	C <sub>5</sub> H <sub>10</sub>	Cyclopentane	70.134	322.4	8.665	-3850.808	5737.704E+02	1600E-01	2924E-01
2	C <sub>6</sub> H <sub>12</sub>	Cyclohexane	84.161	353.9	6.403	-2773.806	4132.971E+02	5610E-02	8664E-02
3	C <sub>6</sub> H <sub>12</sub>	Methylcyclopentane	84.161	345.0	12.430	-6436.630	9590.579E+02	4264E-02	1170E-01
4	C <sub>7</sub> H <sub>14</sub>	Ethylcyclopentane	98.188	376.6	14.360	-7974.533	1188.205E+03	9556E-03	3331E-02
5	C <sub>7</sub> H <sub>14</sub>	1,1-Dimethylcyclopentane	98.188	361.0	12.634	-6723.665	1001.826E+03	2253E-02	6460E-02
6	C <sub>7</sub> H <sub>14</sub>	C-1,2-Dimethylcyclopentane	98.188	372.7	13.907	-7647.583	1139.49E+03	1191E-02	3944E-02
7	C <sub>7</sub> H <sub>14</sub>	T-1,2-Dimethylcyclopentane	98.188	365.0	13.057	-7031.220	1047.652E+03	1816E-02	5472E-02
8	C <sub>7</sub> H <sub>14</sub>	C-1,3-Dimethylcyclopentane	98.188	363.9	12.940	-6946.544	1035.035E+03	1927E-02	5721E-02
9	C <sub>7</sub> H <sub>14</sub>	T-1,3-Dimethylcyclopentane	98.188	364.9	13.042	-7020.299	1046.025E+03	1830E-02	5505E-02
10	C <sub>7</sub> H <sub>14</sub>	Methylcyclohexane	98.188	374.1	13.091	-7084.522	1055.594E+03	1600E-02	4857E-02
11	C <sub>8</sub> H <sub>16</sub>	Propylcyclopentane	112.214	404.1	17.327	-10142.370	1511.213E+03	2040E-03	9996E-03
12	C <sub>8</sub> H <sub>16</sub>	Ethylcyclohexane	112.214	404.9	19.501	-11345.870	1690.535E+03	2916E-03	1724E-02
13	C <sub>8</sub> H <sub>16</sub>	1,1-Dimethylcyclohexane	112.214	392.7	14.606	-8241.599	1227.998E+03	5989E-03	2181E-02
14	C <sub>8</sub> H <sub>16</sub>	C-1,2-Dimethylcyclohexane	112.214	402.9	11.610	-6455.995	9619.432E+02	6000E-03	1649E-02
15	C <sub>8</sub> H <sub>16</sub>	T-1,2-Dimethylcyclohexane	112.214	396.7	15.082	-8586.692	1279.417E+03	4734E-03	1816E-02
16	C <sub>8</sub> H <sub>16</sub>	C-1,3-Dimethylcyclohexane	112.214	393.3	14.671	-8288.887	1235.044E+03	5799E-03	2126E-02
17	C <sub>8</sub> H <sub>16</sub>	T-1,3-Dimethylcyclohexane	112.214	397.6	15.189	-8663.467	1290.857E+03	4493E-03	1748E-02
18	C <sub>8</sub> H <sub>16</sub>	C-1,4-Dimethylcyclohexane	112.214	397.6	15.184	-8660	1290.340E+03	4504E-03	1750E-02
19	C <sub>8</sub> H <sub>16</sub>	T-1,4-Dimethylcyclohexane	112.214	392.5	17.766	-10240.17	1525.786E+03	3840E-03	1911E-02
20	C <sub>9</sub> H <sub>18</sub>	Butylcyclopentane	126.241	429.8	20.741	-12579.69	1874.373E+03	4308E-04	3092E-03
21	C <sub>9</sub> H <sub>18</sub>	Propylcyclopentane	126.241	429.9	18.885	-11356.48	1692.116E+03	6773E-04	4012E-03
22	C <sub>9</sub> H <sub>18</sub>	C-C-1,5-Trimethylcyclohexane	126.241	411.7	16.852	-9869.544	1470.562E+03	1962E-03	9205E-03
23	C <sub>9</sub> H <sub>18</sub>	C-T-1,3,5-Trimethylcyclohexane	126.241	413.7	17.089	-10041.88	1496.240E+03	1740E-03	8387E-03
24	C <sub>10</sub> H <sub>20</sub>	1-Cyclopentylpentane	140.268	453.8	22.985	-1459.12	2124.608E+03	1150E-04	1073E-03
25	C <sub>10</sub> H <sub>20</sub>	Butylcyclohexane	140.268	454.1	21.087	-13014.7	1939.191E+03	1781E-04	1368E-03
26	C <sub>11</sub> H <sub>22</sub>	1-Cyclopentylhexane	154.295	476.3	24.302	-15330.35	2284.223E+03	3800E-05	4200E-04
27	C <sub>11</sub> H <sub>22</sub>	Pentylcyclohexane	154.295	476.9	22.316	-14034.92	2091.204E+03	5850E-05	5279E-04
28	C <sub>12</sub> H <sub>24</sub>	1-Cyclopentylheptane	168.322	497.3	24.467	-15649.31	2331.747E+03	1620E-05	1883E-04
29	C <sub>12</sub> H <sub>24</sub>	1-Cyclohexylhexane	168.322	497.9	22.412	-14310.59	2132.278E+03	2520E-05	2370E-04
30	C <sub>13</sub> H <sub>26</sub>	1-Cyclopentyloctane	182.348	516.9	23.457	-15199.39	2264.709E+03	9000E-06	9750E-05
31	C <sub>13</sub> H <sub>26</sub>	1-Cyclohexylheptane	182.348	518.1	21.326	-13817.11	2058.750E+03	1390E-05	1211E-04
32	C <sub>14</sub> H <sub>28</sub>	1-Cyclopentylnonane	196.375	535.3	21.217	-13948.64	2078.347E+03	6500E-06	5787E-05
33	C <sub>14</sub> H <sub>28</sub>	1-Cyclohexyloctane	196.375	536.8	19.067	-12553.52	1870.474E+03	1010E-05	7217E-05
34	C <sub>15</sub> H <sub>30</sub>	1-Cyclopentyldecane	210.402	552.5	17.98	-12044.88	1794.686E+03	5900E-06	3892E-05
35	C <sub>15</sub> H <sub>30</sub>	1-Cyclohexylnonane	210.402	554.7	15.592	-10498.13	1564.222E+03	9500E-06	4922E-05

*S* – solubility in salt water at 25°C, parts per million by weight, ppm (wt)  
Temperature range is 298–393 K (25–120°C)  
*A, B, and C* – correlation constants for compound  
*M<sub>w</sub>* – molecular weight of compound, g/mol  
*T<sub>B</sub>* – boiling point of compound, K

**TABLE C-19 Solubility of Nitrogen Compounds in Water**

No.	Formula	Substance	T <sub>B</sub>	Code	T	S at T, ppm (wt)	S at T, ppm (mol)
1	CH <sub>3</sub> N	Methylamine	266.82	1	25	High	High
2	C <sub>2</sub> H <sub>7</sub> N	Ethylamine	289.73	1	20	High	High
3	C <sub>2</sub> H <sub>7</sub> N	Dimethylamine	280.03	2	25	High	High
4	C <sub>3</sub> H <sub>9</sub> N	<i>n</i> -Propylamine	321.65	1	100	1.0000E+06	1.0000E+06
5	C <sub>3</sub> H <sub>9</sub> N	<i>iso</i> -Propylamine	305.55	1	100	1.0000E+06	1.0000E+06
6	C <sub>3</sub> H <sub>9</sub> N	Methylethylamine	309.15	2	25	High	High
7	C <sub>3</sub> H <sub>9</sub> N	Trimethylamine	276.02	2	25	High	High
8	C <sub>4</sub> H <sub>11</sub> N	<i>n</i> -Butylamine	350.55	1	100	1.0000E+06	1.0000E+06
9	C <sub>4</sub> H <sub>11</sub> N	<i>iso</i> -Butylamine	340.88	1	20	1.0000E+06	1.0000E+06
10	C <sub>4</sub> H <sub>11</sub> N	<i>sec</i> -Butylamine	336.15	2	20	High	High
11	C <sub>4</sub> H <sub>11</sub> N	<i>tert</i> -Butylamine	317.55	2	20	High	High
12	C <sub>4</sub> H <sub>11</sub> N	Diethylamine	328.60	1	38	1.0000E+06	1.0000E+06
13	C <sub>5</sub> H <sub>13</sub> N	<i>n</i> -Pentylamine	377.65	1	25	High	High
14	C <sub>5</sub> H <sub>13</sub> N	1-Methylbutylamine	364.85	2	25	High	High
15	C <sub>5</sub> H <sub>13</sub> N	1-Ethylpropylamine	362.15	2	25	High	High
16	C <sub>5</sub> H <sub>13</sub> N	2-Methylbutylamine	368.65	2	25	High	High
17	C <sub>5</sub> H <sub>13</sub> N	3-Methylbutylamine	370.15	2	25	High	High
18	C <sub>6</sub> H <sub>15</sub> N	<i>n</i> -Hexylamine	404.65	1	25	1.2000E+04	2.1576E+03
19	C <sub>6</sub> H <sub>15</sub> N	1-Methylpentylamine	403.15	2	25	2.6103E+04	4.7490E+03
20	C <sub>6</sub> H <sub>15</sub> N	1-Ethylbutylamine	400.45	2	25	3.0072E+04	5.4893E+03
21	C <sub>7</sub> H <sub>17</sub> N	<i>n</i> -Heptylamine	430.05	2	25	7.1302E+03	1.1216E+03
22	C <sub>7</sub> H <sub>17</sub> N	Methyl- <i>n</i> -hexylamine	415.15	2	25	1.0213E+04	1.6107E+03
23	C <sub>7</sub> H <sub>17</sub> N	Ethyl- <i>n</i> -pentylamine	407.15	2	25	1.5615E+04	2.4740E+03
24	C <sub>7</sub> H <sub>17</sub> N	Dimethyl- <i>n</i> -pentylamine	395.15	2	25	8.6243E+03	1.3583E+03
25	C <sub>7</sub> H <sub>17</sub> N	Dimethyl-2-Pentylamine	382.15	2	25	1.7197E+04	2.7284E+03
26	C <sub>8</sub> H <sub>19</sub> N	<i>n</i> -Octylamine	452.75	1	25	2.0000E+02	2.7882E+01
27	C <sub>8</sub> H <sub>19</sub> N	2-Ethylhexylamine	442.35	1	20	2.5000E+03	3.4921E+02
28	C <sub>8</sub> H <sub>19</sub> N	Di- <i>n</i> -butylamine	432.00	1	25	3.5000E+03	4.8932E+02
29	C <sub>8</sub> H <sub>19</sub> N	Methyl- <i>n</i> -heptylamine	438.15	2	25	3.3545E+03	4.6893E+02
30	C <sub>8</sub> H <sub>19</sub> N	Ethyl- <i>n</i> -hexylamine	431.15	2	25	4.8771E+03	6.8266E+02
31	C <sub>8</sub> H <sub>19</sub> N	Dimethyl- <i>n</i> -hexylamine	418.15	2	25	2.8303E+03	3.9546E+02
32	C <sub>8</sub> H <sub>19</sub> N	Diethyl- <i>n</i> -butylamine	409.15	2	25	4.5801E+03	6.4092E+02
33	C <sub>9</sub> H <sub>21</sub> N	<i>n</i> -Nonylamine	475.35	2	25	7.7899E+02	9.8706E+01
34	C <sub>9</sub> H <sub>21</sub> N	Methyl- <i>n</i> -octylamine	459.85	2	25	1.1549E+03	1.4638E+02
35	C <sub>9</sub> H <sub>21</sub> N	Ethyl- <i>n</i> -heptylamine	453.85	2	25	1.5929E+03	2.0197E+02
36	C <sub>10</sub> H <sub>23</sub> N	<i>n</i> -Decylamine	493.65	2	25	5.5000E+02	6.3021E+01
37	C <sub>10</sub> H <sub>23</sub> N	Di- <i>n</i> -pentylamine	476.15	2	25	5.3283E+02	6.1053E+01
38	C <sub>10</sub> H <sub>23</sub> N	Methyl- <i>n</i> -nonylamine	480.15	2	25	4.2996E+02	4.9261E+01
39	C <sub>11</sub> H <sub>25</sub> N	<i>n</i> -Undecylamine	516.15	2	25	1.0516E+02	1.1059E+01
40	C <sub>11</sub> H <sub>25</sub> N	Methyl- <i>n</i> -decylamine	499.35	2	25	1.6721E+02	1.7585E+01

Code: 1 – experimental data; 2 – estimate  
T<sub>B</sub> – boiling point temperature, K  
S – solubility in water, ppm  
ppm – parts per million  
T – temperature, °C

**TABLE C-20 Henry's Law Constant for Nitrogen Compounds in Water**

Henry's Law Constant, $H$ , at 25°C							
No.	Formula	$H$ at $T$ (atm/mol frac)	$H$ at $T$ (atm/mol/m <sup>3</sup> )	No.	Formula	$H$ at $T$ (atm/mol frac)	$H$ at $T$ (atm/mol/m <sup>3</sup> )
1	CH <sub>5</sub> N	0.49	8.8019E-06	41	C <sub>6</sub> H <sub>15</sub> N	1.64	2.9579E-05
2	C <sub>2</sub> H <sub>7</sub> N	0.66	1.1897E-05	42	C <sub>6</sub> H <sub>15</sub> N	2.26	4.0759E-05
3	C <sub>2</sub> H <sub>9</sub> N	0.88	1.5886E-05	43	C <sub>6</sub> H <sub>15</sub> N	2.39	4.3037E-05
4	C <sub>3</sub> H <sub>9</sub> N	1.08	1.9362E-05	44	C <sub>6</sub> H <sub>15</sub> N	1.83	3.2888E-05
5	C <sub>3</sub> H <sub>9</sub> N	0.85	1.5374E-05	45	C <sub>6</sub> H <sub>15</sub> N	1.53	2.7546E-05
6	C <sub>3</sub> H <sub>9</sub> N	1.44	2.5904E-05	46	C <sub>6</sub> H <sub>15</sub> N	1.66	2.9888E-05
7	C <sub>3</sub> H <sub>9</sub> N	2.62	4.7117E-05	47	C <sub>6</sub> H <sub>15</sub> N	1.50	2.7003E-05
8	C <sub>4</sub> H <sub>11</sub> N	1.51	2.7141E-05	48	C <sub>6</sub> H <sub>15</sub> N	1.44	2.5967E-05
9	C <sub>4</sub> H <sub>11</sub> N	1.37	2.4672E-05	49	C <sub>6</sub> H <sub>15</sub> N	2.37	4.2731E-05
10	C <sub>4</sub> H <sub>11</sub> N	1.35	2.4361E-05	50	C <sub>6</sub> H <sub>15</sub> N	5.62	1.0111E-04
11	C <sub>4</sub> H <sub>11</sub> N	1.04	1.8769E-05	51	C <sub>6</sub> H <sub>15</sub> N	2.87	5.1655E-05
12	C <sub>4</sub> H <sub>11</sub> N	1.87	3.3586E-05	52	C <sub>6</sub> H <sub>15</sub> N	7.23	1.3021E-04
13	C <sub>4</sub> H <sub>11</sub> N	2.20	3.9552E-05	53	C <sub>6</sub> H <sub>15</sub> N	6.34	1.1417E-04
14	C <sub>4</sub> H <sub>11</sub> N	1.90	3.4228E-05	54	C <sub>6</sub> H <sub>15</sub> N	5.81	1.0455E-04
15	C <sub>4</sub> H <sub>11</sub> N	4.90	8.8180E-05	55	C <sub>6</sub> H <sub>15</sub> N	5.48	9.8659E-05
16	C <sub>5</sub> H <sub>13</sub> N	2.12	3.8121E-05	56	C <sub>6</sub> H <sub>15</sub> N	5.33	9.5861E-05
17	C <sub>5</sub> H <sub>13</sub> N	1.60	2.8776E-05	57	C <sub>6</sub> H <sub>15</sub> N	5.18	9.3157E-05
18	C <sub>5</sub> H <sub>13</sub> N	1.51	2.7255E-05	58	C <sub>6</sub> H <sub>15</sub> N	5.03	9.0544E-05
19	C <sub>5</sub> H <sub>13</sub> N	1.73	3.1144E-05	59	C <sub>6</sub> H <sub>15</sub> N	4.89	8.8017E-05
20	C <sub>5</sub> H <sub>13</sub> N	1.79	3.2159E-05	60	C <sub>6</sub> H <sub>15</sub> N	5.48	9.8659E-05
21	C <sub>5</sub> H <sub>13</sub> N	1.21	2.1851E-05	61	C <sub>6</sub> H <sub>15</sub> N	4.14	7.4531E-05
22	C <sub>5</sub> H <sub>13</sub> N	1.38	2.4756E-05	62	C <sub>6</sub> H <sub>15</sub> N	4.14	7.4531E-05
23	C <sub>5</sub> H <sub>13</sub> N	1.21	2.1851E-05	63	C <sub>6</sub> H <sub>15</sub> N	2.50	4.5008E-05
24	C <sub>5</sub> H <sub>13</sub> N	4.34	7.8193E-05	64	C <sub>6</sub> H <sub>15</sub> N	4.17	7.5143E-05
25	C <sub>5</sub> H <sub>13</sub> N	3.14	5.6540E-05	65	C <sub>6</sub> H <sub>15</sub> N	11.16	2.0095E-04
26	C <sub>5</sub> H <sub>13</sub> N	3.03	5.4504E-05	66	C <sub>6</sub> H <sub>15</sub> N	12.64	2.2744E-04
27	C <sub>5</sub> H <sub>13</sub> N	2.75	4.9531E-05	67	C <sub>6</sub> H <sub>15</sub> N	12.64	2.2744E-04
28	C <sub>5</sub> H <sub>13</sub> N	3.28	5.9117E-05	68	C <sub>6</sub> H <sub>15</sub> N	9.33	1.6793E-04
29	C <sub>5</sub> H <sub>13</sub> N	2.96	5.3200E-05	69	C <sub>6</sub> H <sub>15</sub> N	11.47	2.0645E-04
30	C <sub>5</sub> H <sub>13</sub> N	6.87	1.2367E-04	70	C <sub>6</sub> H <sub>15</sub> N	11.89	2.1401E-04
31	C <sub>5</sub> H <sub>13</sub> N	6.87	1.2368E-04	71	C <sub>6</sub> H <sub>15</sub> N	11.89	2.1401E-04
32	C <sub>5</sub> H <sub>13</sub> N	6.87	1.2368E-04	72	C <sub>7</sub> H <sub>17</sub> N	3.21	5.7839E-05
33	C <sub>6</sub> H <sub>15</sub> N	2.70	4.8577E-05	73	C <sub>7</sub> H <sub>17</sub> N	5.51	9.9229E-05
34	C <sub>6</sub> H <sub>15</sub> N	2.68	4.8233E-05	74	C <sub>7</sub> H <sub>17</sub> N	4.50	8.1079E-05
35	C <sub>6</sub> H <sub>15</sub> N	2.51	4.5156E-05	75	C <sub>7</sub> H <sub>17</sub> N	26.88	4.8377E-04
36	C <sub>6</sub> H <sub>15</sub> N	2.48	4.4721E-05	76	C <sub>7</sub> H <sub>17</sub> N	18.64	3.3546E-04
37	C <sub>6</sub> H <sub>15</sub> N	2.40	4.3140E-05	77	C <sub>7</sub> H <sub>17</sub> N	22.63	4.0727E-04
38	C <sub>6</sub> H <sub>15</sub> N	2.31	4.1631E-05	78	C <sub>7</sub> H <sub>17</sub> N	21.39	3.8501E-04
39	C <sub>6</sub> H <sub>15</sub> N	1.50	2.7003E-05	79	C <sub>7</sub> H <sub>17</sub> N	21.09	3.7968E-04
40	C <sub>6</sub> H <sub>15</sub> N	1.56	2.8105E-05	80	C <sub>7</sub> H <sub>17</sub> N	23.95	4.3107E-04

**TABLE C-21 Coefficient of Thermal Expansion of Liquids**

No.	Formula	Substance	$\alpha$	$T_c$	$m$	$T_{\min}$	$T_{\max}$	$B_{\text{liq}} = \alpha (1 - T/T_c)^m$	density <sub>liq</sub> at 25°C
								$B_{\text{liq}}$ at 25°C	
1	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,1-Trichloroethane	6.9960E-04	545.00	-0.7067	242.75	517.75	1.2240E-03	1.330
2	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,2-Trichloroethane	7.0420E-04	602.00	-0.6900	236.50	571.90	1.290E-03	1.435
3	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,1-Dichloroethane	7.2810E-04	523.00	-0.7130	176.19	496.85	1.3290E-03	1.168
4	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,2-Dichloroethane	6.8990E-04	561.00	-0.6896	237.49	532.95	1.1640E-03	1.246
5	C <sub>4</sub> H <sub>6</sub>	1,3-Butadiene	8.8920E-04	425.37	-0.7093	164.25	404.10	2.0930E-03	0.615
6	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	1,4-Dioxane	6.5840E-04	587.00	-0.6953	284.95	557.65	1.0780E-03	1.029
7	C <sub>4</sub> H <sub>10</sub> O	1-Butanol ( <i>n</i> -Butanol)	5.7680E-04	562.93	-0.7543	183.85	534.78	1.0190E-03	0.806
8	C <sub>4</sub> H <sub>8</sub>	1-Butene	8.9970E-04	419.59	-0.7147	87.80	398.61	2.1820E-03	0.588
9	C <sub>10</sub> H <sub>20</sub>	1-Decene	6.2750E-04	617.05	-0.7144	206.89	586.20	1.0060E-03	0.737
10	C <sub>9</sub> H <sub>20</sub>	1-Nonane ( <i>n</i> -Nonane)	6.5440E-04	595.65	-0.7143	219.63	565.87	1.0740E-03	0.715
11	C <sub>8</sub> H <sub>16</sub>	1-Octene	6.8610E-04	566.60	-0.7143	171.45	538.27	1.1700E-03	0.711
12	C <sub>3</sub> H <sub>8</sub> O	1-Propanol ( <i>n</i> -Propanol)	6.0500E-04	536.71	-0.7506	146.95	509.87	1.1120E-03	0.802
13	C <sub>2</sub> H <sub>4</sub> O	Acetaldehyde	8.1110E-04	461.00	-0.7224	150.15	437.95	1.7200E-03	0.774
14	C <sub>2</sub> H <sub>4</sub> O <sub>2</sub>	Acetic acid	5.9380E-04	592.71	-0.7316	289.81	563.07	9.9030E-04	1.043
15	C <sub>4</sub> H <sub>6</sub> O <sub>3</sub>	Acetic anhydride	6.7520E-04	569.15	-0.7301	200.15	540.69	1.1610E-03	1.077
16	C <sub>3</sub> H <sub>6</sub> O	Acetone	7.9810E-04	508.20	-0.7010	178.45	482.79	1.4830E-03	0.786
17	C <sub>3</sub> H <sub>4</sub> O <sub>2</sub>	Acrylic acid	6.7590E-04	615.00	-0.6930	286.65	584.25	1.0700E-03	1.046
18	NH <sub>3</sub>	Ammonia	9.7330E-04	405.65	-0.7113	195.41	385.37	2.5030E-03	0.602
19	C <sub>6</sub> H <sub>7</sub> N	Aniline	5.6660E-04	699.00	-0.7143	267.13	664.05	8.4290E-04	1.018
20	C <sub>6</sub> H <sub>6</sub>	Benzene	6.6060E-04	562.16	-0.7182	278.68	534.05	1.1370E-03	0.873
21	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Butyric acid	5.9730E-04	628.00	-0.7200	267.95	596.60	9.4950E-04	0.953
22	CS <sub>2</sub>	Carbon disulfide	7.2850E-04	552.00	-0.6774	161.58	524.40	1.2330E-03	1.256
23	CO <sub>2</sub>	Carbon dioxide	1.2800E-03	304.19	-0.7097	216.58	288.98	—	0.713
24	CO	Carbon monoxide	2.8090E-03	132.92	-0.7095	68.15	126.27	—	—
25	CCl <sub>4</sub>	Carbon tetrachloride	6.6990E-04	556.35	-0.7100	250.33	528.53	1.1550E-03	1.583
26	Cl <sub>2</sub>	Chlorine	8.9690E-04	417.15	-0.7117	172.12	396.29	2.1900E-03	1.398
27	CHCl <sub>3</sub>	Chloroform	7.3760E-04	536.40	-0.7123	209.63	509.58	1.3150E-03	1.480
28	C <sub>6</sub> H <sub>12</sub>	Cyclohexane	6.6670E-04	553.54	-0.7149	279.69	525.86	1.1590E-03	0.773
29	C <sub>6</sub> H <sub>12</sub> O	Cyclohexanol	6.4580E-04	625.15	-0.7143	296.60	593.89	1.0260E-03	0.774
30	C <sub>3</sub> H <sub>6</sub>	Cyclopropane	9.2960E-04	397.91	-0.7143	145.73	378.01	2.4970E-03	0.619
31	CH <sub>2</sub> Cl <sub>2</sub>	Dichloromethane	7.7360E-04	510.00	-0.7098	178.01	484.50	1.4430E-03	1.318
32	C <sub>4</sub> H <sub>10</sub> O	Diethyl ether	8.0960E-04	466.70	-0.7064	156.85	443.37	1.6620E-03	0.708
33	C <sub>5</sub> H <sub>10</sub> O	Diethyl ketone	6.9030E-04	560.95	-0.7264	234.18	532.90	1.1970E-03	0.810
34	C <sub>2</sub> H <sub>7</sub> N	Dimethylamine	7.4650E-04	437.65	-0.7520	180.96	415.77	1.7640E-03	0.650
35	C <sub>2</sub> H <sub>6</sub>	Ethane	1.2030E-03	305.42	-0.7167	90.35	290.15	—	0.315
36	C <sub>2</sub> H <sub>6</sub> O	Ethanol	6.1070E-04	516.25	-0.7633	159.05	490.44	1.1790E-03	0.787
37	C <sub>4</sub> H <sub>8</sub> O <sub>2</sub>	Ethyl acetate	7.1860E-04	523.30	-0.7220	189.60	497.14	1.3210E-03	0.894
38	C <sub>2</sub> H <sub>5</sub> Cl	Ethyl chloride	6.4960E-04	460.35	-0.7686	136.75	437.33	1.4480E-05	0.890
39	C <sub>8</sub> H <sub>10</sub>	Ethylbenzene	6.2960E-04	617.17	-0.7079	178.20	586.31	1.0050E-03	0.865
40	C <sub>2</sub> H <sub>4</sub>	Ethylene	1.2860E-03	282.36	-0.7143	104.01	268.24	—	—
41	C <sub>2</sub> H <sub>6</sub> O <sub>2</sub>	Ethyleneglycol	3.6440E-04	645.00	-0.8280	260.15	612.75	6.0910E-04	1.110
42	C <sub>2</sub> H <sub>4</sub> O	Ethylene oxide	8.0920E-04	469.15	-0.7175	161.45	445.69	1.6690E-03	0.862
43	F <sub>2</sub>	Fluorine	2.5210E-03	144.31	-0.7100	53.48	137.00	—	—
44	C <sub>3</sub> H <sub>8</sub> O <sub>3</sub>	Glycerol	2.9630E-04	723.00	-0.8459	291.33	686.85	4.6460E-04	1.257
45	H <sub>2</sub>	Hydrogen	8.7840E-03	33.18	-0.7244	13.95	31.52	—	—
46	HCl	Hydrogen chloride	1.2870E-03	324.65	-0.6813	158.97	308.42	7.0940E-03	0.796
47	CHN	Hydrogen cyanide	1.0390E-03	456.65	-0.7179	259.91	433.82	2.2220E-03	0.680
48	H <sub>2</sub> O <sub>2</sub>	Hydrogen peroxide	5.4650E-04	730.15	-0.7123	272.74	693.64	7.9430E-04	1.443
49	C <sub>4</sub> H <sub>10</sub>	<i>i</i> -Butane ( <i>iso</i> -Butane)	8.6860E-04	408.14	-0.7270	113.54	387.73	2.2530E-03	0.552
50	CH <sub>4</sub>	Methane	1.8090E-03	190.58	-0.7230	90.67	181.05	—	—
51	CH <sub>4</sub> O	Methanol	5.9220E-04	512.58	-0.7669	175.47	486.95	1.1550E-03	0.787

**TABLE C-21—(continued)**

No.	Formula	Substance	$B_{\text{liq}} = a (1 - T/T_c)^m$						
			a	T <sub>c</sub>	m	T <sub>min</sub>	T <sub>max</sub>	B <sub>liq</sub> at 25°C	density <sub>liq</sub> at 25°C
52	CH <sub>3</sub> Br	Methyl bromide	8.0180E-04	467.00	-0.7197	179.55	443.65	1.6670E-03	1.662
53	CH <sub>3</sub> Cl	Methyl chloride	9.2560E-04	416.25	-0.7131	175.45	395.44	2.2730E-03	0.913
54	C <sub>6</sub> H <sub>12</sub> O	Methyl isobutyl ketone	6.7570E-04	571.40	-0.7143	189.15	542.83	1.1450E-03	0.796
55	CH <sub>5</sub> N	Methylamine	8.1550E-04	430.05	-0.7725	179.69	408.55	2.0320E-03	0.655
56	C <sub>8</sub> H <sub>10</sub>	<i>m</i> -Xylene	5.9600E-04	617.05	-0.7276	225.30	586.20	9.6340E-04	0.861
57	C <sub>10</sub> H <sub>8</sub>	Naphthalene	5.0520E-04	748.35	-0.7270	353.43	710.93	—	—
58	C <sub>4</sub> H <sub>10</sub>	<i>n</i> -Butane	8.7570E-04	425.18	-0.7137	134.86	403.92	2.0740E-03	0.573
59	C <sub>10</sub> H <sub>22</sub>	<i>n</i> -Decane	6.3600E-04	618.45	-0.7143	243.49	587.53	1.0180E-03	0.728
60	C <sub>7</sub> H <sub>16</sub>	<i>n</i> -Heptane	6.9550E-04	540.26	-0.7209	182.57	513.25	1.2410E-03	0.682
61	C <sub>6</sub> H <sub>14</sub>	<i>n</i> -Hexane	7.2780E-04	507.43	-0.7219	177.84	482.06	1.3790E-03	0.656
62	C <sub>6</sub> H <sub>5</sub> NO <sub>2</sub>	Nitrobenzene	5.5520E-04	719.00	-0.7143	278.91	683.05	8.1390E-04	1.199
63	N <sub>2</sub>	Nitrogen	2.9130E-03	126.10	-0.7075	63.15	119.80	—	—
64	C <sub>9</sub> H <sub>20</sub>	<i>n</i> -Nonane	6.5440E-04	595.65	-0.7143	219.63	565.87	1.0740E-03	0.715
65	C <sub>8</sub> H <sub>18</sub>	<i>n</i> -Octane	6.4760E-04	568.83	-0.7306	216.38	540.39	1.1140E-03	0.699
66	C <sub>5</sub> H <sub>12</sub>	<i>n</i> -Pentane	7.8830E-04	469.65	-0.7179	143.42	446.17	1.6250E-03	0.621
67	O <sub>2</sub>	Oxygen	2.3560E-03	154.58	-0.7076	54.35	146.85	—	—
68	C <sub>8</sub> H <sub>10</sub>	<i>o</i> -Xylene	5.8440E-04	630.37	-0.7259	247.98	598.85	9.3020E-04	0.876
69	C <sub>6</sub> H <sub>6</sub> O	Phenol	5.2480E-04	694.25	-0.6788	314.06	659.54	—	—
70	C <sub>8</sub> H <sub>10</sub>	<i>p</i> -Xylene	6.0980E-04	616.26	-0.7210	286.41	585.45	9.8230E-04	0.858
71	C <sub>5</sub> H <sub>5</sub> N	Pyridine	6.9420E-04	619.95	-0.6955	231.53	588.95	1.0950E-03	0.979
72	C <sub>8</sub> H <sub>8</sub>	Styrene	5.8860E-04	648.00	-0.7143	242.54	615.60	9.1420E-04	0.900
73	SO <sub>2</sub>	Sulfur dioxide	9.1740E-04	430.75	-0.7107	197.67	409.21	2.1190E-03	1.366
74	C <sub>7</sub> H <sub>8</sub>	Toluene	6.5930E-04	591.79	-0.7011	178.18	562.20	1.0780E-03	0.865
75	C <sub>4</sub> H <sub>6</sub> O <sub>2</sub>	Vinyl acetate	7.3090E-04	524.00	-0.7173	180.35	497.80	1.3370E-03	0.926
76	H <sub>2</sub> O	Water	5.7160E-04	647.13	-0.7143	273.16	614.77	8.8850E-04	1.027

 $B_{\text{liq}}$  – coefficient of thermal expansion of liquid, 1/Cdensity<sub>liq</sub> – density of liquid, g/cm<sup>3</sup>T<sub>min</sub> – minimum temperature, KT<sub>max</sub> – maximum temperature, K

**TABLE C-22 Adsorption Capacity of Activated Carbon**

No.	Formula	Substance	$\log_{10} Q = A + B [\log_{10} y] + C [\log_{10} y]^2$							
			A	B	C	$y_{\min}$	$y_{\max}$	Q at 10 ppmv	Q at 100 ppmv	Q at 1000 ppmv
1	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,1-Trichloroethane	0.9733	2.8737E-01	-2.2770E-02	10	10000	17.29	28.64	42.70
2	C <sub>2</sub> H <sub>3</sub> Cl <sub>3</sub>	1,1,2-trichloroethane	1.1716	2.7791E-01	-2.7460E-02	10	10000	26.43	41.46	57.31
3	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,1-Dichloroethane	0.5449	3.6091E-01	-2.1920E-02	10	10000	7.65	15.10	26.93
4	C <sub>2</sub> H <sub>4</sub> Cl <sub>2</sub>	1,2-Dichloroethane	0.5534	3.7072E-01	-2.1610E-02	10	10000	7.99	16.16	29.59
5	C <sub>3</sub> H <sub>8</sub> O	1-Propanol ( <i>n</i> -Propanol)	0.3864	4.8033E-01	-4.5050E-02	10	10000	6.63	14.69	26.42
6	C <sub>2</sub> H <sub>4</sub> O	Acetaldehyde	-1.1705	6.2766E-01	-2.4750E-02	10	10000	0.27	0.97	3.09
7	C <sub>2</sub> H <sub>4</sub> O <sub>2</sub>	Acetic acid	-0.0555	6.8410E-01	-6.0710E-02	10	10000	3.70	11.74	28.21
8	C <sub>3</sub> H <sub>6</sub> O	Acetone	-0.1455	4.7497E-01	-2.2860E-02	10	10000	2.03	5.16	11.85
9	C <sub>3</sub> H <sub>4</sub> O <sub>2</sub>	Acrylic acid	0.7555	4.7108E-01	-5.6150E-02	10	5221	14.81	29.72	46.06
10	CS <sub>2</sub>	Carbon disulfide	-0.1890	4.7093E-01	-1.4810E-02	10	10000	1.85	4.94	12.32
11	CO <sub>2</sub>	Carbon dioxide	-3.6522	8.0180E-01	-3.2800E-03	10	10000	1.4E-03	8.7E-03	5.3E-02
12	CO	Carbon monoxide	-5.1878	9.0121E-01	-1.3580E-02	10	10000	5.0E-05	3.6E-04	2.5E-03
13	CCl <sub>4</sub>	Carbon tetrachloride	1.0748	2.8186E-01	-2.2730E-02	10	10000	21.57	35.29	51.98
14	CHCl <sub>3</sub>	Chloroform	0.6710	3.1648E-01	-2.2880E-02	10	10000	10.22	20.07	35.45
15	CH <sub>2</sub> Cl <sub>2</sub>	Dichloromethane	-0.0704	4.9210E-01	-2.2760E-02	10	10000	2.51	6.65	15.89
16	C <sub>2</sub> H <sub>6</sub>	Ethane	-2.4039	6.8107E-01	-1.9250E-02	10	10000	1.8E-02	7.6E-02	0.29
17	C <sub>2</sub> H <sub>6</sub> O	Ethanol	-0.5115	6.7525E-01	-4.4730E-02	10	10000	1.32	4.57	12.93
18	C <sub>2</sub> H <sub>5</sub> Cl	Ethyl chloride	-0.5083	5.0364E-01	-2.1790E-02	10	10000	0.94	2.58	6.40
19	C <sub>2</sub> H <sub>4</sub>	Ethylene	-2.2710	6.1731E-01	-1.4670E-02	10	10000	2.1E-02	8.0E-02	0.28
20	C <sub>2</sub> H <sub>6</sub> O <sub>2</sub>	Ethylene glycol	1.4047	1.8738E-01	-2.6630E-02	10	121	36.77	47.09	-
21	C <sub>2</sub> H <sub>4</sub> O	Ethylene oxide	-2.4238	9.4878E-01	-4.0620E-02	10	10000	0.03	0.20	1.14
22	CH <sub>2</sub> O	Formaldehyde	-2.4852	6.9123E-01	-3.7500E-03	10	10000	1.6E-02	7.6E-02	0.36
23	CH <sub>2</sub> O <sub>2</sub>	Formic acid	-1.7773	1.0950E+00	-6.3540E-02	10	10000	0.18	1.44	8.63
24	CHN	Hydrogen cyanide	-4.3925	1.0895E+00	-7.4000E-03	10	10000	4.9E-04	5.7E-03	6.4E-02
25	CH <sub>4</sub>	Methane	-4.3101	7.7883E-01	-6.2800E-03	10	10000	2.9E-04	1.7E-03	9.3E-03
26	CH <sub>4</sub> O	Methanol	-1.9674	8.2107E-01	-1.3930E-02	10	10000	0.07	0.42	2.35
27	C <sub>3</sub> H <sub>6</sub> O <sub>2</sub>	Methyl acetate	0.1331	4.2849E-01	-2.1880E-02	10	10000	3.47	7.99	16.66
28	CH <sub>3</sub> Br	Methyl bromide	-1.2384	7.8564E-01	-5.5210E-02	10	10000	0.31	1.29	4.19
29	CH <sub>3</sub> Cl	Methyl chloride	-1.9187	6.2053E-03	-5.4900E-03	10	10000	0.05	0.20	0.78
30	CH <sub>3</sub> N	Methylamine	-1.9355	6.4710E-01	-1.0570E-02	10	10000	0.05	0.21	0.81
31	C <sub>3</sub> H <sub>7</sub> NO	<i>N,N</i> -Dimethylformamide	0.9025	3.7875E-01	-4.5230E-02	10	5220	17.22	30.14	42.83
32	CH <sub>3</sub> NO <sub>2</sub>	Nitromethane	-0.3285	7.0602E-01	-5.1110E-02	10	10000	2.12	7.57	21.36
33	C <sub>3</sub> H <sub>8</sub> O	Propane	-0.7946	4.9029E-01	-2.3980E-02	10	10000	0.47	1.23	2.89
34	C <sub>3</sub> H <sub>6</sub> O <sub>2</sub>	Propionic acid	0.7785	4.4570E-01	-5.2090E-02	10	4872	14.86	28.94	44.34
35	C <sub>2</sub> HCl <sub>3</sub>	Trichloroethylene	1.0241	2.9929E-01	-2.5390E-02	10	10000	19.86	33.20	49.38

*Q* – adsorption capacity at equilibrium, g of compound/100 g of carbon*y* – concentration of compound in gas at 25° C and 1 atm, parts per million by volume, ppmv*A, B, and C* – regression coefficients for chemical compound*y<sub>min</sub>* – minimum concentration, ppmv*y<sub>max</sub>* – maximum concentration, ppmv

# Appendix D

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## D-1 Alphabetical Conversion Factors

### A

Acre	$\times 10$	= Square chain (Gunters)
Acre	$\times 160$	= Rods
Acre	$\times 1 \times 10^5$	= Square links (Gunters)
Acre	$\times 0.4047$	= Hectare or square hectometer
Acres	$\times 43,560$	= Square feet
Acres	$\times 4,047$	= Square meters
Acres	$\times 1.562 \times 10^{-3}$	= Square miles
Acres	$\times 4,840$	= Square yards
Acre-feet	$\times 43,560$	= Cubic feet
Acre-feet	$\times 3.259 \times 10^5$	= Gallons
Amperes/square centimeters	$\times 6.452$	= Amperes/square inch
Amperes/square centimeters	$\times 10^4$	= Amperes/square meter
Amperes/square inch	$\times 0.1550$	= Amperes/square centimeters
Amperes/square inches	$\times 1,550$	= Amperes/square meter
Amperes/square meter	$\times 10^{-4}$	= Amperes/square centimeter
Amperes/square meter	$\times 6.452 \times 10^{-4}$	= Amperes/square inch
Ares	$\times 0.02471$	= Acre (USA)
Ares	$\times 119.60$	= Square yards
Ares	$\times 100$	= Square meters
Atmospheres	$\times 14.7$	= Pounds/square inch
Atmospheres	$\times 1.058$	= Tonnes/square foot
Atmospheres	$\times 29.92$	= Inches of mercury at 0°C
Atmospheres	$\times 76$	= Centimeters of mercury
Atmospheres	$\times 33.90$	= Feet of water at 4°C
Atmospheres	$\times 1.0333$	= Kilograms/square centimeter
Atmospheres	$\times 10,332$	= Kilograms/square meter
Atmospheres	$\times 1,013.2$	= Millibar
Atmospheres	$\times 760$	= Millimeters of mercury
Atmospheres	$\times 10.332$	= Meters of water at 4°C

### B

Barrels (USA, dry)	$\times 7,056$	= Cubic inches
Barrels (USA, dry)	$\times 105$	= Quarts (dry)
Barrels (USA, liquid)	$\times 31.5$	= Gallons
Barrels (oil)	$\times 42$	= Gallons (oil)
Barrels (oil)	$\times 0.159$	= Cubic meters
Barrels (oil)	$\times 159$	= Liters
Barrels/day	$\times 6.6245 \times 10^{-3}$	= Cubic meters/hour
Barrels (oil)	$\times 5,6154$	= Cubic feet
Barrels/day	$\times 29.167 \times 10^{-3}$	= Gallons per minute
Bars	$\times 0.9869$	= Atmospheres
Bars	$\times 10^6$	= Dynes/square centimeter
Bars	$\times 1.020 \times 10^4$	= Kilograms/square meter
Bars	$\times 2,089$	= Pounds/square foot
Bars	$\times 14.50$	= Pounds/square inch
Board feet $\times 144$ sq.in.	$\times 1$ in.	= Cubic inches
Board feet	$\times 0.0833$	= Cubic feet
Btu	$\times 1.055 \times 10^{10}$	= Ergs
Btu	$\times 778.3$	= Foot-pounds
Btu	$\times 252$	= Gram-calories
Btu	$\times 3.931 \times 10^{-4}$	= Horsepower-hours
Btu	$\times 1,054.8$	= Joules
Btu	$\times 0.252$	= Kilogram-calories
Btu	$\times 107.5$	= Kilogram-meters
Btu	$\times 2.928 \times 10^{-4}$	= Kilowatt-hours
Btu/hour	$\times 0.2162$	= Foot-pounds/second
Btu/hour	$\times 0.070$	= Gram-calorie/second

Btu/hour	$\times 3.929 \times 10^{-4}$	= Horsepower-hours (British)
Btu/hour	$\times 0.2931$	= Watts
Btu/hour foot °F	$\times 1.4882$	= Kilocalorie/meter hour °C
Btu/hour foot <sup>2</sup>	$\times 2.7125$	= Kilocalorie/meter <sup>2</sup> hour
Btu/pound	$\times 0.5556$	= Kilocalorie/kilogram
Btu/pound °F	$\times 1$	= Kilocalorie/kilogram °C
Btu inches/hour foot <sup>2</sup> °F	$\times 0.12402$	= Kilocalorie/meter hour °C
Btu/minute	$\times 12.96$	= Foot-pounds/second
Btu/minute	$\times 0.02356$	= Horsepower
Btu/minute	$\times 0.01757$	= Kilowatts
Btu/minute	$\times 17.57$	= Watts
Btu/square foot/minute	$\times 0.1221$	= Watts/square inch
Bucket (British-dry)	$\times 1.818 \times 10^4$	= Cubic centimeters
Bushels	$\times 1.2445$	= Cubic feet
Bushels	$\times 2,150.4$	= Cubic inches
Bushels	$\times 0.03524$	= Cubic meters
Bushels	$\times 35.24$	= Liters
Bushels	$\times 4$	= Pecks
Bushels	$\times 64$	= Pints (dry)
Bushels	$\times 32$	= Quarts (dry)

**C**

Calories, gram (mean)	$\times 3.9685 \times 10^{-3}$	= Btu (mean)
Calories/gram	$\times 1.8$	= Btu/pound
Calories/gram-mol-°C	$\times 1.0$	= Btu/pound-mol-°F
Candle/square centimeter	$\times 3.142$	= Lamberts
Candle/square inch	$\times 0.487$	= Lamberts
Centares	$\times 1.0$	= Square meters
Centigrade	$\times 9/5 + 32$	= Fahrenheit
Centiliter	$\times 0.3382$	= Fluid ounce (USA)
Centiliter	$\times 0.6103$	= Cubic inch
Centiliter	$\times 2.705$	= Drams
Centimeters	$\times 3.281 \times 10^{-2}$	= Feet
Centimeters	$\times 0.3937$	= Inches
Centimeters	$\times 1.094 \times 10^{-2}$	= Yards
Centimeters	$\times 393.7$	= mils
Centimeters of mercury	$\times 0.1934$	= Pounds/square inch
Centimeters of mercury	$\times 0.01316$	= Atmospheres
Centimeters of mercury	$\times 0.4461$	= Feet of water
Centimeters of mercury	$\times 136$	= Kilograms/square meter
Centimeters of mercury	$\times 27.85$	= Pounds/square foot
Centimeters of mercury	$\times 0.1934$	= Pounds/square inch
Centimeters/second	$\times 1.1969$	= Feet/minute
Centimeters/second	$\times 0.03281$	= Feet/second
Centimeters/second	$\times 0.036$	= Kilometers/hour
Centimeters/second	$\times 0.1943$	= Knots
Centimeters/second	$\times 0.6$	= Meters/minute
Centimeters/second	$\times 0.02237$	= Miles/hour
Chain	$\times 792$	= Inches
Chain	$\times 20.12$	= Meters
Circumference	$\times 6.283$	= Radians
Cords	$\times 8$	= Cord feet
Cord feet	$\times 16$	= Cubic feet
Coulomb	$\times 2.998 \times 10^9$	= Statcoulombs
Coulombs	$\times 1.036 \times 10^{-5}$	= Faradays
Coulombs/square centimeter	$\times 64.52$	= Coulombs/square inch
Coulombs/square centimeter	$\times 10^4$	= Coulombs/square meter
Coulombs/square inch	$\times 0.155$	= Coulombs/square centimeter
Coulombs/square inch	$\times 1,550$	= Coulombs/square meter
Cubic centimeters	$\times 3.531 \times 10^{-5}$	= Cubic feet
Cubic centimeters	$\times 0.06102$	= Cubic inches
Cubic centimeters	$\times 10^{-6}$	= Cubic meters
Cubic centimeters	$\times 1.308 \times 10^{-6}$	= Cubic yards
Cubic centimeters	$\times 2.624 \times 10^{-4}$	= Gallons (USA liquid)
Cubic centimeters	$\times 0.001$	= Liters
Cubic centimeters	$\times 2.113 \times 10^{-3}$	= Pints (USA liquid)
Cubic centimeters	$\times 1.057 \times 10^{-3}$	= Quarts (USA liquid)
Cubic feet	$\times 0.8036$	= Bushels (dry)
Cubic feet	$\times 28,320$	= Cubic centimeters

Cubic feet	$\times 1,728$	= Cubic inches
Cubic feet	$\times 0.02832$	= Cubic meters
Cubic feet	$\times 0.03704$	= Cubic yards
Cubic feet	$\times 7,48052$	= Gallons (USA liquid)
Cubic feet	$\times 6.232$	= Gallons (imperial)
Cubic feet	$\times 6.428$	= Gallons (USA dry)
Cubic feet	$\times 62.425$	= Pounds (water)
Cubic feet	$\times 28.32$	= Liters
Cubic feet	$\times 59.84$	= Pints (USA liquid)
Cubic feet	$\times 29.92$	= Quarts (USA liquid)
Cubic feet	$\times 0.1781$	= Barrels (oil, USA)
Cubic feet/minute	$\times 472$	= Cubic centimeters/second
Cubic feet/minute	$\times 0.1247$	= Gallons/second
Cubic feet/minute	$\times 0.4720$	= Liters/second
Cubic feet/minute	$\times 62.43$	= Pounds of water/minute
Cubic feet/minute	$\times 1.6989$	= Cubic meters/hour
Cubic feet/minute	$\times 4.719 \times 10^{-4}$	= Cubic meters/second
Cubic feet/second	$\times 0.646317$	= Million gallons/day
Cubic feet/second	$\times 448.831$	= Gallons/minute
Cubic feet/second	$\times 101.94$	= Cubic meters/hour
Cubic inches	$\times 16.39$	= Cubic centimeters
Cubic inches	$\times 5.787 \times 10^{-4}$	= Cubic feet
Cubic inches	$\times 1.639 \times 10^{-5}$	= Cubic meters
Cubic inches	$\times 2.143 \times 10^{-5}$	= cubic yards
Cubic inches	$\times 4.329 \times 10^{-3}$	= Gallons (U.S)
Cubic inches	$\times 0.01639$	= Liters
Cubic inches	$\times 1.061 \times 10^5$	= Mil-feet
Cubic inches	$\times 0.03463$	= Pints (USA liquid)
Cubic inches	$\times 0.01732$	= Quarts (USA liquid)
Cubic meters	$\times 6.290$	= Barrels(USA oil)
Cubic meters	$\times 28.38$	= Bushes(dry)
Cubic meters	$\times 10^6$	= Cubic centimeters
Cubic meters	$\times 35.314$	= Cubic feet
Cubic meters	$\times 61,023$	= Cubic inches
Cubic meters	$\times 1.308$	= Cubic yards
Cubic meters	$\times 264.17$	= Gallons (USA liquid)
Cubic meters	$\times 220$	= Gallons (imperial)
Cubic meters	$\times 1,000$	= Liters
Cubic meters	$\times 2,113$	= Prints (USA liquid)
Cubic meters	$\times 1.057$	= Quarts (USA liquid)
Cubic meters/hour	$\times 9.810 \times 10^{-3}$	= Cubic feet/second
Cubic meters/hour	$\times 0.5886$	= Cubic feet/minute
Cubic meters/hour	$\times 4.4033$	= Gallons/minute (USA)
Cubic meters/hour	$\times 150.95$	= Barrels/day
Cubic meters/hour	$\times 3.6651$	= Imperial gallons/minute
Cubic meters/hour	$\times 35.31$	= Cubic feet/hour
Cubic meters/hour	$\times 277.8$	= Cubic centimeters/second
Cubic yards	$\times 7,646 \times 10^5$	= Cubic centimeters
Cubic yards	$\times 27$	= Cubic feet
Cubic yards	$\times 46,656$	= Cubic inches
Cubic yards	$\times 0.7646$	= Cubic meters
Cubic yards	$\times 202$	= Gallons (USA liquid)
Cubic yards	$\times 764.6$	= Liters
Cubic yards	$\times 1,615.9$	= Pints (USA liquid)
Cubic yards	$\times 807.9$	= Quarts (USA liquid)
Cubic yards/minute	$\times 0.45$	= Cubic feet/second
Cubic yards/minute	$\times 3.367$	= Gallons/second
Cubic yards/minute	$\times 12.74$	= Liters/second
Cubit (from The Bible)	$\times 21.8$	= Inch
Cup	$\times 0.5$	= Pint
Cup	$\times 16$	= Tablespoon

**D**

Dalton	$\times 1.650 \times 10^{-24}$	= Gram
Days	$\times 1,440$	= Minutes
Days	$\times 86,400$	= Seconds
Decigrams	$\times 0.1$	= Grams
Deciliters	$\times 0.1$	= Liters
Decimeters	$\times 0.1$	= Meters

Degrees (Angle)	$\times 60$	= Minutes
Degrees (Angle)	$\times 3600$	= Seconds
Degrees (Angle)	$\times 0.01745$	= Radians
Degrees (Angle)	$\times 0.01111$	= Quadrants
Degrees/second	$\times 0.1667$	= Revolutions/minute
Degrees/second	$\times 2.778 \times 10^{-3}$	= Revolutions/second
Dekagrams	$\times 10$	= Grams
Dekaliters	$\times 10$	= Liters
Dekameters	$\times 10$	= Meters
Drams (apothecaries or troy)	$\times 0.1371429$	= Ounces (avoirdupois)
Drams (apothecaries or troy)	$\times 0.125$	= Ounces (troy)
Drams	$\times 27.34375$	= Grains
Drams	$\times 1.771845$	= Grams
Drams	$\times 0.0625$	= Ounces
Dyne/centimeter	$\times 0.01$	= Erg/square millimeters
Dyne/square centimeters	$\times 9.869 \times 10^{-7}$	= Atmospheres
Dyne/square centimeters	$\times 2.953 \times 10^{-5}$	= Inch of mercury at 0°C
Dyne/square centimeters	$\times 4.015 \times 10^{-4}$	= Inch of water at 4°C
Dynes/square centimeters	$\times 10^{-6}$	= Bars
Dynes	$\times 1.020 \times 10^{-3}$	= Grams
Dynes	$\times 10^{-7}$	= Joules/centimeters
Dynes	$\times 10^{-5}$	= Joules/meters (newtons)
Dynes	$\times 1.020 \times 10^{-6}$	= Kilograms
Dynes	$\times 7.233 \times 10^{-5}$	= Poundsals
Dynes	$\times 2.248 \times 10^{-6}$	= Pounds

**E**

Ell	$\times 114.30$	= Centimeters
Ell	$\times 45$	= Inches
Em, pica	$\times 0.167$	= Inch
Em, pica	$\times 0.4233$	= Centimeter
Erg/second	$\times 1$	= Dyne-centimeter/second
Erg/second	$\times 1.0 \times 10^{-7}$	= Watt
Erg	$\times 9.480 \times 10^{-11}$	= Btu
Erg	$\times 1$	= Dyne-centimeter
Erg	$\times 7.367 \times 10^{-8}$	= Foot-pounds
Erg	$\times 1.0 \times 10^{-7}$	= Joules
Expansion coefficient, °F	$\times 1.8$	= Expansion coefficient, °C

**F**

Fahrenheit minus 32°F	$\times 0.5556$	= Centigrade
Famm	$\times 5.8455$	= Foot, USA
Famm	$\times 1.7814$	= Meter
Faradays	$\times 26.8$	= Ampere-hour
Fathom, British	$\times 6.08$	= Feet
Fathom, British	$\times 1.8532$	= Meter
Fathom, USA	$\times 6$	= Feet
Fathom, USA	$\times 1.8288$	= Meter
Fathom, USA	$\times 2$	= Yard
Feet	$\times 30.48$	= Centimeters
Feet	$\times 1.645 \times 10^{-4}$	= Miles (nautical)
Feet, USA	$\times 0.3048$	= Meters
Feet, USA	$\times 0.3333$	= Yards
Feet, USA	$\times 0.18939 \times 10^{-3}$	= Miles, USA statute
Feet, USA	$\times 12$	= Inches
Feet, USA	$\times 1.2 \times 10^4$	= Mils
Feet, USA	$\times 0.0606$	= Rod
Feet of water	$\times 0.0295$	= Atmospheres
Feet of water	$\times 0.8826$	= Inches of mercury
Feet of water	$\times 0.03048$	= Kilograms/square centimeters
Feet of water	$\times 304.8$	= Kilograms/square meter
Feet of water	$\times 62.43$	= Pounds/square foot
Feet of water	$\times 0.4335$	= Pounds/square inch
Feet/hour	$\times 0.01666$	= Feet/minute
Feet/hour	$\times 0.2777 \times 10^{-3}$	= Feet/second

Feet/hour	$\times 0.1894 \times 10^{-3}$	= Miles/hour
Feet/minute	$\times 0.5080$	= Centimeter/second
Feet/minute	$\times 0.01666$	= Feet/second
Feet/minute	$\times 0.18288$	= Kilometer/hour
Feet/minute	$\times 0.009868$	= Knot
Feet/minute	$\times 0.3048$	= Meter/minute
Feet/minute	$\times 0.00508$	= Meter/second
Feet/minute	$\times 0.01136$	= Mile/hour
Feet/minute	$\times 0.1894 \times 10^{-3}$	= Mile/minute
Feet/second	$\times 30.48$	= Centimeters/second
Feet/second	$\times 1.097$	= Kilometers/hour
Feet/second	$\times 0.5921$	= Knots
Feet/second	$\times 18.29$	= Meters/minute
Feet/second	$\times 0.681818$	= Miles/hour
Feet/second	$\times 0.0113636$	= Miles/minute
Feet/second	$\times 3600$	= Feet/hour
Feet/second	$\times 60$	= Feet/minute
Feet/second/second	$\times 30.48$	= Centimeters/second/second
Feet/second/second	$\times 1.097$	= Kilometers/hour/second
Feet/second/second	$\times 0.3048$	= Meters/second/second
Feet/second/second	$\times 0.6818$	= Miles/hour/second
Feet/100 feet	$\times 1$	= Percent grade
Firkin	$\times 9$	= Gallon, liquid, USA
Firkin	$\times 34.06798$	= Liter
Foot-candle	$\times 10.764$	= Lumen/square meter
Foot-candle	$\times 1$	= Lumen/square foot
Foot-candle	$\times 10.764$	= Lux
Foot-candle	$\times 1.076$	= Milliphot
Foot-candle	$\times 0.001076$	= Phot
Foot-candle	$\times$ distance in feet <sup>2</sup>	= Candlepower
Foot-Lambert	$\times 0.3425 \times 10^{-3}$	= Candle/square centimeters
Foot-Lambert	$\times 0.3183$	= Candle/square foot
Foot-Lambert	$\times 0.00221$	= Candle/square inch
Foot-Lambert	$\times 0.001076$	= Lambert
Foot-Lambert	$\times$ square foot Area	= Lumen
Foot-Lambert	$\times 1.076$	= Millilambert
Foot-Lambert	$\times 0.342 \times 10^{-3}$	= Stilb
Foot-pound	$\times 1.2853 \times 10^{-3}$	= Btu
Foot-pound	$\times 1.356 \times 10^7$	= Ergs
Foot-pound	$\times 0.32389$	= Gram-calorie
Foot-pound	$\times 5.0505 \times 10^{-7}$	= Horsepower-hours, USA
Foot-pound	$\times 5.12 \times 10^{-7}$	= Horsepower-hours, metric
Foot-pound	$\times 12$	= Inch-pound
Foot-pound	$\times 1.35582$	= Joule absolute
Foot-pound	$\times 1.3554$	= Joule international
Foot-pound	$\times 3.238 \times 10^{-4}$	= Kilogram-calories
Foot-pound	$\times 0.1383$	= Kilogram-meters
Foot-pound	$\times 3.766 \times 10^{-7}$	= Kilowatt-hours
Foot-pound	$\times 0.001356$	= Kilowatt-second
Foot-pound	$\times 0.01338$	= Liter-atmosphere
Foot-pound	$\times 0.3766 \times 10^{-3}$	= Watt-hour
Foot-pound	$\times 1.356$	= Watt-second
Foot-pound/minute	$\times 0.077118$	= Btu/hour
Foot-pound/minute	$\times 1.286 \times 10^{-3}$	= Btu/minute
Foot-pound/minute	$\times 2.259 \times 10^5$	= Erg/second
Foot-pound/minute	$\times 0.01666$	= Foot-pound/second
Foot-pound/minute	$\times 3.066 \times 10^{-5}$	= Horsepower, metric
Foot-pound/minute	$\times 3.0303 \times 10^{-5}$	= Horsepower, USA
Foot-pound/minute	$\times 2.2597 \times 10^{-5}$	= Kilowatt
Foot-pound/minute	$\times 0.022597$	= Watt
Foot-pound/second	$\times 0.0771$	= Btu/minute
Foot-pound/second	$\times 4.6263$	= Btu/hour
Foot-pound/second	$\times 1.843 \times 10^{-3}$	= Horsepower, metric
Foot-pound/second	$\times 1.818 \times 10^{-3}$	= Horsepower, USA
Foot-pound/second	$\times 1.356$	= Joule
Foot-pound/second	$\times 1.3558 \times 10^{-3}$	= Kilowatts
Foot-pound/second	$\times 1.3558$	= Watt
Fot	$\times 0.974$	= Foot, USA
Fot	$\times 100$	= Lines
Fot	$\times 0.2969$	= Meter

Fot	$\times 10$	= Turn
Foute	$\times 1$	= Foot, USA
Furlong	$\times 6.6$	= Chain, engineer
Furlong	$\times 10$	= Chain, Gunter
Furlong	$\times 660$	= Feet
Furlong	$\times 201.168$	= Meters
Furlong	$\times 0.125$	= Mile, statue, USA
Furlong	$\times 220$	= Yards
Furlong	$\times 40$	= Rods
Fuss	$\times 0.9842$	= Foot, USA
Fuss	$\times 0.300$	= Meter

**G**

Gallon, British, Imperial Liquid	$\times 0.125$	= Bushel, dry, British
Gallon, British, Imperial Liquid	$\times 4546$	= Cubic centimeter
Gallon, British, Imperial Liquid	$\times 0.16046$	= Cubic foot
Gallon, British, Imperial Liquid	$\times 0.0045$	= Cubic meter
Gallon, British, Imperial Liquid	$\times 1.032$	= Gallon, dry, USA
Gallon, British, Imperial Liquid	$\times 1.20095$	= Gallon, liquid, USA
Gallon, British, Imperial Liquid	$\times 4.54596$	= Kilogram
Gallon, British, Imperial Liquid	$\times 10$	= Pound, water, 62° F
Gallon, dry, USA	$\times 0.125$	= Bushel, USA
Gallon, dry, USA	$\times 4404.92$	= Cubic centimeter
Gallon, dry, USA	$\times 0.155555$	= Cubic foot
Gallon, dry, USA	$\times 268.803$	= Cubic inch
Gallon, dry, USA	$\times 1.16365$	= Gallon, liquid, USA
Gallon, dry, USA	$\times 4.4049$	= Liter
Gallon, dry, USA	$\times 0.05$	= Peck
Gallon, dry, USA	$\times 8$	= Pint
Gallon, dry, USA	$\times 4.6546$	= Quart, liquid, USA
Gallon, liquid, USA	$\times 0.0238$	= Barrel, oil
Gallon, liquid, USA	$\times 3785.434$	= Cubic centimeter
Gallon, liquid, USA	$\times 3.785434$	= Cubic decimeter
Gallon, liquid, USA	$\times 0.13368$	= Cubic foot
Gallon, liquid, USA	$\times 231$	= Cubic inch, water, 62° F
Gallon, liquid, USA	$\times 3.7854 \times 10^{-3}$	= Cubic meter
Gallon, liquid, USA	$\times 4.951 \times 10^{-3}$	= Cubic yard
Gallon, liquid, USA	$\times 0.859365$	= Gallon, dry, USA
Gallon, liquid, USA	$\times 0.832673$	= Gallon, liquid, British
Gallon, liquid, USA	$\times 3.7853$	= Liter
Gallon, liquid, USA	$\times 8$	= Pint, liquid, USA
Gallon, liquid, USA	$\times 4$	= Quart, liquid, USA
Gallon, liquid, USA	$\times 8.3453$	= Pounds, water
Gallon/hour, USA	$\times 0.1337$	= Cubic feet/hour
Gallon/hour, USA	$\times 2.228 \times 10^{-3}$	= Cubic feet/minute
Gallon/hour, USA	$\times 0.01666$	= Gallons/minute
Gallon/hour, USA	$\times 2.777 \times 10^{-4}$	= Gallons/second
Gallon/minute, USA	$\times 34.2857$	= Barrels/day, oil
Gallon/minute, USA	$\times 1.42857$	= Barrels/hour, oil
Gallon/minute, USA	$\times 0.023809$	= Barrels/minute, oil
Gallon/minute, USA	$\times 192.49999$	= Cubic feet/day
Gallon/minute, USA	$\times 8.021$	= Cubic feet/hour
Gallon/minute, USA	$\times 0.13368$	= Cubic feet/minute
Gallon/minute, USA	$\times 2.228 \times 10^{-3}$	= Cubic feet/second
Gallon/minute, USA	$\times 0.2271$	= Cubic meters/hour
Gallon/minute, USA	$\times 1440$	= Gallons/day
Gallon/minute, USA	$\times 60$	= Gallons/hour
Gallon/minute, USA	$\times 0.01666$	= Gallons/second
Gallon/minute, USA	$\times 5.35565$	= Tonnes, long, water, 62° F/day
Gallon/minute, USA	$\times 5.99839$	= Tonnes, short, water, 62° F/day
Gallon/minute, USA	$\times 0.06308$	= Liters/second
Gallon/second, USA	$\times 481$	= Cubic feet/hour
Gallon/second, USA	$\times 8.02$	= Cubic feet/minute
Gallon/second, USA	$\times 0.1337$	= Cubic feet/second
Gallon/second, USA	$\times 60$	= Gallons/minute, USA
Gills, British	$\times 142.07$	= Cubic centimeter
Gills, British	$\times 0.1183$	= Liters
Gills, British	$\times 0.25$	= Pints, liquid
Grade	$\times 0.0025$	= Circle

Grade	$\times 9,000$	= Degree
Grade	$\times 54$	= Minute
Grade	$\times 0.01571$	= Radian
Grain	$\times 0.01666$	= Dram, apothecary
Grain	$\times 0.03657$	= Dram, avoirdupois
Grain (troy)	$\times 1$	= Grain (avdp)
Grain (troy)	$\times 0.0648$	= Grams
Grain (troy)	$\times 2.0833 \times 10^{-3}$	= Ounces (troy)
Grain (troy)	$\times 2.286 \times 10^{-3}$	= Ounces (avdp)
Grain/U.S. gallon	$\times 17.118$	= Parts/million
Grain/U.S. gallon	$\times 142.86$	= Pounds/million gallon
Grain/Imperial gallon	$\times 14.286$	= Parts/million
Gram	$\times 5$	= Carat
Gram	$\times 3.858$	= Carat, metric
Gram	$\times 100$	= Centigram
Gram	$\times 0.2572$	= Dram, apothecary
Gram	$\times 0.56438$	= Dram, avoirdupois
Gram	$\times 980.665$	= Dyne
Gram	$\times 15.4324$	= Grain
Gram	$\times 9.807 \times 10^{-5}$	= Joules/centimeters
Gram	$\times 9.807 \times 10^{-3}$	= Joules/meter (newtons)
Gram	$\times 0.001$	= Kilograms
Gram	$\times 1000$	= Milligrams
Gram	$\times 0.03527$	= Ounces, avoirdupois
Gram	$\times 0.03215$	= Ounces, troy
Gram	$\times 0.07093$	= Poundals
Gram	$\times 2.205 \times 10^{-3}$	= Pounds
Grams	$\times 1/MW$	= Gram-mols
Grams/centimeter	$\times 5.6 \times 10^{-3}$	= Pounds/inch
Grams/cubic centimeter	$\times 62.43$	= Pounds/cubic foot
Grams/cubic centimeter	$\times 0.03613$	= Pounds/cubic foot
Grams/liter	$\times 58.417$	= Grains/gallon, USA
Grams/liter	$\times 8.345$	= Pounds/1,000 gallons
Grams/liter	$\times 0.062427$	= Pounds/cubic foot
Grams/liter	$\times 1,000$	= Parts/million
Grams/square centimeter	$\times 2.0481$	= Pounds/square foot
Gram-calories	$\times 3.968 \times 10^{-3}$	= Btu
Gram-calories	$\times 4.1868 \times 10^7$	= Ergs
Gram-calories	$\times 3.088$	= Foot-pounds
Gram-calories	$\times 1.55856 \times 10^{-6}$	= Horsepower/hours
Gram-calories	$\times 1.163 \times 10^{-6}$	= Kilowatt-hours
Gram-calories	$\times 1.163 \times 10^{-3}$	= Watt-hours
Gram/calories/second	$\times 14.286$	= Btu/hour
Gram-centimeters	$\times 9.29658 \times 10^{-8}$	= Btu
Gram-centimeters	$\times 980.7$	= Ergs
Gram-centimeters	$\times 9.807 \times 10^{-5}$	= Joules
Gram-mol	$\times 22.414$	= Liters at 0° C and 1 atm
Gross	$\times 12$	= Dozen
Gross, great	$\times 144$	= Dozen
Gross, great	$\times 12$	= Gross

**H**

Hand	$\times 10.16$	= Centimeter
Hand	$\times 4$	= Inch
Hand	$\times 48$	= Foot
Hand	$\times 1,016$	= Meter
Head, feet elevation, water	$\times 0.433$	= Pounds/square inch
Hectare	$\times 2.471$	= Acre
Hectare	$\times 100$	= Are
Hectare	$\times 1.07639 \times 10^5$	= Square feet
Hectare	$\times 0.01$	= Square kilometer
Hectare	$\times 10,000$	= Square meters
Hectare	$\times 3.861 \times 10^{-3}$	= Square miles
Hectare	$\times 11,960$	= Square yard
Hectogram	$\times 100$	= Gram
Hectoliter	$\times 3.532$	= Cubic feet
Hectoliter	$\times 0.1$	= Cubic meter
Hectoliter	$\times 0.1308$	= Cubic yard

Hectoliter	$\times 26.42$	= Gallon, USA
Hectoliter	$\times 100$	= Liter
Hectometer	$\times 328.089$	= Feet
Hectometer	$\times 100$	= Meter
Hectometer	$\times 0.06214$	= Mile, statute, USA
Hectometer	$\times 109.36$	= Yard
Hectowatts	$\times 100$	= Watts
Henries	$\times 1,000$	= Millihenries
Hogsheads, British	$\times 10.114$	= Cubic feet
Hogsheads, USA	$\times 8.42184$	= Cubic feet
Hogsheads, USA	$\times 63$	= Gallons, USA
Hogsheads, USA	$\times 238.476$	= Liter
Hogsheads, USA	$\times 504$	= Pint
Hogsheads, USA	$\times 252$	= Quart
Horsepower, USA	$\times 42.44$	= Btu/minute
Horsepower, USA	$\times 33,000$	= Foot-pounds/minute
Horsepower, USA	$\times 550$	= Foot-pounds/second
Horsepower, USA	$\times 0.7457$	= Kilowatts
Horsepower, USA	$\times 1.014$	= Horsepower (metric)
Horsepower, boiler	$\times 33,479$	= Btu/hour
Horsepower, boiler	$\times 34.5$	= Pounds water/hour
Horsepower, boiler	$\times 9,803$	= Kilowatts
Horsepower, electric	$\times 0.7072$	= Btu/second
Horsepower, electric	$\times 746$	= Joule/second
Horsepower, electric	$\times 0.746$	= Kilowatts
Horsepower, electric	$\times 746$	= Watts
Horsepower (metric)	$\times 0.98632$	= Horsepower (USA)
Horsepower (metric)	$\times 542.5$	= Foot-pounds/second
Horsepower, hours, USA	$\times 2,547$	= Btu
Horsepower, hours, USA	$\times 2.6845 \times 10^{13}$	= Ergs
Horsepower, hours, USA	$\times 1.98 \times 10^6$	= Foot-pounds
Horsepower, hours, USA	$\times 641,190$	= Gram-calories
Horsepower, hours, USA	$\times 1.01387$	= Horsepower-hour, metric
Horsepower, hours, USA	$\times 2,376 \times 10^4$	= Inch-pounds
Horsepower, hours, USA	$\times 26.8453 \times 10^5$	= Joule
Horsepower, hours, USA	$\times 0.7457$	= Kilowatt-hour
Horsepower-hours, metric	$\times 2509.83$	= Btu
Horsepower-hours, metric	$\times 1.9529 \times 10^6$	= Foot-pounds
Horsepower-hours, metric	$\times 0.98632$	= Horsepower-hour, USA
Horsepower-hours, metric	$\times 632,467$	= Gram-calories
Horsepower-hours, metric	$\times 26.4761$	= Joule
Horsepower-hours, metric	$\times 0.73545$	= Kilowatt-hour
Hours	$\times 0.0417$	= Day
Hours	$\times 60$	= Minute
Hours	$\times 0.00137$	= Month
Hours	$\times 0.1142 \times 10^{-3}$	= Year
Hours	$\times 5.952 \times 10^{-3}$	= Week
Hundredweight (long)	$\times 112$	= Pounds
Hundredweight (long)	$\times 0.05$	= Tonnes, long
Hundredweight (short)	$\times 1.8$	= Cubic foot
Hundredweight (short)	$\times 45.36$	= Kilograms
Hundredweight (short)	$\times 100$	= Pounds
Hundredweight (short)	$\times 0.05$	= Ton, short
Hundredweight (short)	$\times 0.04536$	= Tonnes, metric
Hundredweight (short)	$\times 0.044643$	= Tonnes, long

## I

Inch	$\times 254 \times 10^6$	= Angstrom
Inch	$\times 2.54$	= Centimeter
Inch	$\times 0.833 \times 10^{-3}$	= Chain, engineer
Inch	$\times 1.2626 \times 10^{-3}$	= Chain, Gunter
Inch	$\times 0.254$	= Decrometer
Inch	$\times 0.08333$	= Foot, USA
Inch	$\times 0.0254$	= Meter
Inch	$\times 1.578 \times 10^{-5}$	= Mile, statute, USA
Inch	$\times 25.4$	= Millimeters
Inch	$\times 1,000$	= Mils
Inch	$\times 5.05 \times 10^{-3}$	= Rods
Inch	$\times 0.02778$	= Yards

Inch, mercury	$\times 0.03342$	= Atmospheres
Inch, mercury	$\times 1.133$	= Feet of water
Inch, mercury	$\times 13.61$	= Inch height, water
Inch, mercury	$\times 70.73$	= Pound/square foot
Inch, mercury	$\times 0.49116$	= Pound/square inch
Inch, mercury	$\times 0.03453$	= Kilograms/square centimeter
Inches of mercury	$\times 345.3$	= Kilograms/square meter
Inch-pound	$\times 1.07 \times 10^{-4}$	= Btu
Inch-pound	$\times 0.0833$	= Foot-pound
Inch, water, 4°C	$\times 2.458 \times 10^{-3}$	= Atmospheres
Inch, water, 4°C	$\times 0.07355$	= Inches of mercury
Inch, water, 4°C	$\times 2.54 \times 10^{-3}$	= Kilograms/square centimeter
Inch, water, 4°C	$\times 0.5781$	= Ounces/square inch
Inch, water, 4°C	$\times 5.204$	= Pounds/square foot
Inch, water, 4°C	$\times 0.03613$	= Pounds/square inch

**J**

Joule	$\times 9.48 \times 10^{-4}$	= Btu
Joule	$\times 10^7$	= Erg
Joule	$\times 0.7376$	= Foot-pound
Joule	$\times 2.389 \times 10^{-4}$	= Kilogram-calorie
Joule	$\times 1.0197 \times 10^4$	= Gram-centimeter
Joule	$\times 2.778 \times 10^{-4}$	= Watt-hours

**K**

Kilogram	$\times 1,000$	= Grams
Kilogram	$\times 70.93$	= Poundals
Kilogram	$\times 2.205$	= Pounds (avdp)
Kilogram	$\times 9.842 \times 10^{-4}$	= Tonnes, long
Kilogram	$\times 1.102 \times 10^{-3}$	= Tonnes, short
Kilogram	$\times 0.001$	= Tonnes, metric
Kilocalories	$\times 3.968$	= Btu
Kilocalories	$\times 3,088$	= Foot-lbs.
Kilocalories	$\times 1.56 \times 10^{-3}$	= Hp.-hrs.
Kilocalories	$\times 1.163 \times 10^{-3}$	= Kilowatt-hrs.
Kilocalories	$\times 0.205$	= Btu/square foot/hr./°F
Kilogram-meters	$\times 9.294 \times 10^{-3}$	= Btu
Kilograms/cubic meter	$\times 0.001$	= Grams/cubic centimeter
Kilograms/cubic meter	$\times 0.06243$	= Pounds/cubic foot
Kilograms/cubic meter	$\times 3.613 \times 10^{-5}$	= Pounds/cubic inch
Kilograms/cubic meter	$\times 3.405 \times 10^{-10}$	= Pounds/mil-foot
Kilograms/cubic meter	$\times 8.428 \times 10^{-3}$	= Ton, short/cubic yard
Kilograms/meter	$\times 10$	= Gram/centimeter
Kilograms/meter	$\times 391.983$	= Gram/inch
Kilograms/meter	$\times 0.672$	= Pounds/foot
Kilograms/meter	$\times 0.056$	= Pounds/inch
Kilograms/square centimeter	$\times 0.9678$	= Atmospheres
Kilograms/square centimeter	$\times 32.81$	= Feet of water
Kilograms/square centimeter	$\times 28.96$	= Inch of mercury
Kilograms/square centimeter	$\times 2,048$	= Pounds/square foot
Kilograms/square centimeter	$\times 14.223$	= Pounds/square inch
Kilograms/square meter	$\times 9.678 \times 10^{-5}$	= Atmospheres
Kilograms/square meter	$\times 3.281 \times 10^{-3}$	= Feet of water
Kilograms/square meter	$\times 2.896 \times 10^{-3}$	= Inches of mercury
Kilograms/square meter	$\times 0.03937$	= Inches of water
Kilograms/square meter	$\times 0.2048$	= Pounds/square foot
Kilograms/square meter	$\times 1.422 \times 10^{-3}$	= Pounds/square inch
Kilometers	$\times 3281$	= Feet
Kilometers	$\times 3.937 \times 10^4$	= Inches
Kilometers	$\times 1,000$	= Meters
Kilometers	$\times 0.6214$	= Miles (statute)
Kilometers	$\times 0.5295$	= Miles (nautical)
Kilometers	$\times 1,094$	= Yards
Kilometers/hour	$\times 27.78$	= Centimeters/second
Kilometers/hour	$\times 54.68$	= Feet/minute
Kilometers/hour	$\times 0.9113$	= Feet/second

Kilometers/hour	$\times 0.5396$	= Knots
Kilometers/hour	$\times 16.67$	= Meters/minute
Kilometers/hour	$\times 0.6214$	= Miles/hour
Kilowatts	$\times 56.92$	= Btu/minute
Kilowatts	$\times 1.35972$	= Horsepower, metric
Kilowatts	$\times 1.341$	= Horsepower, USA
Kilowatts	$\times 1,000$	= Watts
Kilowatts	$\times 4.426 \times 10^4$	= Foot-lbs./minute
Kilowatts	$\times 737.6$	= Foot-lbs./second
Kilowatt-hrs.	$\times 2.655 \times 10^6$	= Foot-lbs.
Kilowatt-hours	$\times 1000$	= Watt hours
Kilowatt-hours	$\times 3,413$	= Btu
Kilowatt-hours	$\times 3.6 \times 10^{13}$	= Ergs
Kilowatt-hours	$\times 1.36$	= Horsepower-hour, metric
Kilowatt-hours	$\times 1.341$	= Horsepower-hour, USA
Kilowatt-hours	$\times 3.6 \times 10^6$	= Joules
Kilowatt-hours	$\times 860.5$	= Kilogram-calories
Kilowatt-hours	$\times 3.671 \times 10^5$	= Kilogram-meters
Kilowatt-hours	$\times 3.53$	= Pounds of water*
Kilowatt-hours	$\times 22.75$	= Pounds of water†
Kip	$\times 1$	= Kilopound
Kip	$\times 1,000$	= Pound
Knott, USA	$\times 51.48$	= Centimeter/second
Knott, USA	$\times 6080.2$	= Feet/hour
Knott, USA	$\times 1.8532$	= Kilometer/hour
Knott, USA	$\times 30.887$	= Meter/minute
Knott, USA	$\times 1.15155$	= Mile/hour (statute)
Knott, USA	$\times 2027$	= Yards/hour

**L**

League, land	$\times 24$	= Furlong
League, land	$\times 4.828$	= Kilometer
League, land	$\times 3$	= Mile
League, marine	$\times 5.56$	= Kilometer
League, marine	$\times 3$	= Mile, nautical
League, marine	$\times 3.45$	= Mile, statute
Light year	$\times 5.9 \times 10^{12}$	= Miles
Light year	$\times 9.46091 \times 10^{12}$	= Kilometers
Links, engineers'	$\times 12$	= Inches
Links, surveyors'	$\times 7.92$	= Inches
Links, surveyors'	$\times 0.66$	= Feet
Links, surveyors'	$\times 0.22$	= Yard
Liters	$\times 0.02838$	= Bushels, USA, dry
Liters	$\times 100$	= Centiliters
Liters	$\times 1,000$	= Cubic centimeters
Liters	$\times 0.035316$	= Cubic feet
Liters	$\times 61.02$	= Cubic inches
Liters	$\times 6.291 \times 10^{-3}$	= Barrels, oil, USA
Liters	$\times 61.027$	= Cubic inches
Liters	$\times 0.001$	= Cubic meter
Liters	$\times 1.308 \times 10^{-3}$	= Cubic yard
Liters	$\times 0.2642$	= Gallon, USA, liquid
Liters	$\times 0.2199$	= Gallons (Imperial)
Liters	$\times 1.7598$	= Pint, USA, dry
Liters	$\times 2.1134$	= Pint, USA, liquid
Liters	$\times 2.202$	= Pounds of water
Liters/minute	$\times 5.886 \times 10^{-4}$	= Cubic foot/second
Liters/minute	$\times 4.403 \times 10^{-3}$	= Gallons/second
Liters/second	$\times 2.1186$	= Cubic feet/minute
Lumen	$\times 0.07958$	= Candlepower
Lumen	$\times 1.47 \times 10^{-3}$	= Watt
Lumens/square foot	$\times 1$	= Foot-candles
Lux	$\times 0.0929$	= Foot-candles

\* Evaporated from and at 212° F

† Raised from 62° to 212° F

**M**

Maas	$\times 1.5$	= Liter
Meter	$\times 10^{10}$	= Angstrom units
Meter	$\times 100$	= Centimeter
Meter	$\times 3.2808$	= Feet, USA
Meter	$\times 0.01$	= Hectometer
Meter	$\times 39.37$	= Inches
Meter	$\times 0.001$	= Kilometer
Meter	$\times 5.396 \times 10^{-4}$	= Miles, nautical
Meter	$\times 6.214 \times 10^{-4}$	= Miles, statute
Meter	$\times 1000$	= Millimeters
Meter	$\times 1.094$	= Yards
Meter	$\times 1.179$	= Vara
Meters/minute	$\times 1.667$	= Centimeters/second
Meters/minute	$\times 3.281$	= Feet/minute
Meters/minute	0.05468	= Feet/second
Meters/minute	$\times 0.06$	= Kilometers/hour
Meters/minute	$\times 0.03238$	= Knots
Meters/minute	$\times 0.03728$	= Miles/hour
Meters/second	$\times 196.8$	= Feet/minute
Meters/second	$\times 3.281$	= Feet/second
Meters/second	$\times 3.6$	= Kilometers/hour
Meters/second	$\times 0.06$	= Kilometers/minute
Meters/second	$\times 2.237$	= Miles/hour
Meters/second	$\times 0.03728$	= Miles/minute
Microns	$\times 39.37 \times 10^{-6}$	= Inches
Microns	$\times 1 \times 10^{-6}$	= Meters
Micron	$\times 0.0001$	= Centimeter
Micron	$\times 1000$	= Millimicron
Mile, USA, nautical	$\times 6,080.2$	= Feet, USA
Mile, USA, nautical	$\times 6,080$	= Feet, British
Mile, USA, nautical	$\times 72,962.5$	= Inches
Mile, USA, nautical	$\times 1.853$	= Kilometer
Mile, USA, nautical	$\times 0.333$	= League
Mile, USA, nautical	$\times 1,853.248$	= Meter
Mile, USA, nautical	$\times 1.15155$	= Mile, USA, statute
Mile, USA, nautical	$\times 2,026.73$	= Yard
Mile, USA, statute	$\times 5,280$	= Feet, USA
Miles, USA, statute	$\times 8$	= Furlongs
Miles, USA, statute	$\times 63,360$	= Inches
Miles, USA, statute	$\times 1.60935$	= Kilometer
Miles, USA, statute	$\times 8,000$	= Link
Miles, USA, statute	$\times 1,609.35$	= Meters
Miles, USA, statute	$\times 0.8684$	= Mile, USA, nautical
Miles, USA, statute	$\times 1,900.8$	= Vara
Miles, USA, statute	$\times 1,706$	= Yard
Miles/hour	$\times 44.7$	= Centimeters/second
Miles/hour	$\times 88$	= Feet/minute
Miles/hour	$\times 1.467$	= Feet/second
Miles/hour	$\times 1.609$	= Kilometers/hour
Miles/hour	$\times 0.02682$	= Kilometers/minute
Miles/hour	$\times 0.8684$	= Knots
Miles/hour	$\times 26.82$	= Meters/minute
Miles/hour	$\times 0.4470$	= Meters/second
Miles/hour	$\times 0.01667$	= Miles/minute
Miles/minute	$\times 5,280$	= Feet/minute
Miles/minute	$\times 316,800$	= Feet/hour
Miles/minute	$\times 88$	= Feet/second
Miles/minute	$\times 60$	= Miles/hour
Miles/minute	$\times 1.609$	= Kilometers/minute
Miles/minute	$\times 0.8684$	= Knots/minute
Millimeter	$\times 0.1$	= Centimeter
Millimeter	$\times 3.281 \times 10^{-3}$	= Feet
Millimeter	$\times 0.03937 \times 10^{-2}$	= Inches
Millimeter	$\times 10^{-6}$	= Kilometers
Millimeter	$\times 0.001$	= Meters
Millimeter	$\times 6.214 \times 10^{-7}$	= Miles
Millimeter	$\times 39.37$	= Mils
Millimeter	$\times 1.094 \times 10^{-3}$	= Yards
Million gallons/day	$\times 1.54723$	= Cubic feet/second

Mils	$\times 2.540 \times 10^{-3}$	= Centimeters
Mils	$\times 8.333 \times 10^{-5}$	= Feet
Mils	$\times 0.001$	= Inches
Mils	$\times 2.540 \times 10^{-8}$	= Kilometers
Mils	$\times 2.778 \times 10^{-5}$	= Yards

**N**

Nail	$\times 2.5$	= Inch
Nepers	$\times 8.686$	= Decibels
Newton	$\times 1 \times 10^5$	= Dynes

**O**

Ounces (avdp)	$\times 16$	= Drams
Ounces (avdp)	$\times 437.5$	= Grains
Ounces (avdp)	$\times 28.349527$	= Grams
Ounces (avdp)	$\times 0.0625$	= Pounds
Ounces (avdp)	$\times 0.9115$	= Ounces, troy
Ounces	$\times 2.79 \times 10^{-5}$	= Tonnes, long
Ounces	$\times 2.835 \times 10^{-5}$	= Tonnes, metric
Ounces, fluid	$\times 1.805$	= Cubic inches
Ounces, fluid	$\times 0.02957$	= Liters
Ounces, troy	$\times 480$	= Grains
Ounces, troy	$\times 31.103481$	= Grams
Ounces, troy	$\times 1.09714$	= Ounces, avoirdupois
Ounces, troy	$\times 0.08333$	= Pounds, troy
Ounces/square inch	$\times 0.0625$	= Pounds/square inch

**P**

Parsec	$\times 19 \times 10^{12}$	= Miles, USA, statute
Parsec	$\times 3.084 \times 10^{13}$	= Kilometers
Parts/million	$\times 0.05833$	= Grains/gallon, USA
Parts/million	$\times 0.07016$	= Grains/gallon, British
Parts/million	$\times 8.345$	= Pounds/million gallons, USA
Peck, British	$\times 554.6$	= Cubic inches
Peck, British	$\times 2$	= Gallons, British
Peck, British	$\times 9.0919$	= Liters
Peck, USA	$\times 0.25$	= Bushels
Peck, USA	$\times 537.605$	= Cubic inches
Peck, USA	$\times 8.809582$	= Liters
Peck, USA	$\times 8$	= Quarts, dry
Peck, USA	$\times 9.3092$	= Quarts, liquid
Pennyweights, troy	$\times 24$	= Grains
Pennyweights, troy	$\times 0.05$	= Ounces, troy
Pennyweights, troy	$\times 1.55517$	= Grams
Pennyweights, troy	$\times 4.1667 \times 10^{-3}$	= Pounds, troy
Percent	$\times 10^4$	= Parts/million
Pfund, Germany	$\times 500$	= Gram
Pint, USA, dry	$\times 0.015625$	= Bushel
Pint, USA, dry	$\times 550.6136$	= Cubic centimeter
Pint, USA, dry	$\times 0.01945$	= Cubic feet
Pint, USA, dry	$\times 33.6$	= Cubic inches
Pint, USA, dry	$\times 2$	= Cup
Pint, USA, dry	$\times 0.125$	= Gallon, USA, dry
Pint, USA, dry	$\times 0.145545$	= Gallon, USA, liquid
Pint, USA, dry	$\times 0.5506$	= Liter
Pint, USA, dry	$\times 0.0625$	= Peck
Pint, USA, dry	$\times 0.5$	= Quart, USA, dry
Pint, USA, dry	$\times 0.58182$	= Quart, USA, liquid
Pint, USA, liquid	$\times 437.2$	= Cubic centimeters
Pint, USA, liquid	$\times 0.01671$	= Cubic feet
Pint, USA, liquid	$\times 28.875$	= Cubic inch
Pint, USA, liquid	$\times 2$	= Cup
Pint, USA, liquid	$\times 0.1074$	= Gallon, USA, dry
Pint, USA, liquid	$\times 0.125$	= Gallon, USA, liquid

Pint, USA, liquid	$\times 4$	= Gill
Pint, USA, liquid	$\times 0.4732$	= Liters
Pint, USA, liquid	$\times 16$	= Ounces
Pint, USA, liquid	$\times 0.5$	= Quarts, USA, liquid
Pint, USA, liquid	$\times 0.42968$	= Quarts, USA, dry
Pint, USA, liquid	$\times 128$	= Dram, fluid
Poise	$\times 100$	= Centipoise
Pole	$\times 16.5$	= Feet
Pole	$\times 5.0292$	= Meter
Pole	$\times 1$	= Rod
Pole	$\times 5.5$	= Yard
Ponce	$\times 2.71$	= Centimeter
Pood	$\times 1,000$	= Cubic inch
Pood	$\times 40$	= Funt
Pood	$\times 4.32$	= Gallon, USA
Pood	$\times 16.3805$	= Kilogram
Poundals	$\times 13,826$	= Dynes
Poundals	$\times 14.098$	= Grams
Poundals	$\times 1.383 \times 10^{-3}$	= Joules/centimeter
Poundals	$\times 0.1383$	= Joules/meter
Poundals	$\times 0.0141$	= Kilograms
Poundals	$\times 0.1383$	= Newton
Poundals	$\times 0.03108$	= Pound-force
Pound-mol	$\times 359.05$	= Cubic feet at 32° F and 1 atm
Pounds	$\times 1/\text{Mol. Wt.}$	= Pound-mols
Pounds	$\times 2267.9616$	= Carats
Pounds (avdp)	$\times 256$	= Drams
Pounds (avdp)	$\times 7,000$	= Grains
Pounds (avdp)	$\times 453.5924$	= Grams (metric)
Pounds	$\times 0.04448$	= Joules/centimeters
Pounds (avdp)	$\times 0.4536$	= Kilograms
Pounds (avdp)	$\times 16$	= Ounces
Pounds (avdp)	$\times 14.5833$	= Ounces, troy
Pounds	$\times 32.174$	= Poundals
Pounds (avdp)	$\times 1.21528$	= Pounds, troy
Pounds	$\times 4.464 \times 10^{-4}$	= Tons, long
Pounds	$\times 4.536 \times 10^{-4}$	= Tonnes, metric
Pounds	$\times 5 \times 10^{-4}$	= Tonnes, short
Pounds, troy	$\times 5,760$	= Grains
Pounds, troy	$\times 373.24177$	= Grams
Pounds, troy	$\times 13.1657$	= Ounces, avoirdupois
Pounds, troy	$\times 12$	= Ounces, troy
Pounds, troy	$\times 240$	= Pennyweights, troy
Pounds, troy	$\times 0.822857$	= Pounds, avoirdupois
Pounds, troy	$\times 3.6735 \times 10^{-4}$	= Tonnes, long
Pounds, troy	$\times 3.7324 \times 10^{-4}$	= Tonnes, metric
Pounds, troy	$\times 4.1143 \times 10^{-4}$	= Tonnes, short
Pounds of water	$\times 0.01602$	= Cubic feet
Pounds of water	$\times 27.68$	= Cubic inches
Pounds of water	$\times 0.1198$	= Gallons
Pounds of water/minute	$\times 2.67 \times 10^{-4}$	= Cubic feet/second
Pounds/cubic foot	$\times 0.01602$	= Grams/cubic centimeters
Pounds/cubic foot	$\times 16.02$	= Kilograms/cubic meter
Pounds/cubic foot	$\times 5.787 \times 10^{-4}$	= Pounds/cubic inch
Pounds/cubic foot	$\times 27$	= Pounds/cubic yard
Pounds/cubic inch	$\times 27.68$	= Grams/cubic centimeter
Pounds/cubic inch	$\times 2.768 \times 10^4$	= Kilograms/cubic meter
Pounds/cubic inch	$\times 1,728$	= Pounds/cubic foot
Pounds/cubic inch	$\times 46,656$	= Pounds/cubic yard
Pounds/hour	$\times 10.714 \times 10^{-3}$	= Tonnes/day, long
Pounds/hour	$\times 12 \times 10^{-3}$	= Tonnes/day, short
Pounds/hour	$\times 10.886 \times 10^{-3}$	= Tonnes/day, metric
Pounds/hour	$\times 0.45359$	= Kilograms/hour
Pounds/square foot	$\times 4.725 \times 10^{-4}$	= Atmospheres
Pounds/square foot	$\times 0.01602$	= Feet of water
Pounds/square foot	$\times 0.01414$	= Inches of mercury
Pounds/square foot	$\times 4.8824$	= Kilograms/square meter
Pounds/square foot	$\times 0.1111$	= Ounce/square inch
Pounds/square foot	$\times 0.107638$	= Pound/square centimeter
Pounds/square foot	$\times 6.944 \times 10^{-3}$	= Pound/square inch
Pounds/square foot	$\times 10.76387$	= Pound/square meter

Pounds/square inch	$\times 0.068046$	= Atmospheres
Pounds/square inch	$\times 2.307$	= Feet of water
Pounds/square inch	$\times 27.7$	= Inch of water
Pounds/square inch	$\times 2.036$	= Inch of mercury
Pounds/square inch	$\times 0.0703$	= Kilogram/square centimeter
Pounds/square inch	$\times 703.1$	= Kilogram/square meter
Pounds/square inch	$\times 51.714$	= Millimeters of mercury
Pounds/square inch	$\times 2,304$	= Ounce/square foot
Pounds/square inch	$\times 144$	= Pound/square foot

**Q**

Quadrant	$\times 0.25$	= Circumference
Quadrant	$\times 90$	= Degrees
Quadrant	$\times 5,400$	= Minutes
Quadrant	$\times 1.571$	= Radians
Quarts, USA, dry	$\times 0.03125$	= Bushel
Quarts, USA, dry	$\times 1,101.2$	= Cubic centimeter
Quarts, USA, dry	$\times 0.03889$	= Cubic foot
Quarts, USA, dry	$\times 67.20$	= Cubic inches
Quarts, USA, dry	$\times 1.1012$	= Liter
Quarts, USA, dry	$\times 1.16365$	= Quart, USA, liquid
Quarts, USA, liquid	$\times 946.331$	= Cubic centimeter
Quarts, USA, liquid	$\times 0.03342$	= Cubic foot
Quarts, USA, liquid	$\times 57.75$	= Cubic inches
Quarts, USA, liquid	$\times 9,464 \times 10^{-4}$	= Cubic meters
Quarts, USA, liquid	$\times 1.238 \times 10^{-3}$	= Cubic yard
Quarts, USA, liquid	$\times 4$	= Cup
Quarts, USA, liquid	$\times 256$	= Dram fluid
Quarts, USA, liquid	$\times 0.25$	= Gallons
Quarts, USA, liquid	$\times 0.946331$	= Liter
Quarts, USA, liquid	$\times 5.9523 \times 10^{-3}$	= Oil, barrel
Quarts, USA, liquid	$\times 32$	= Ounces
Quarts, USA, liquid	$\times 2$	= Pint
Quarts, USA, liquid	$\times 0.8594$	= Quart, USA, dry

**R**

Radians	$\times 57.3$	= Degrees
Radians	$\times 3,438$	= Minutes
Radians	$\times 0.6366$	= Quadrants
Radians	$\times 2.063 \times 10^5$	= Seconds
Revolutions/minute	$\times 6.0$	= Degrees/second
Revolutions/minute	$\times 0.1047$	= Radians/second
Revolutions/minute	$\times 0.01667$	= Revolutions/second
Rod	$\times 0.165$	= Chain, engineer
Rod	$\times 0.25$	= Chain, Gunters
Rod	$\times 16.5$	= Foot
Rod	$\times 0.025$	= Furlong
Rod	$\times 198$	= Inch
Rod	$\times 25$	= Link
Rod	$\times 5.029$	= Meter
Rod	$\times 5.5$	= Yard

**S**

Seconds, angle	$\times 2.778 \times 10^{-4}$	= Degrees
Seconds, angle	$\times 16.67 \times 10^{-3}$	= Minutes
Seconds, time	$\times 2.777 \times 10^{-4}$	= Hour
Seconds, time	$\times 0.166$	= Minutes
Slugs	$\times 14.59$	= Kilograms
Slugs	$\times 32.17$	= Pounds
Slugs/cubic foot	$\times 0.5154$	= Gm/cubic centimeter
Snow, cubic foot	$\times 7.2$	= Pounds, 32° F
Snow, inch deep	$\times 0.1$	= Inch, water
Square centimeter	$\times 1.076 \times 10^{-3}$	= Square foot
Square centimeter	$\times 0.155$	= Square inch

Square centimeter	$\times 0.0001$	= Square meter
Square centimeter	$\times 3.861 \times 10^{-11}$	= Square miles
Square centimeter	$\times 100$	= Square millimeters
Square centimeter	$\times 1.196 \times 10^{-4}$	= Square yards
Square feet, USA	$\times 2.296 \times 10^{-5}$	= Acre
Square feet, USA	$\times 9.29 \times 10^{-4}$	= Are
Square feet, USA	$\times 929.034$	= Square centimeters
Square feet, USA	$\times 144$	= Square inches
Square feet, USA	$\times 0.0929$	= Square meter
Square feet, USA	$\times 3.587 \times 10^{-8}$	= Square miles
Square feet, USA	$\times 9.29 \times 10^4$	= Square millimeters
Square feet, USA	$\times 0.1111$	= Square yards
Square inches	$\times 6.452$	= Square centimeters
Square inches	$\times 6.944 \times 10^{-3}$	= Square feet
Square inches	$\times 645.2$	= Square millimeters
Square inches	$\times 7.716 \times 10^{-4}$	= Square yard
Square kilometer	$\times 247.1$	= Acre
Square kilometer	$\times 100$	= Hectare
Square kilometer	$\times 10.76 \times 10^6$	= Square feet
Square kilometer	$\times 1.55 \times 10^9$	= Square inches
Square kilometer	$\times 10^6$	= Square meters
Square kilometer	$\times 0.3861$	= Square mile, USA
Square kilometer	$\times 1.196 \times 10^6$	= Square yards
Square meters	$\times 2.471 \times 10^{-4}$	= Acre
Square meters	$\times 0.01$	= Are
Square meters	$\times 0.0001$	= Hectare
Square meters	$\times 10,000$	= Square centimeters
Square meters	$\times 10.764$	= Square feet
Square meters	$\times 1,550$	= Square inches
Square meters	$\times 3.861 \times 10^{-7}$	= Square miles
Square meters	$\times 1.196$	= Square yards
Square miles	$\times 640$	= Acre
Square miles	$\times 259$	= Hectare
Square miles	$\times 27.88 \times 10^6$	= Square feet
Square miles	$\times 2.59$	= Square kilometers
Square miles	$\times 2.59 \times 10^6$	= Square meters
Square miles	$\times 3.098 \times 10^6$	= Square yards
Square millimeters	$\times 0.01$	= Square centimeters
Square millimeters	$\times 1.076 \times 10^{-5}$	= Square feet
Square millimeters	$\times 1.55 \times 10^{-3}$	= Square inches
Square rods	$\times 0.00625$	= Acre
Square rods	$\times 272.25$	= Square feet
Square rods	$\times 25.293$	= Square meter
Square rods	$\times 30.25$	= Square yard
Square vara	$\times 7.716$	= Square feet
Square yard	$\times 2.066 \times 10^{-4}$	= Acres
Square yard	$\times 8361$	= Square centimeter
Square yard	$\times 9$	= Square feet
Square yard	$\times 1,296$	= Square inches
Square yard	$\times 0.8361$	= Square meters
Square yard	$\times 3.228 \times 10^{-7}$	= Square miles
Square yard	$\times 8.361 \times 10^5$	= Square millimeters
Square yard	$\times 0.03306$	= Square rods
Stone	$\times 14$	= Pound
Stone	$\times 6.35$	= Kilogram

**T**

Tablespoon	$\times 0.0625$	= Cup
Tablespoon	$\times 3$	= Teaspoon
Teaspoon	$\times 0.0208$	= Cup
Teaspoon	$\times 0.333$	= Tablespoon
Temperature, °C + 17.78	$\times 1.8$	= °F
Temperature, °F - 32	$\times 0.5556$	= °C
Ton, long	$\times 1,016$	= Kilogram
Ton, long	$\times 2,240$	= Pounds
Ton, long	$\times 1.016$	= Metric tonnes
Ton, long	$\times 1.12$	= Short tonnes
Ton, metric	$\times 7.454$	= Barrel, oil, 36 API

Ton, metric	$\times 1,000$	= Kilograms
Ton, metric	$\times 2,205$	= Pounds
Ton, metric	$\times 0.9842$	= Ton, long
Ton, metric	$\times 1.1023$	= Ton, short
Ton, shipping, USA	$\times 40$	= Cubic feet
Ton, shipping, USA	$\times 2.8317$	= Cubic meter
Ton, shipping, USA	$\times 1.050$	= Ton, shipping, British
Ton, short	$\times 40$	= Cubic feet
Ton, short	$\times 268.8$	= Gallons, USA, liquid
Ton, short	$\times 4$	= Hogshead
Ton, short	$\times 907.18486$	= Kilograms
Ton, short	$\times 1,000$	= Liter
Ton, short	$\times 32,000$	= Ounces
Ton, short	$\times 2,000$	= Pounds
Ton, short	$\times 0.89286$	= Tonnes, long
Ton, short	$\times 0.907$	= Tonnes, metric
Tonnes, short/square foot	$\times 9,765$	= Kilograms/square meter
Tonnes, short/square foot	$\times 2,000$	= Pounds/square inch
Tonnes, short/day	$\times 83.333$	= Pounds/hour
Tonnes, short/day	$\times 0.16643$	= Gallons/minute
Tonnes, short/day	$\times 0.9072$	= Tonnes, metric/day
Tonnes, short/day	$\times 0.8929$	= Tonnes, long/day
Tonnes, short/day	$\times 37.8$	= Kilograms/hour
Tonnes, metric/day	$\times 91.859$	= Pounds/hour
Tonnes, metric/day	$\times 41.667$	= Kilograms/hour
Tonnes, metric/day	$\times 0.9843$	= Tonnes, long/day
Tonnes, metric/day	$\times 1.1023$	= Tonnes, short/day
Tonnes, long/day	$\times 1.12$	= Tonnes, short/day
Tonnes, long/day	$\times 1.016$	= Tonnes, metric/day
Tonnes, long/day	$\times 93.333$	= Pounds/hour
Tonnes, long/day	$\times 42.335$	= Kilograms/hour

**V**

Vara	$\times 2.7777$	= Feet
Vara	$\times 33.3333$	= Inch
Vara	$\times 0.9259$	= Yard
Volt/inch	$\times 0.3937$	= Volt/centimeter

**W**

Water, 62° F, Gallon	$\times 8.3311$	= Pound
Water height in feet	$\times 0.4335$	= Pound/square inch
Water height in feet	$\times 0.03048$	= Kilograms/square centimeters
Water height in inches	$\times 0.03613$	= Pound/square inch
Water height in inches	$\times 0.00254$	= Kilograms/square centimeter
Water height in meters	$\times 1.42067$	= Pound/square inch
Water height in meters	$\times 0.100$	= Kilograms/square centimeters
Watts	$\times 3.4128$	= Btu/hour
Watts	$\times 0.05688$	= Btu/minute
Watts	$\times 107$	= Ergs/second
Watts	$\times 44.27$	= Foot-pounds/minute
Watts	$\times 0.7378$	= Foot-pounds/second
Watts	$\times 1.341 \times 10^{-3}$	= Horsepower, USA
Watts	$\times 1.36 \times 10^{-3}$	= Horsepower, metric
Watts	$\times 0.001$	= Kilowatt
Watts	$\times 1$	= Joules/second
Watts (abs.)	$\times 0.056884$	= Btu (mean)/minute
Watt-hours	$\times 3.4128$	= Btu
Watt-hours	$\times 3.60 \times 10^{10}$	= Ergs
Watt-hours	$\times 2,656$	= Foot-pounds
Watt-hours	$\times 858.85$	= Gram-calories
Watt-hours	$\times 1.341 \times 10^{-3}$	= Horsepower-hours, USA
Watt-hours	$\times 1.3596 \times 10^{-3}$	= Horsepower-hours, metric
Watt-hours	$\times 0.8605$	= Kilogram-calories
Watt-hours	$\times 367.2$	= Kilogram-meters
Watt-hours	$\times 0.001$	= Kilowatt-hours

**Y**

Yard, USA	$\times 91.4402$	= Centimeter
Yard, USA	$\times 3$	= Feet
Yard, USA	$\times 36$	= Inch
Yard, USA	$\times 9.144 \times 10^{-4}$	= Kilometer
Yard, USA	$\times 0.9144$	= Meter
Yard, USA	$\times 4.934 \times 10^{-4}$	= Mile, nautical, USA
Yard, USA	$\times 5.682 \times 10^{-4}$	= Mile, Statute, USA
Yard, USA	$\times 914.402$	= Millimeters
Yard, USA	$\times 0.1818$	= Rod
Year	$\times 8,765$	= Hours
Year	$\times 525,948$	= Minutes

## D-2 Physical Property Conversion Factors

$$\begin{aligned}\text{Acceleration of gravity} &= 32.172 \text{ ft./sec./sec.} \\ &= 980.6 \text{ cm./sec./sec.}\end{aligned}$$

**Electrical conductance:**

$$\begin{aligned}1 \text{ mho} &= 1 \text{ ohm}^{-1} \\ &= 10^{-4} \text{ megamho} \\ &= 10^6 \text{ micromho}\end{aligned}$$

### Heat Value of Fuel

$$\begin{aligned}\text{Lower heating value} &= \text{Higher heating value} - 10.3(9H_2 + H_2O), \text{ Btu/lb.} \\ \text{where: } H_2 &= \text{weight \% hydrogen in fuel}\end{aligned}$$

$H_2O = \text{weight \% water vapor in fuel}$

$$\text{GPM} = (\text{pounds/hour}) / (500 \times \text{Sp.Gr.})$$

$$\text{Velocity, feet/sec.} = \frac{0.321 \text{ (GPM)}}{(\text{Flow Area, sq.in.})}$$

$$\text{Head, feet} = 2.31 \text{ (Pressure or head, psi) / Sp.Gr.}$$

$$\text{Brake horsepower, BHP} = \frac{(\text{GPM}) \text{ (Sp.Gr.) (Head, feet)}}{3960 \text{ (Efficiency, fraction)}}$$

### Weight/Volume (avoirdupois unless otherwise stated)

$$\text{Density of sea water} = 1.025 \text{ grams/cc.}$$

$$1 \text{ gram-molecular volume of a gas at 760 mm. Hg and } 0^\circ \text{ C.} = 22.4 \text{ liters}$$

$$1 \text{ U. S. gallon} = (8.34 \times \text{Sp.Gr. of fluid}), \text{ pounds}$$

$$\text{Weight of one cu.ft. liquid} = (62.32 \text{ pounds} \times \text{Sp.Gr. of fluid}), \text{ pounds/cu.ft.}$$

$$1 \text{ pound avoirdupois} = 1.2153 \text{ pound apothecaries'}$$

$$1 \text{ grain avoirdupois} = 1 \text{ grain troy} = 1 \text{ grain apothecaries' weight}$$

### Air Analysis\*

	By Weight %	By Volume %
Nitrogen	75.47	78.2
Oxygen	23.19	21.0

\* Neglects trace gases such as argon, xenon, helium, krypton and assumes dry basis.

### Gas Constants, (R), Universal

$$\begin{aligned}R &= 0.0821 \text{ (atm) (liter)/(g-mol) } (^\circ\text{K}) \\ &= 1.987 \text{ (g-cal.)/(g-mol) } (^\circ\text{K}) \\ &= 1.987 \text{ Btu/(lb.-mol) } (^\circ\text{R}) \\ &= 1.987 \text{ (Chu)/(lb.-mol) } (^\circ\text{K}) \\ &= 8.314 \text{ joules/(g-mol) } (^\circ\text{K}) \\ &= 1.546 \text{ (ft.)(lb.force)/(lb.-mol) } (^\circ\text{R})\end{aligned}$$

$$\begin{aligned}&= 10.73 \text{ (lb-force/sq. in abs.) (cu.ft.)/(lb.-mol) } (^\circ\text{R}) \\ &= 18,510 \text{ (lb.-force/sq.in.) (cu.in.)/(lb.-mol) } (^\circ\text{R}) \\ &= 0.7302 \text{ (Atm) (cu.ft.)/(lb.-mol) } (^\circ\text{R})\end{aligned}$$

$$R_i = R/\text{mol.wt. gas}$$

where:  $R_i$  = individual gas constant

$$\text{Avogadro Constant, } N_a = 6.02252 \times 10^{23} \text{ molecules/mol}$$

### Density Vapor or Gases (Ideal), $\rho$

$$\rho = \frac{\text{mol. wt., vapor}}{359} \left( \frac{14.7 + p}{14.7} \right) \left( \frac{460 + 32}{460 + ^\circ\text{F}} \right), \text{ lbs./cu.ft.}$$

where:  $p$  = gage pressure at actual condition, psig

$^\circ\text{F}$  = fahrenheit temperature at actual condition

$$\rho = \frac{144 P}{R_i T}, \text{ pounds/cu.ft.}$$

where:  $P$  = absolute pressure, pounds/sq. in. abs.

$T$  = absolute temperature, "Rankine, "R

$o$  = standard conditions ( $0^\circ\text{C}$  & 760 mm Hg)

$$V = V_o (P_o/P') (T/T_o)$$

$$P'V = 1543 \text{ nT}; P' = \text{PSF abs.}; V = \text{cu. ft.}$$

$$T = "R"; n = \text{lb. moles}$$

$$\text{cu.ft.} = \frac{\text{lb}}{\text{MW}} \left( \frac{273 + ^\circ\text{C}}{273} \right) \left( \frac{14.7}{p + 14.7} \right) \text{ at } p, "C$$

### Specific Volume, Gas or Vapor

$$\bar{V} = 1/\rho, \text{ cu.ft./pound}$$

$$\text{Velocity of sound in dry air @ } 0^\circ \text{ C. and 1 atm.} = 1,089 \text{ ft./sec.}$$

$$\text{Density of dry air @ } 0^\circ \text{ C. and 1 atm.}$$

$$= 0.001293 \text{ gm/cu.cm.}$$

$$= 0.0808 \text{ lb./cu.ft.}$$

### Viscosity (Dynamic)

$$\begin{aligned}1 \text{ Poise} &= 1 \text{ gram/cm.sec.} = 1 \text{ dyne-sec./sq.cm.} \\ &= 0.1 \text{ kg/meter-sec.}\end{aligned}$$

$$1 \text{ Poise} \times 100 = \text{Centipoise } (\mu)$$

$$\text{Poise} \times 2.09 \times 10^{-4} = \text{slug}/\text{ft.-sec.}$$

$$= \text{pounds (force)-sec./sq.ft.}$$

$$\text{Poise} \times 0.10 = \text{pascal-sec.}$$

## D-2

### Physical Property Conversion Factors

Poise  $\times 0.0672 = \text{pounds (mass)}/(\text{ft.}\cdot\text{sec.})$   
 $= \text{poundal}\cdot\text{sec.}/\text{sq.ft.}$

Poise  $\times 0.10 = \text{Newton}\cdot\text{sec.}/\text{sq. meter}$

Centipoise  $\times 0.01 = \text{gm./cm.}\cdot\text{sec.}$

Centipoise  $\times 6.72 \times 10^{-4} = \text{pound}/\text{ft.}\cdot\text{sec.}$

Centipoise  $\times 2.4 = \text{pound}/\text{ft.}\cdot\text{hr.}$

Millipoise  $\times 1000 = \text{poise}$

Micropoise  $\times 1,000,000 = \text{poise}$

Slugs/ft. $\cdot$ sec.  $\times 47,900 = \text{centipoise}$

1 centistoke  $= 1.076 \times 10^{-5} \text{ ft.}^2/\text{sec.}$

1 centipoise (cp)  $= 0.01 \text{ gm./cm. sec.}$

Slugs/ft. $\cdot$ sec.  $\times 32.2 = \text{pounds (mass)}/\text{ft.}\cdot\text{sec.}$

Pounds/ft. $\cdot$ sec.  $\times 3600 = \text{lb.}/\text{ft.}\cdot\text{hr.}$

Pounds (mass)/ft. $\cdot$ sec.  $\times 1487 = \text{centipoise}$

Pounds (mass)/ft. $\cdot$ sec.  $\times 0.0311 = \text{slugs}/\text{ft.}\cdot\text{sec.}$

= pounds (force) $\cdot$ sec./sq.ft.

Viscosity of air @ 68° F.  $= 180.8 \times 10^{-4} \text{ poise}$

Viscosity of water @ 66° F.  $= 0.010087 \text{ poise}$

#### Viscosity (Kinematic)

Kinematic viscosity,

centistokes  $\times 1.076 \times 10^{-5} = \text{ft.}^2/\text{sec.}$

Kinematic viscosity, centistokes (\*)  $= \frac{\text{Dynamic viscosity, centipoise}}{\text{Fluid density, gm./cu.cm.}}$   
 $= \frac{\text{Centipoise}}{\text{Sp.Gr. of liquid relative to water at } 39.2^\circ \text{ F. (} 4^\circ \text{ C.)}}$

Centistokes  $\times 0.01 = \text{stokes, sq.cm./sec.}$

Centistokes  $\times 1.076 \times 10^{-5} = \text{sq.ft./sec.}$

Centistokes  $\times 0.01 = \text{Stokes, sq.cm./sec.}$

#### Thermal Conductivity (through a homogeneous material)

$\frac{\text{Btu (ft.)}}{(\text{sq.ft.}) (\text{°F.}) (\text{hr.})} \times 4.134 \times 10^{-3} = \frac{(\text{g.-cal.}) (\text{cm.})}{(\text{sq.cm.}) (\text{°C.}) (\text{sec.})}$   
 $\times 1.200 \times 10 = \frac{(\text{Btu}) (\text{in.})}{(\text{sq.ft.}) (\text{°F.}) (\text{hr.})}$

$\times 3.518 \times 10^{-3} = \frac{(\text{kilowatt hr.}) (\text{in.})}{(\text{sq.ft.}) (\text{°F.}) (\text{hr.})}$

$\frac{(\text{g.-cal.}) (\text{cm.})}{(\text{sq.cm.}) (\text{°C.}) (\text{hr.})} \times 8.063 \times 10^{-1} = \frac{\text{Btu (in.)}}{(\text{sq.ft.}) (\text{°F.}) (\text{hr.})}$   
 $\times 6.719 \times 10^{-4} = \frac{\text{Btu (ft.)}}{(\text{sq.ft.}) (\text{°F.}) (\text{hr.})}$

$\frac{(\text{g.-cal.}) (\text{cm.})}{(\text{sq.cm.}) (\text{°C.}) (\text{sec.})} \times 2.903 \times 10^3 = \frac{\text{Btu (in.)}}{(\text{sq.ft.}) (\text{°F.}) (\text{hr.})}$

$$\times 8.063 \times 10^{-1} = \frac{\text{Btu (in.)}}{(\text{sq.ft.}) (\text{°F.}) (\text{sec.})}$$

$$\times 8.506 \times 10^2 = \frac{(\text{joules}) (\text{in.})}{(\text{sq.ft.}) (\text{°F.}) (\text{sec.})}$$

#### Specific Gravity (Liquid)

$$s = \frac{\rho \text{ of liquid @ } 60^\circ \text{ F.}^*}{\rho \text{ of water @ } 60^\circ \text{ F.}^*}$$

\* or at other specified temperature

#### Oil

$$s \text{ at } 60^\circ \text{ F. / } 60^\circ \text{ F.} = \frac{141.5}{131.5 + \text{degrees API}}$$

#### Liquids Lighter Than Water

$$s \text{ @ } 60^\circ \text{ F. / } 60^\circ \text{ F.} = \frac{140}{130 + \text{degrees Baume'}}$$

#### Liquids Heavier Than Water

$$s \text{ @ } 60^\circ \text{ F. / } 60^\circ \text{ F.} = \frac{145}{145 - \text{degrees Baume'}}$$

#### Specific Gravity (Gases)

$$S_r = \frac{R \text{ of air}}{R \text{ of gas}} = \frac{53.3}{R \text{ of gas}}, \text{ where } R = \text{gas constant}$$

$$S_t = \frac{\text{mol. wt. (gas)}}{\text{mol. wt. (air)}} = \frac{\text{mol. wt. (gas)}}{29}$$

#### Density, Liquid $\rho$

Density liquid,  $\rho = (62.3 \text{ lb./cu. ft. water}) (\text{Sp. Gr. liquid}),$   
 $\text{pounds / cu. ft.}$

#### Metric

1 gram = 10 decigrams  
= 100 centigrams  
= 1,000 milligrams  
= 1,000,000 microgram  
= 0.001 kilogram  
=  $10^{-6}$  megagram

1 liter = 10 deciliters = 1.0567 liquid quarts

10 liters = 1 dekaliter = 2.6417 liquid gallons

10 dekaliters = 1 hectoliter = 2.8375 U. S. bushels

## D-2

### Physical Property Conversion Factors

1 meter = 10 decimeters = 39.37 inches  
   = 100 centimeters  
   = 1,000 millimeters  
   = 1,000,000 microns = 1,000,000 micrometers  
   = 1/1,000 kilometer  
   =  $10^{10}$  Angstrom units

10 millimeters = 1 centimeter = 0.3937 inches  
 10 centimeters = 1 decimeter = 3.937 inches  
 25.4 millimeters = 1 inch

Kg-cal/(hr.) (sq. m.) ( $^{\circ}$ C.)  $\times 0.2048$   
   = Btu/(hr.) (sq. ft.) ( $^{\circ}$ F.)  
   G-cal/(sec.) (sq. cm.) ( $^{\circ}$ C.)  $\times 7,380$   
   = Btu/(hr.) (sq. ft.) ( $^{\circ}$ F.)  
   Watts/(sq. in.) ( $^{\circ}$ F.)  $\times 490$  = Btu/(hr.) (sq. ft.) ( $^{\circ}$ F.)

#### Specific Heat

$$\begin{aligned} \frac{(\text{gram-cal.})}{(\text{gram}) (\text{ }^{\circ}\text{C.})} \times 1.8 &= \frac{\text{Btu}}{(\text{pound}) (\text{ }^{\circ}\text{C.})} \\ &\times 1.0 = \frac{\text{Btu}}{(\text{pound}) (\text{ }^{\circ}\text{F.})} \\ &\times 4.186 = \frac{\text{joules}}{(\text{gram}) (\text{ }^{\circ}\text{C.})} \\ &\times 1055 = \frac{\text{joules}}{(\text{pound}) (\text{ }^{\circ}\text{F.})} \\ &\times 1.163 \times 10^{-3} = \frac{\text{kilowatt-hours}}{(\text{kilogram}) (\text{ }^{\circ}\text{C.})} \\ &\times 2.930 \times 10^{-4} = \frac{\text{kilowatt-hours}}{(\text{pound}) (\text{ }^{\circ}\text{F.})} \end{aligned}$$

Specific heat of water at 1 atm. = 0.238 cal./gm. $^{\circ}$ C.  
 Btu/lb. -  $^{\circ}$ F.  $\times 0.2390$  = Btu/lb. -  $^{\circ}$ R

#### Heat Transfer Coefficient

$$\begin{aligned} \text{PCU}/(\text{hr.})(\text{sq. ft.}) (\text{ }^{\circ}\text{C.}) \times 1.0 \\ = \text{Btu}/(\text{hr.})(\text{sq. ft.}) (\text{ }^{\circ}\text{F.}) \end{aligned}$$

#### Energy Units

$$\begin{aligned} \text{Pound-Centigrade-Unit(PCU)} &\times 1.8 = \text{Btu} \\ &\times 0.45359 = \text{calorie} \\ &\times 1400.4 = \text{ft.-lb.} \\ &\times 0.0005276 = \\ &\quad \text{kilowatt-hr.} \\ &\times 1899.36 = \text{joules} \end{aligned}$$

$$\begin{aligned} \text{Calories} &\times 3.9683 = \text{Btu} \\ &\times 3091.36 = \text{ft.-lb.} \\ &\times 0.001559 = \text{horsepower-hr.} \\ &\times 0.001163 = \text{kilowatt-hr.} \\ &\times 4187.37 = \text{joules} \end{aligned}$$

#### Pressure

$$\begin{aligned} 1 \text{ mm Hg} &= 1,333 \text{ dynes/sq. cm.} \\ 750 \text{ mm Hg} &= 10 \text{ dynes/sq. cm.} = 1 \text{ megabar @ } ^{\circ}\text{C.} \\ \text{and } g &= 980.6 \end{aligned}$$

**D-3**  
**Synchronous Speeds**

$$\text{Synchronous Speed} = \frac{\text{Frequency} \times 120}{\text{No. of Poles}}$$

Poles	FREQUENCY			Poles	FREQUENCY	
	60 cycle	50 cycle	25 cycle		60 cycle	50 cycle
2	3600	3000	1500	42	171.4	142.9
4	1800	1500	750	44	163.6	136.4
6	1200	1000	500	46	156.3	130.4
8	900	750	375	48	150	125
10	720	600	300	50	144	120
12	600	500	250	52	138.5	113.4
14	514.3	428.6	214.3	54	133.3	111.1
16	450	375	187.5	56	128.6	107.1
18	400	333.3	166.7	58	124.1	103.3
20	360	300	150	60	120	100
22	327.2	272.7	136.4	62	116.3	96.8
24	300	250	125	64	112.5	93.7
26	276.9	230.4	115.4	66	109.1	90.9
28	257.1	214.3	107.1	68	105.9	88.2
30	240	200	100	70	102.9	85.7
32	225	187.5	93.7	72	100	83.3
34	211.8	176.5	88.2	74	97.3	81.1
36	200	166.7	83.3	76	94.7	78.9
38	189.5	157.9	78.9	78	92.3	76.9
40	180	150	75	80	90	75

Courtesy Ingersoll-Rand Co.

## Conversion Factors

Units of Length	Multiply units in left column by proper factor below							
	in.	ft.	yd.	miile	mm.	cm.	m.	km.
1 inch	1	0.0833	0.0278	—	25.40	2.540	0.0254	—
1 foot	12	1	0.3333	—	304.8	30.48	0.3048	—
1 yard	36	3	1	—	914.4	91.44	0.9144	—
1 mile	—	5280	1760	1	—	—	1609.3	1.609
1 millimeter	0.0394	0.0033	—	—	1	0.100	0.001	—
1 centimeter	0.3937	0.0328	0.0109	—	10	1	0.01	—
1 meter	39.37	3.281	1.094	—	1000	100	1	0.001
1 kilometer	—	3281	1094	0.6214	—	—	1000	1

(1 micron = 0.001 millimeter)

Courtesy Ingersoll-Rand Co.

**D-4**  
**(Continued). Conversion Factors**

Units of Weight	Multiply units in left column by proper factor below						
	grain	oz.	lb.	tow	stone	kg.	metric ton
1 grain	1	—	—	—	0.0648	—	—
1 ounce	437.5	1	0.0625	—	28.35	0.0283	—
1 pound	7000	16	1	0.0005	453.6	0.4536	—
1 ton	—	32,000	2000	1	—	907.2	0.9072
1 gram	15.43	0.0353	—	—	1	0.001	—
1 kilogram	—	35.27	2.205	—	1000	1	0.001
1 metric ton	—	35,274	2205	1,1023	—	1000	1

Units of Density	Multiply units in left column by proper factor below				
	lb./cu. in.	lb./cu. ft.	lb./gal.	g./cu. cm.	g./liter
1 pound/cu. in.	1	1728	231.0	27.68	27,680
1 pound/cu. ft.	—	1	0.1337	0.0160	16.019
1 pound/gal.	0.00433	7.481	1	0.1198	119.83
1 gram/cu. cm.	0.0353	62.43	8.345	1	1000.0
1 gram/liter	—	0.0624	0.00835	0.001	1

Units of Area	Multiply units in left column by proper factor below						
	sq. in.	sq. ft.	acre	sq. mile	sq. cm.	sq. m.	hectare
1 sq. inch	1	0.0069	—	—	6,452	—	—
1 sq. foot	144	1	—	—	929.0	0.0929	—
1 acre	—	43,560	1	0.0016	—	4047	0.4047
1 sq. mile	—	—	640	1	—	—	259.0
1 sq. centimeter	0.1550	—	—	—	1	0.0001	—
1 sq. meter	1550	10.76	—	—	10,000	1	—
1 hectare	—	—	2,471	—	—	10,000	1

Units of Volume	Multiply units in left column by proper factor below							
	cu. in.	cu. ft.	cu. yd.	cu. cm.	cu. meter	liter	U.S. gal.	Imp. gal.
1 cu. inch	1	—	—	16,387	—	0.0164	—	—
1 cu. foot	1728	1	0.0370	28,317	0.0283	28.32	7.481	6.229
1 cu. yard	46,656	27	1	—	0.7646	764.5	202.0	168.2
1 cu. centimeter	0.0610	—	—	1	—	0.0010	—	—
1 cu. meter	61,023	35.31	1.308	1,000,000	1	999.97	264.2	220.0
1 liter	61.023	0.0353	—	1000.023	0.0010	1	0.2642	0.2200
1 U.S. gallon	231	0.1337	—	3785.4	—	3.785	1	0.8327
1 imperial gallon	277.4	0.1605	—	4546.1	—	4.546	1.201	1

**D-4**  
**(Continued). Conversion Factors**

Units of Pressure	Multiply units in left column by proper factor below						
	lb/sq. in.	lb/sq. ft.	in. abs.	kg/cm <sup>2</sup>	mm Hg at 32°F	in. Hg at 32°F	ft. water at 32°F
1 pound/sq. in.	1	144	—	0.0703	31.713	2.0329	2.307
1 pound/sq. ft.	0.00694	1	—	—	0.3691	0.01414	0.01692
1 Intern. atmosphere	14.696	2116.2	1	1.0133	760	29.921	33.93
1 kilogram/sq. cm.	14.223	2048.1	0.9678	1	735.58	28.958	32.81
1 millimeter-mercury— 1 torr (Dynes/cm <sup>2</sup> )—	0.0193	2.733	—	—	1	0.0394	0.0446
1 inch mercury	0.4912	70.78	0.0334	0.0345	25.400	1	1.123
1 foot water	0.4335	62.42	—	0.0305	22.418	0.0028	1

Units of Energy	Multiply units in left column by proper factor below					
	hp-hr.	hp	ft-lb.	Joule	kw-hr.	kg-km
1 foot-pounds	1	0.001285	0.3240	1.3395	—	—
1 Btu	778.2	1	252.15	1054.9	—	—
1 gram calorie	3.0869	0.003968	1	4.1833	—	—
1 int. Joule	0.7377	0.000948	0.2390	1	—	—
1 int. kilowatt-hour	2,655,856	3412.8	860,583	—	1	1.3412
1 horsepower-hour	1,980,000	2544.5	641,817	—	0.7456	1

Units of Specific Energy	Multiply units in left column by proper factor below				
	absolute Joule/gm	int. Joule/gm	cal/g	int. cal/g	Btu/lb.
1 absolute Joule/gm	1	0.99944	0.23901	0.23885	0.42933
1 int. Joule/gm	1.000165	1	0.23904	0.23882	0.43000
1 calorie/gm	4.1840	4.1833	1	0.99935	1.7988
1 int. calorie/gm	4.1857	4.1850	1.00055	1	1.8000
1 Btu/lb	2.3260	2.3258	0.53582	0.53556	1

Units of Power (rate of energy use)	Multiply units in left column by proper factor below								
	hp	watt	kw	Btu/min.	hp/in.	Btu/sec.	Btu/min.	in. cal/sec.	metric hp
1 horsepower	1	745.7	0.7475	42.41	2544.3	559	33,000	175.2	1.014
1 watt	—	1	0.001	0.0569	3.413	0.7376	44.25	0.2390	0.00136
1 kilowatt	1.3410	1000	1	56.88	3412.8	737.8	44,254	239.0	1.360
1 Btu per minute	—	—	—	1	60	12.97	778.2	4.203	0.0239
1 metric hp	0.9863	735.5	0.7355	41.83	2509.8	542.5	32,530	175.7	1

Units of Refrigeration	Multiply units in left column by factor below					
	Btu(IT)/min.	Btu(IT)/hr.	kg cal/hr.	ton (U.S.) com	ton (BRIT.) com	horsepower
1 ton (U.S.) com	200	12,000	3025.8	1	0.8965	3025.8
1 ton (Brit.) com	223.08	13,385	3375.2	1.1154	1	3375.2
1 frigorie/hr.	0.06609	3.9657	1	0.0003305	0.0002963	1

Note.—Btu is International Steam Table Btu(IT).

1 frigorie = 1 kg cal (Met IT).

## D-5

### Temperature Conversion

**NOTE:** The center column of numbers in boldface refers to the temperature in degrees, either Centigrade or Fahrenheit, which it is desired to convert into the other scale. If converting from Fahrenheit to Centigrade degrees, the equivalent temperature will be found in the left column; while if converting from degrees Centigrade to degrees Fahrenheit, the answer will be found in the column on the right.

Centigrade	Fahrenheit	Centigrade	Fahrenheit	Centigrade	Fahrenheit	Centigrade	Fahrenheit	
-273.17	-459.7	-20.6	-5	23.0	11.1	52	125.6	
-268	-458	-17.8	0	32.0	11.7	53	127.4	
-252	-444	-14.4	4	33.8	12.2	54	129.2	
-237	-428	-17.2	1	33.8	12.8	55	131.0	
-231	-420	-16.7	2	35.6	13.3	56	132.8	
-246	-410	-16.1	3	37.4				
-240	-400	-15.6	4	39.2	13.9	57	134.6	
-234	-390	-15.0	5	41.0	14.4	58	136.4	
-229	-380	-14.4	6	42.8	15.0	59	138.2	
-223	-370	-13.9	7	44.6	15.6	60	140.0	
-218	-360	-13.3	8	46.4	16.1	61	141.8	
-212	-350	-12.8	9	48.2	16.7	62	143.6	
-207	-340	-12.2	10	50.0	17.2	63	145.4	
-201	-330	-11.7	11	51.8	17.8	64	147.2	
-196	-320	-11.1	12	53.6	18.3	65	149.0	
-190	-310	-10.6	13	55.4	18.9	66	150.8	
-184	-300	-10.0	14	57.2	19.4	67	152.6	
-179	-290	-9.4	15	59.0	20.0	68	154.4	
-173	-280	-8.9	16	60.8	20.6	69	156.2	
-167	-273	-459.4	-8.3	17	62.6	21.1	70	158.0
-161	-270	-454	-7.8	18	64.4	21.7	71	159.8
-155	-260	-436	-7.2	19	66.2	22.2	72	161.6
-157	-250	-418	-6.7	20	68.0	22.8	73	163.4
-151	-240	-400	-6.1	21	69.8	23.3	74	165.2
-145	-230	-382	-5.6	22	71.6	23.9	75	167.0
-140	-220	-364	-4.4	23	73.4	24.4	76	168.8
-134	-210	-346	-3.4	24	75.2	25.0	77	170.6
-129	-200	-328	-3.9	25	77.0	25.6	78	172.4
-123	-190	-310	-3.3	26	78.8	26.1	79	174.2
-118	-180	-292	-2.8	27	80.6	26.7	80	176.0
-112	-170	-274	-2.2	28	82.4	27.2	81	177.8
-107	-160	-256	-1.7	29	84.2	27.8	82	179.6
-101	-150	-238	-1.1	30	86.0	28.3	83	181.4
-96	-140	-220	-0.6	31	87.8	28.9	84	183.2
-90	-130	-202	0.0	32	89.6	29.4	85	185.0
-84	-120	-184	0.6	33	91.4	30.0	86	186.8
-79	-110	-166	1.1	34	93.2	30.6	87	188.6
-73.3	-100	-148.0	1.7	35	95.0	31.1	88	190.4
-67.8	-90	-130.0	2.2	36	96.8	31.7	89	192.2
-62.2	-80	-112.0	2.8	37	98.6	32.2	90	194.0
-59.4	-75	-103.0	3.3	38	100.4	32.8	91	195.8
-56.7	-70	-94.0	3.9	39	102.2	33.3	92	197.6
-53.9	-65	-85.0	4.4	40	104.0	33.9	93	199.4
-51.1	-60	-76.0	5.0	41	105.8	34.4	94	201.2
-48.3	-55	-67.0	5.6	42	107.6	35.0	95	203.0
-45.6	-50	-58.0	6.1	43	109.4	35.6	96	204.8
-42.8	-45	-49.0	6.7	44	111.2	36.1	97	206.6
-40.0	-40	-40.0	7.2	45	113.0	36.7	98	208.4
-37.2	-35	-31.0	7.8	46	114.8	37.2	99	210.2
-34.4	-30	-22.0	8.3	47	116.6	37.8	100	212.0
-31.7	-25	-13.0	8.9	48	118.4	40.6	105	221
-28.9	-20	-4.0	9.4	49	120.2	46.1	115	239
-26.1	-15	5.0	10.0	50	122.0	48.9	123	248
-23.3	-10	4.0	10.6	51	123.8	51.7	125	257

The formulas at the right may also be used for converting Centigrade or Fahrenheit degrees into the other scales.

$$\text{Degrees Cent., } ^\circ\text{C} = \frac{5}{9} (\text{ }^\circ\text{F} + 40) - 40$$

$$= \frac{5}{9} (\text{ }^\circ\text{F} - 32)$$

$$\text{Degrees Kelvin, } ^\circ\text{K} = ^\circ\text{C} + 273.2$$

$$\text{Degrees Fahr., } ^\circ\text{F} = \frac{9}{5} (^\circ\text{C} + 40) - 40$$

$$= \frac{9}{5} \text{ C} + 32$$

$$\text{Degrees Rankine, } ^\circ\text{R} = ^\circ\text{F} + 459.7$$

**D-6**  
**Altitude and Atmospheric Pressures**

Altitude above Sea Level			Tempera- ture <sup>**</sup>		Barometer <sup>*</sup>		Atmospheric Pressure	
Feet <sup>#</sup>	Miles	Meters <sup>#</sup>	°F	°C	Indic. Hg Abs.	Int. Hg Abs.	PSIA	Kg/sq cm Abs.
-5000		-1526	77	25	35.38	903.7	17.48	1.229
-4500		-1373	75	24	35.00	889.0	17.19	1.209
-4000		-1220	73	23	34.42	874.3	16.90	1.188
-3500		-1068	71	22	33.84	859.5	16.62	1.169
-3000		-915	70	21	33.27	845.1	16.34	1.149
-2500		-763	68	20	32.70	830.6	16.06	1.129
-2000		-610	66	19	32.14	816.4	15.78	1.109
-1500		-458	64	18	31.58	802.1	15.51	1.091
-1000		-305	63	17	31.02	787.9	15.23	1.071
-500		-153	61	16	30.47	773.9	14.96	1.052
0		0	59	15	29.92	760.0	14.696	1.0333
500		153	57	14	29.38	744.3	14.43	1.013
1000		305	55	13	28.86	733.0	14.16	.996
1500		458	54	12	28.33	719.6	13.91	.978
2000		610	52	11	27.82	706.6	13.66	.960
2500		763	50	10	27.32	693.9	13.41	.943
3000		915	48	9	26.82	681.2	13.17	.926
3500		1068	47	8	26.33	668.8	12.93	.909
4000		1220	45	7	25.84	656.3	12.69	.892
4500		1373	43	6	25.37	644.4	12.46	.876
5000	0.95	1526	41	5	24.90	632.5	12.23	.860
6000	1.1	1831	38	3	23.99	609.3	11.78	.828
7000	1.3	2136	34	1	23.10	586.7	11.34	.797
8000	1.5	2441	31	-1	22.23	564.6	10.91	.767
9000	1.7	2746	27	-3	21.39	543.3	10.50	.738
10,000	1.9	3050	23	-5	20.58	522.7	10.10	.710
15,000	2.8	4577	6	-14	16.89	429.0	8.29	.583
20,000	3.8	6102	-12	-24	13.76	349.5	6.76	.475
25,000	4.7	7628	-30	-34	11.12	282.4	5.46	.384
30,000	5.7	9153	-48	-44	8.903	226.1	4.37	.307
35,000	6.6	10,679	-66		7.040	179.3	3.47	.244
40,000	7.6	12,204	-70	-57	5.558	141.2	2.73	.192
45,000	8.5	13,730	-70	-57	4.375	111.1	2.15	.151
50,000	9.3	15,255	-70	-57	3.444	87.5	1.69	.119
55,000	10.4	16,781	-70	-57	2.712	68.9	1.33	.0935
60,000	11.4	18,306	-70	-57	2.135	54.2	1.03	.0738
70,000	13.3	21,357	-67	-55	1.325	33.7	.651	.0458
80,000	15.2	24,408	-62	-52	78.273 <sup>-1</sup>	21.0	.406	.0285
90,000	17.1	27,459	-57	-59	5.200 <sup>-1</sup>	13.2	.235	.0179
100,000	18.9	30,510	-51	-46	3.290 <sup>-1</sup>	8.36	.162	.0114
120,000	22.8	36,612	-26	-46	1.358 <sup>-1</sup>	3.45		
140,000	26.6	42,714	4	-14	5.942 <sup>-2</sup>	1.51		
160,000	30.4	48,816	28	-2	2.746 <sup>-2</sup>	16.97 <sup>-1</sup>		
180,000	34.2	54,918	19	-7	1.284 <sup>-2</sup>	3.26 <sup>-1</sup>		
200,000	37.9	61,020	-3	-19	5.846 <sup>-3</sup>	1.48 <sup>-1</sup>		
220,000	41.7	67,122	-44	-42	2.523 <sup>-2</sup>	6.41 <sup>-2</sup>		
240,000	45.5	73,224	-66	-66	9.955 <sup>-4</sup>	2.53 <sup>-2</sup>		
260,000	49.3	79,326	-129	-90	3.512 <sup>-4</sup>	8.92 <sup>-3</sup>		
280,000	53.1	85,428	-135	-93	1.143 <sup>-4</sup>	3.67 <sup>-3</sup>		
300,000	56.9	91,530	-127	-88	3.737 <sup>-5</sup>	9.49 <sup>-4</sup>		
400,000	75.9	122,040	—	—	6.3 <sup>-7</sup>	1.60 <sup>-8</sup>		
500,000	94.8	152,550	—	—	1.4 <sup>-7</sup>	3.56 <sup>-4</sup>		
600,000	114	183,060	—	—	5.9 <sup>-8</sup>	1.50 <sup>-4</sup>		
800,000	152	244,080	—	—	1.6 <sup>-8</sup>	4.06 <sup>-7</sup>		
1,000,000	189	305,100	—	—	5.1 <sup>-9</sup>	1.30 <sup>-7</sup>		
1,200,000	228	366,120	—	—	2.0 <sup>-9</sup>	5.08 <sup>-8</sup>		
1,400,000	266	427,140	—	—	8.2 <sup>-10</sup>	2.08 <sup>-4</sup>		
1,600,000	304	488,160	—	—	3.8 <sup>-10</sup>	9.65 <sup>-9</sup>		
1,800,000	342	549,180	—	—	1.3 <sup>-10</sup>	4.57 <sup>-9</sup>		
2,000,000	379	610,200	—	—	9.2 <sup>-11</sup>	2.34 <sup>-7</sup>		

Data from NASA Standard Atmosphere (1962).

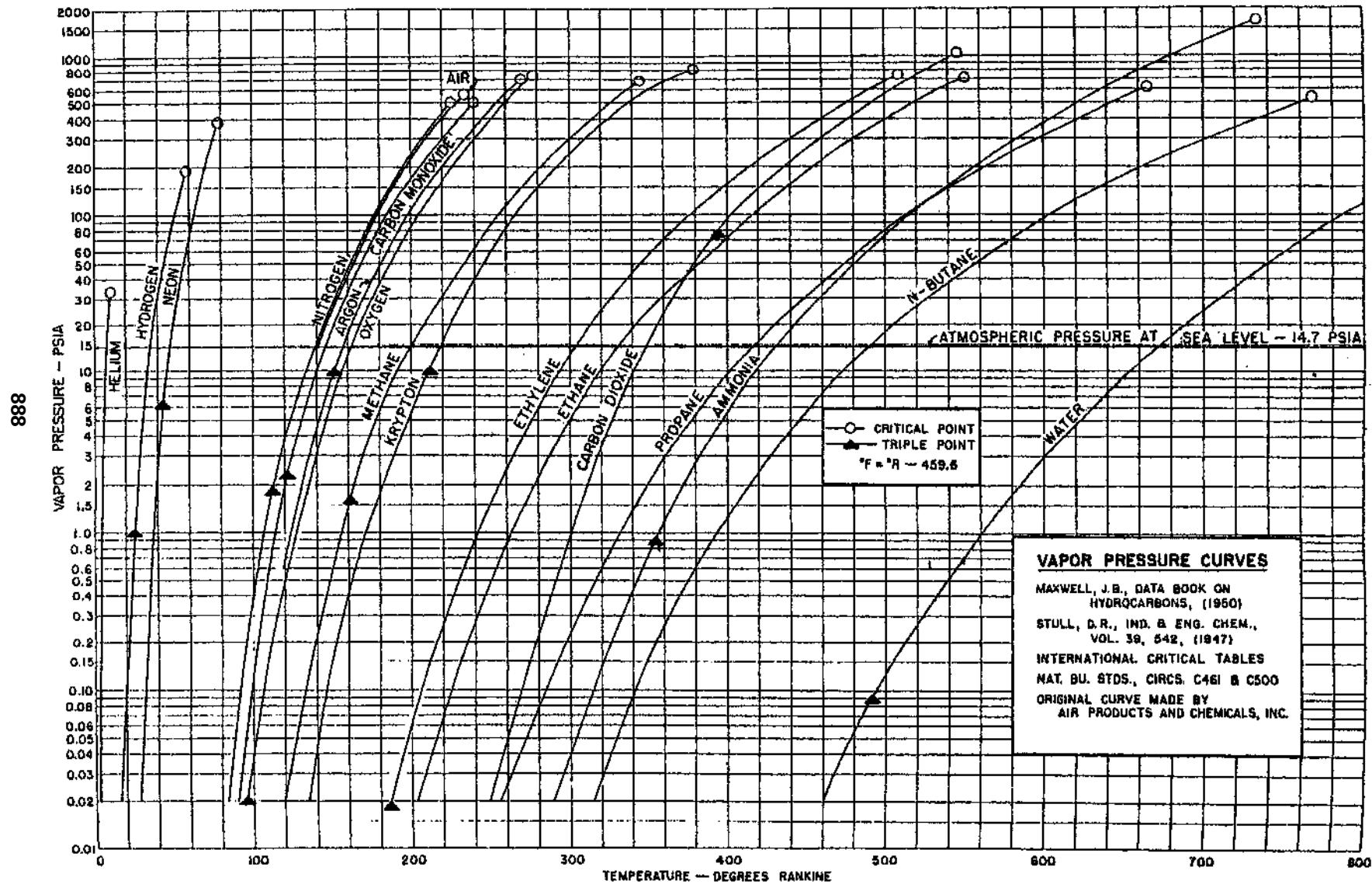
\*Temperature and barometer are approximate for negative altitudes.

\*\*Temperatures are average existing at 40° latitude and are rounded to even numbers.

†Negative exponent shows number of spaces the decimal point must be moved to the left.

Courtesy Ingersoll-Rand Co.

D-7  
Vapor Pressure Curves. (Courtesy Ingersoll-Rand Co.)



MULTIPLY GIVEN NUMBER OF

BY FACTOR TO OBTAIN —————

GIVEN	lb/in <sup>2</sup>	in H <sub>2</sub> O (at +32°F)	cm H <sub>2</sub> O (at +4°C)	in Hg (at +32°F)	mm Hg (Torr) (at 0°C)	dynes/cm <sup>2</sup> (1 μ bar)	newton/m <sup>2</sup> (PASCAL)	kgm/cm <sup>2</sup>	bar	atm. (A <sub>n</sub> )	lb/ft <sup>2</sup>	ft H <sub>2</sub> O (at +32°F)
lb/in <sup>2</sup>	1.0000	2.7680x10 <sup>1</sup>	7.0308x10 <sup>1</sup>	2.0380	5.1715x10 <sup>1</sup>	6.8948x10 <sup>4</sup>	6.8948x10 <sup>3</sup>	7.0308x10 <sup>-2</sup>	6.8847x10 <sup>-2</sup>	6.8045x10 <sup>-2</sup>	1.4400x10 <sup>2</sup>	2.3087
in H <sub>2</sub> O (at +32°F)	3.6127x10 <sup>-2</sup>	1.0000	2.5400	7.3554x10 <sup>-2</sup>	1.6683	2.4908x10 <sup>2</sup>	2.4908x10 <sup>3</sup>	2.6399x10 <sup>-3</sup>	2.4908x10 <sup>-3</sup>	2.4582x10 <sup>-3</sup>	6.2022	8.3333x10 <sup>-2</sup>
cm H <sub>2</sub> O (at +4°C)	1.4223x10 <sup>-2</sup>	0.3937	1.0000	2.8958x10 <sup>-2</sup>	0.7355	9.8084x10 <sup>2</sup>	9.8084x10 <sup>1</sup>	9.9997x10 <sup>-4</sup>	9.8084x10 <sup>-4</sup>	9.6761x10 <sup>-4</sup>	2.0481	3.2808x10 <sup>-2</sup>
in Hg (at +32°F)	4.9116x10 <sup>-1</sup>	1.3586x10 <sup>1</sup>	3.4532x10 <sup>1</sup>	1.0000	2.5400x10 <sup>1</sup>	3.3884x10 <sup>4</sup>	3.3884x10 <sup>3</sup>	3.4832x10 <sup>-2</sup>	3.3884x10 <sup>-2</sup>	3.3421x10 <sup>-2</sup>	7.0727x10 <sup>1</sup>	1.1330
mm Hg (Torr) (at 0°C)	1.9337x10 <sup>-1</sup>	5.3525x10 <sup>-1</sup>	1.3685	3.9370x10 <sup>-2</sup>	1.0000	1.3332x10 <sup>3</sup>	1.3332x10 <sup>2</sup>	1.3585x10 <sup>-3</sup>	1.3332x10 <sup>-3</sup>	1.3156x10 <sup>-3</sup>	2.7845	4.4805x10 <sup>-2</sup>
dynes/cm <sup>2</sup> (1 μ bar)	1.4604x10 <sup>-3</sup>	4.0147x10 <sup>-4</sup>	1.0197x10 <sup>-3</sup>	2.9530x10 <sup>-4</sup>	7.5006x10 <sup>-4</sup>	1.0000	1.0000x10 <sup>-1</sup>	1.0197x10 <sup>-6</sup>	1.0000x10 <sup>-6</sup>	9.8692x10 <sup>-3</sup>	2.0886x10 <sup>-3</sup>	3.3456x10 <sup>-6</sup>
newton/m <sup>2</sup> (PASCAL)	1.4504x10 <sup>-4</sup>	4.0147x10 <sup>-5</sup>	1.0197x10 <sup>-4</sup>	2.9530x10 <sup>-5</sup>	7.5006x10 <sup>-5</sup>	1.0000x10 <sup>1</sup>	1.0000	1.0197x10 <sup>-8</sup>	1.0000x10 <sup>-8</sup>	9.8692x10 <sup>-5</sup>	2.0886x10 <sup>-5</sup>	3.3456x10 <sup>-4</sup>
kgm/cm <sup>2</sup>	1.4224x10 <sup>1</sup>	3.9371x10 <sup>1</sup>	1.00003x10 <sup>2</sup>	2.8958x10 <sup>1</sup>	7.3556x10 <sup>3</sup>	9.8080x10 <sup>4</sup>	9.8080x10 <sup>3</sup>	1.0000	9.8080x10 <sup>-1</sup>	9.678x10 <sup>-1</sup>	2.0482x10 <sup>3</sup>	3.2809x10 <sup>1</sup>
bar	1.4504x10 <sup>1</sup>	4.0147x10 <sup>2</sup>	1.0197x10 <sup>2</sup>	2.9530x10 <sup>1</sup>	7.5006x10 <sup>2</sup>	1.0000x10 <sup>6</sup>	1.0000x10 <sup>5</sup>	1.0197	1.0000	9.8692x10 <sup>-1</sup>	2.0885x10 <sup>3</sup>	3.3456x10 <sup>4</sup>
atm. (A <sub>n</sub> )	1.4696x10 <sup>1</sup>	4.0679x10 <sup>2</sup>	1.0333x10 <sup>3</sup>	2.9921x10 <sup>1</sup>	7.6000x10 <sup>2</sup>	1.0133x10 <sup>6</sup>	1.0133x10 <sup>5</sup>	1.0332	1.0133	1.0000	2.1182x10 <sup>3</sup>	3.3900x10 <sup>1</sup>
lb/ft <sup>2</sup>	8.9446x10 <sup>-3</sup>	1.9223x10 <sup>-1</sup>	4.882x10 <sup>-1</sup>	1.4139x10 <sup>-2</sup>	3.691x10 <sup>-1</sup>	4.7880x10 <sup>3</sup>	4.7880x10 <sup>1</sup>	4.8824x10 <sup>-4</sup>	4.7880x10 <sup>-4</sup>	4.7264x10 <sup>-4</sup>	1.0000	1.6019x10 <sup>-3</sup>
ft H <sub>2</sub> O (at +32°F)	4.3352x10 <sup>-1</sup>	1.2000x10 <sup>1</sup>	3.0480x10 <sup>1</sup>	8.826x10 <sup>-1</sup>	2.2419x10 <sup>1</sup>	2.9890x10 <sup>4</sup>	2.9890x10 <sup>3</sup>	3.0479x10 <sup>-2</sup>	2.9890x10 <sup>-2</sup>	2.9499x10 <sup>-2</sup>	6.2427x10 <sup>1</sup>	1.0000

**D-8**  
Pressure Conversion Chart

**D-9**  
**Vacuum Conversion**

Torr	Absolute Pressure		Inches Hg (Aba.)	Psin	Vacuum* Inches Hg
	Microns Hg	MM Hg			
	762	30.00	14.74		—
	750	29.53	14.50		0.47
	700	27.56	13.54		2.44
	650	25.59	12.57		4.41
	600	23.62	11.60		6.38
	550	21.65	10.64		8.35
	500	19.68	9.67		10.32
	450	17.72	8.70		12.28
	400	15.75	7.74		14.25
	350	13.78	6.77		16.22
	300	11.81	5.80		18.19
	250	9.84	4.84		20.16
	200	7.84	3.87		22.13
	150	5.91	2.900		24.09
	100	3.94	1.934		26.06
	50	1.97	0.967		28.03
	40	1.57	0.774		28.43
	30	1.181	0.580		28.82
	20	0.787	0.3868		
	10	0.394	0.1934		
	5	0.197	0.0967		
	4	0.158	0.0774		
	3	0.1181	0.0580		
	2	0.0787	0.0387		
1.0	1000	1	0.0992	0.0193	
0.5	500	0.50	0.0197		Low Vacuum
$1 \times 10^{-1}$	100	0.10	0.0039		
$5 \times 10^{-2}$	50	0.050			
$1 \times 10^{-3}$	10	0.010			
$5 \times 10^{-4}$	5	0.005			
$1 \times 10^{-5}$	1	0.001			
$1 \times 10^{-6}$	to				High Vacuum
$1 \times 10^{-7}$	to				Very High Vac.
$1 \times 10^{-8}$	to				
$1 \times 10^{-9}$					Ultra High Vac.
and beyond					

\*Refers to 30" Barometer

**Conversion Factors:**

1 millimeter = 1000 microns  
1 Torr = 1 mm Hg Abs.

1 inch Hg = 25.4 mm Hg  
1 atmosphere = 14.7 pounds per sq. in. = 760 mm Hg = 29.92 in. Hg

**D-10**  
Decimal and Millimeter Equivalent of Fractions

Inches		1000- Millimeters	Inches		1000- Millimeters
Fractions	Decimals		Fractions	Decimals	
$\frac{1}{16}$	.06250	1.588	$\frac{23}{64}$	.359375	13.007
$\frac{1}{8}$	.12500	3.175	$\frac{25}{64}$	.390625	13.406
$\frac{3}{16}$	.18750	4.750	$\frac{27}{64}$	.421875	13.795
$\frac{1}{4}$	.25000	6.333	$\frac{29}{64}$	.453125	14.184
$\frac{5}{16}$	.31250	7.917	$\frac{31}{64}$	.484375	14.573
$\frac{3}{8}$	.37500	9.492	$\frac{33}{64}$	.515625	14.962
$\frac{7}{16}$	.43750	11.075	$\frac{35}{64}$	.546875	15.351
$\frac{1}{2}$	.50000	12.650	$\frac{37}{64}$	.578125	15.740
$\frac{9}{16}$	.56250	14.325	$\frac{39}{64}$	.609375	16.129
$\frac{5}{8}$	.62500	15.900	$\frac{41}{64}$	.640625	16.518
$\frac{11}{16}$	.68750	17.475	$\frac{43}{64}$	.671875	16.907
$\frac{3}{4}$	.75000	19.050	$\frac{45}{64}$	.703125	17.296
$\frac{13}{16}$	.81250	20.625	$\frac{47}{64}$	.734375	17.685
$\frac{7}{8}$	.87500	22.200	$\frac{49}{64}$	.765625	18.074
$\frac{15}{16}$	.93750	23.775	$\frac{51}{64}$	.806875	18.463
$\frac{1}{1}$	1.00000	25.350	$\frac{53}{64}$	.838125	18.852
$\frac{17}{16}$	1.06250	26.925	$\frac{55}{64}$	.869375	19.241
$\frac{9}{8}$	1.12500	28.500	$\frac{57}{64}$	.900625	19.630
$\frac{19}{16}$	1.18750	30.075	$\frac{59}{64}$	.931875	20.019
$\frac{11}{8}$	1.25000	31.650	$\frac{61}{64}$	.963125	20.408
$\frac{21}{16}$	1.31250	33.225	$\frac{63}{64}$	.994375	20.797
$\frac{13}{8}$	1.37500	34.800	$\frac{65}{64}$	1.025625	21.186
$\frac{23}{16}$	1.43750	36.375	$\frac{67}{64}$	1.056875	21.575
$\frac{15}{8}$	1.50000	37.950	$\frac{69}{64}$	1.088125	21.964
$\frac{25}{16}$	1.56250	39.525	$\frac{71}{64}$	1.119375	22.353
$\frac{17}{8}$	1.62500	41.100	$\frac{73}{64}$	1.150625	22.742
$\frac{27}{16}$	1.68750	42.675	$\frac{75}{64}$	1.181875	23.131
$\frac{19}{8}$	1.75000	44.250	$\frac{77}{64}$	1.213125	23.520
$\frac{29}{16}$	1.81250	45.825	$\frac{79}{64}$	1.244375	23.909
$\frac{21}{8}$	1.87500	47.400	$\frac{81}{64}$	1.275625	24.298
$\frac{31}{16}$	1.93750	48.975	$\frac{83}{64}$	1.306875	24.687
$\frac{23}{8}$	2.00000	50.550	$\frac{85}{64}$	1.338125	25.076
$\frac{33}{16}$	2.06250	52.125	$\frac{87}{64}$	1.369375	25.465
$\frac{25}{8}$	2.12500	53.700	$\frac{89}{64}$	1.400625	25.854
$\frac{35}{16}$	2.18750	55.275	$\frac{91}{64}$	1.431875	26.243
$\frac{27}{8}$	2.25000	56.850	$\frac{93}{64}$	1.463125	26.632
$\frac{37}{16}$	2.31250	58.425	$\frac{95}{64}$	1.494375	27.021
$\frac{29}{8}$	2.37500	60.000	$\frac{97}{64}$	1.525625	27.410
$\frac{39}{16}$	2.43750	61.575	$\frac{99}{64}$	1.556875	27.799
$\frac{31}{8}$	2.50000	63.150	$\frac{101}{64}$	1.588125	28.188
$\frac{41}{16}$	2.56250	64.725	$\frac{103}{64}$	1.619375	28.577
$\frac{33}{8}$	2.62500	66.300	$\frac{105}{64}$	1.650625	28.966
$\frac{45}{16}$	2.68750	67.875	$\frac{107}{64}$	1.681875	29.355
$\frac{35}{8}$	2.75000	69.450	$\frac{109}{64}$	1.713125	29.744
$\frac{47}{16}$	2.81250	71.025	$\frac{111}{64}$	1.744375	30.133
$\frac{37}{8}$	2.87500	72.600	$\frac{113}{64}$	1.775625	30.522
$\frac{49}{16}$	2.93750	74.175	$\frac{115}{64}$	1.806875	30.911
$\frac{39}{8}$	3.00000	75.750	$\frac{117}{64}$	1.838125	31.299
$\frac{51}{16}$	3.06250	77.325	$\frac{119}{64}$	1.869375	31.688
$\frac{41}{8}$	3.12500	78.900	$\frac{121}{64}$	1.900625	32.077
$\frac{53}{16}$	3.18750	80.475	$\frac{123}{64}$	1.931875	32.466
$\frac{43}{8}$	3.25000	82.050	$\frac{125}{64}$	1.963125	32.855
$\frac{55}{16}$	3.31250	83.625	$\frac{127}{64}$	1.994375	33.244
$\frac{45}{8}$	3.37500	85.200	$\frac{129}{64}$	2.025625	33.633
$\frac{57}{16}$	3.43750	86.775	$\frac{131}{64}$	2.056875	34.022
$\frac{47}{8}$	3.50000	88.350	$\frac{133}{64}$	2.088125	34.411
$\frac{59}{16}$	3.56250	89.925	$\frac{135}{64}$	2.119375	34.799
$\frac{49}{8}$	3.62500	91.500	$\frac{137}{64}$	2.150625	35.188
$\frac{61}{16}$	3.68750	93.075	$\frac{139}{64}$	2.181875	35.577
$\frac{51}{8}$	3.75000	94.650	$\frac{141}{64}$	2.213125	35.966
$\frac{63}{16}$	3.81250	96.225	$\frac{143}{64}$	2.244375	36.355
$\frac{53}{8}$	3.87500	97.800	$\frac{145}{64}$	2.275625	36.744
$\frac{65}{16}$	3.93750	99.375	$\frac{147}{64}$	2.306875	37.133
$\frac{55}{8}$	4.00000	100.950	$\frac{149}{64}$	2.338125	37.522
$\frac{67}{16}$	4.06250	102.525	$\frac{151}{64}$	2.369375	37.911
$\frac{57}{8}$	4.12500	104.100	$\frac{153}{64}$	2.400625	38.299
$\frac{69}{16}$	4.18750	105.675	$\frac{155}{64}$	2.431875	38.688
$\frac{59}{8}$	4.25000	107.250	$\frac{157}{64}$	2.463125	39.077
$\frac{71}{16}$	4.31250	108.825	$\frac{159}{64}$	2.494375	39.466
$\frac{61}{8}$	4.37500	110.400	$\frac{161}{64}$	2.525625	39.855
$\frac{73}{16}$	4.43750	111.975	$\frac{163}{64}$	2.556875	40.244
$\frac{63}{8}$	4.50000	113.550	$\frac{165}{64}$	2.588125	40.633
$\frac{75}{16}$	4.56250	115.125	$\frac{167}{64}$	2.619375	41.022
$\frac{65}{8}$	4.62500	116.700	$\frac{169}{64}$	2.650625	41.411
$\frac{77}{16}$	4.68750	118.275	$\frac{171}{64}$	2.681875	41.799
$\frac{67}{8}$	4.75000	119.850	$\frac{173}{64}$	2.713125	42.188
$\frac{79}{16}$	4.81250	121.425	$\frac{175}{64}$	2.744375	42.577
$\frac{69}{8}$	4.87500	122.000	$\frac{177}{64}$	2.775625	42.966

**D-11**  
Particle Size Measurement

Meshes/Lineal Inch US and ASTM		Actual Opening		Meshes/Lineal Inch US and ASTM		Actual Opening	
Std. Sieve No.	Inches	Microns	Std. Sieve No.	Inches	Microns		
10	.0787	2000	170	.0035	88		
12	.0661 1/6	1680	200	.0029	74		
14	.0555	1410	230	.0026	65		
16	.0469 3/64	1190	270	.0024	62		
18	.0394	1000	310	.0021	53		
20	.0331 1/32	840	350	.0020	50		
25	.0280	710	390	.0017	44		
30	.0232	590	430	.0016	40		
35	.0197 1/64	500	470	.00142	36		
40	.0165	420	510	.00118	30		
45	.0138	350	550	.00099	25		
50	.0117	297	625	.00079	20		
60	.0098	250	750	.00059	15		
70	.0083	210	1,250	.000394	10		
80	.0070	177	1,750	.000315	8		
100	.0059	149	2,500	.000197	5		
120	.0049	125	5,000	.000099	2.5		
140	.0041	105	12,000	.0000394	1		

\* 1 micron ( $\mu$ ) = 1 micrometer ( $\mu m$ ), new National Bureau of Standards terminology

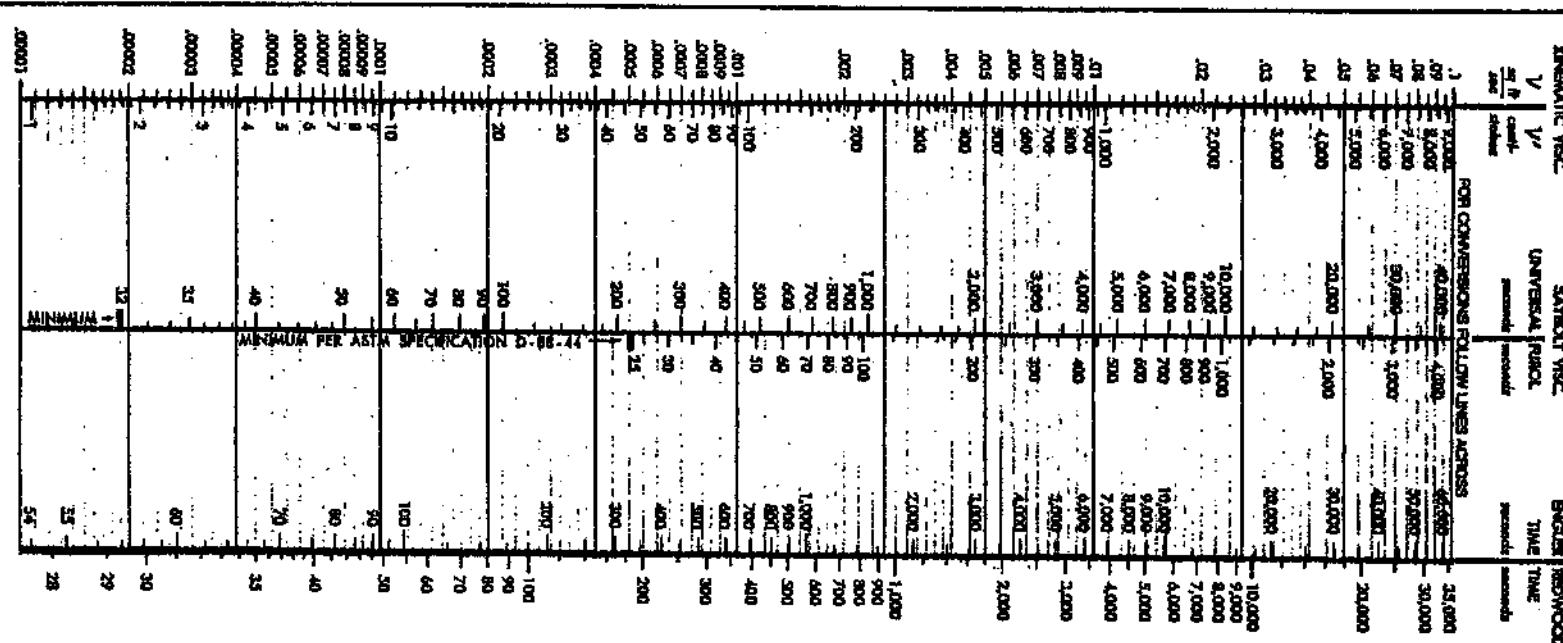
1 micron = one-millionth of a meter

Inches  $\times$  25,400 = microns or micrometers

Reference ASTM E 11-70

## D-12

Viscosity Conversions. (By permission, Tube Turns Div., Chemetron Corp., Bull. TT 725.)



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To convert other units into kinematic viscosity in English units  $\nu$  (sq ft per sec) or in Metric units  $\nu'$  (centistokes), use the chart or the formulas to the right:

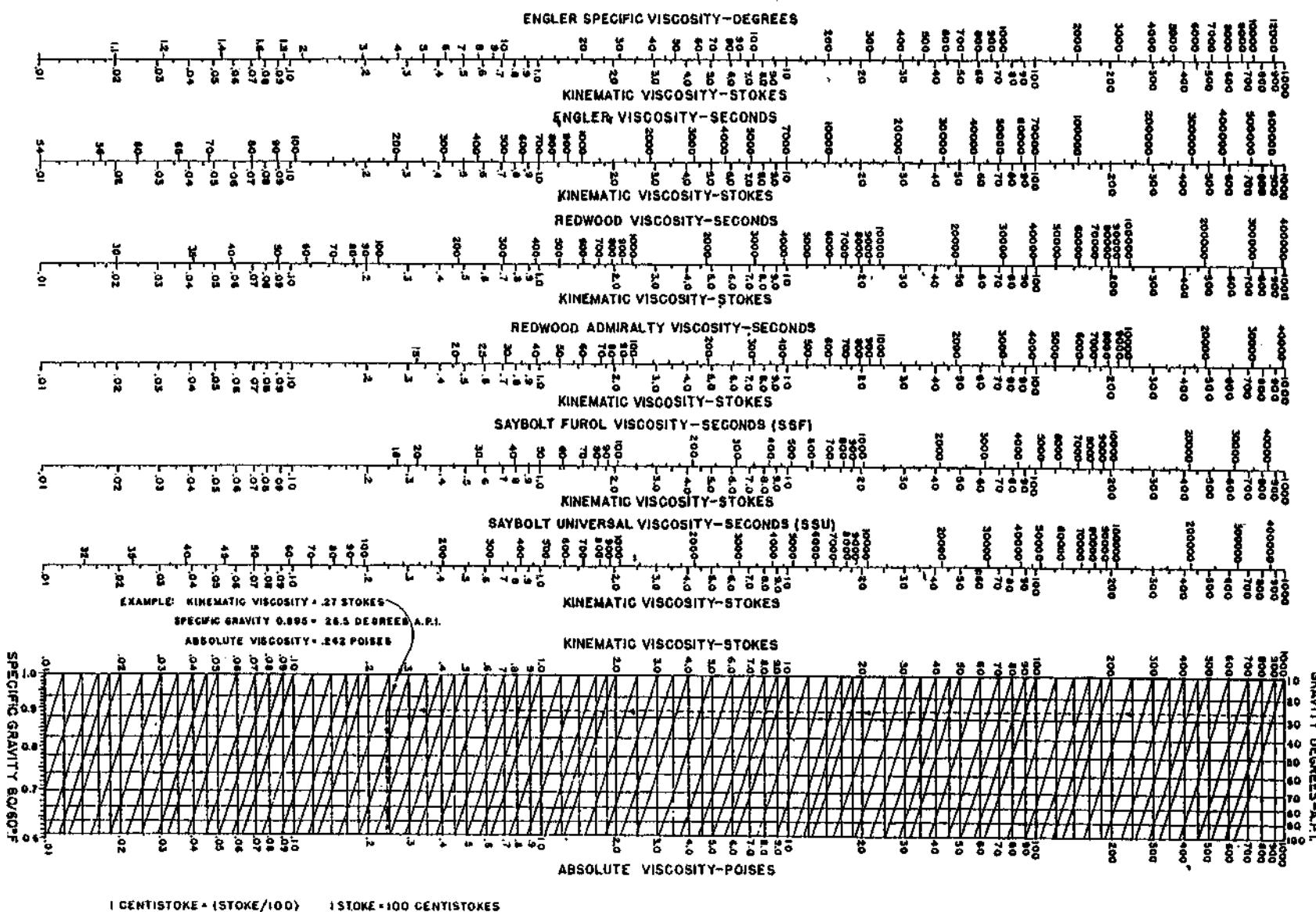
To convert:	into centistokes ( $\nu'$ )	into sq ft per sec ( $\nu$ )
from Metric units (centistokes)	$\nu' = 92800 \nu$	$\nu = 0.000\ 010\ 76 \nu'$
from English units (sq ft per sec)	see Table I in ASTM Spec. D-446-39 (plotted for basic temperature 100°F)	converted from ASTM Spec. D-446-39
from Saybolt Universal (seconds)	see Table I in ASTM Spec. D-888-44 (plotted for std temp of 122°F)	converted from ASTM Spec. D-888-44
from Saybolt Furu (seconds)	$\nu' = 0.147 \text{ Engler} - \frac{374}{\text{Engler}}$	$\nu = 0.000\ 001\ 58 \text{ Engler} - \frac{0.00403}{\text{Engler}}$
from Engler (seconds)	$\nu' = 0.280 \text{ Redwood} - \frac{171.8}{\text{Redwood}}$	$\nu = 0.000\ 002\ 80 \text{ Redwood} - \frac{0.00188}{\text{Redwood}}$
from Redwood standard (seconds)	$\nu' = \frac{\text{centipoises}}{\text{density}}$	$\nu = 32.2 \mu \frac{(\text{lb sec per sq ft})}{\rho (\text{lb per cu ft})}$
from absolute viscosity		

To convert degrees API and Baumé into Specific Gravity, use the formulas to right:

Liquids lighter than water (API Formula)	Liquids heavier than water (U.S. Bureau of Stds.)
Specific gravity 60/80°F = $\frac{141.5}{131.5 + \text{Degrees API}}$	Specific gravity = $\frac{145}{145 - \text{Degrees Baumé}}$

D-13

Viscosity Conversions. (Courtesy Kinney Vacuum Div., The New York Air Brake Co.)



**D-14**  
**Commercial Wrought Steel Pipe Data**  
**(Based on ANSI B36.10 wall thicknesses)**

Nominal Pipe Size  Inches	Outside Diam- eter  Inches	Thick- ness  Inches	Inside Diameter		Inside Diameter Functions (in inches)				Transverse Internal Area		
			<i>d</i>	<i>D</i>	<i>d</i> <sup>2</sup>	<i>d</i> <sup>3</sup>	<i>d</i> <sup>4</sup>	<i>d</i> <sup>5</sup>	<i>a</i> Sq. In.	<i>A</i> Sq. Ft.	
			Inches	Feet							
<b>Schedule 10</b>	14	14	0.250	13.5	1.125	182.25	2460.4	33215.	448400.	143.14	0.994
	16	16	0.250	15.5	1.291	240.25	3723.9	57720.	894660.	188.69	1.310
	18	18	0.250	17.5	1.4583	306.25	5359.4	93789.	1641309.	240.53	1.670
	20	20	0.250	19.5	1.625	380.25	7414.9	144590.	2819500.	298.65	2.074
	24	24	0.250	23.5	1.958	557.25	12977.	304980.	7167030.	433.74	3.012
	30	30	0.312	29.376	2.448	862.95	25350.	744288.	21864218.	677.76	4.707
<b>Schedule 20</b>	8	8.625	0.250	8.125	0.6771	66.02	536.38	4359.3	35409.	51.85	0.3601
	10	10.75	0.250	10.25	0.8542	105.06	1076.9	11038.	113141.	82.52	0.5731
	12	12.75	0.250	12.25	1.021	150.06	1838.3	22518.	275855.	117.86	0.8185
	14	14	0.312	13.376	1.111	178.92	2393.2	32012.	428185.	140.52	0.9758
	16	16	0.312	15.376	1.281	236.42	3635.2	55894.	859442.	185.69	1.290
	18	18	0.312	17.376	1.448	301.92	5246.3	91156.	1583978.	237.13	1.647
<b>Schedule 30</b>	20	20	0.375	19.250	1.604	370.56	7133.3	137317.	2643352.	291.04	2.021
	24	24	0.375	23.25	1.937	540.56	12568.	292205.	6793832.	424.56	2.948
	30	30	0.500	29.00	2.417	841.0	24389.	707281.	20511149.	660.52	4.587
	8	8.625	0.277	8.071	0.6726	65.51	525.75	4243.2	34248.	51.16	0.3553
	10	10.75	0.307	10.136	0.8447	102.74	1041.4	10555.	106987.	80.69	0.5603
	12	12.75	0.330	12.09	1.0075	146.17	1767.2	21366.	258304.	114.80	0.7972
<b>Schedule 40</b>	14	14	0.375	13.25	1.1042	175.56	2326.2	30821.	408394.	137.88	0.9575
	16	16	0.375	15.25	1.2708	232.56	3546.6	54084.	824801.	182.65	1.268
	18	18	0.438	17.124	1.4270	293.13	5021.3	85984.	1472397.	230.30	1.599
	20	20	0.500	19.00	1.5833	361.00	6859.0	130321.	2476099.	283.53	1.969
	24	24	0.562	22.876	1.9063	523.31	11971.	273853.	6264703.	411.00	2.854
	30	30	0.625	28.75	2.3958	826.56	23764.	683201.	19642160.	649.18	4.508
<b>Schedule 40</b>	1/8	0.405	0.068	0.169	0.0224	0.0724	0.0195	0.005242	0.00141	0.057	0.00040
	1/4	0.540	0.088	0.364	0.0303	0.1325	0.0482	0.01756	0.00639	0.104	0.00072
	3/8	0.675	0.091	0.493	0.0411	0.2430	0.1198	0.05905	0.02912	0.191	0.00133
	1/2	0.840	0.109	0.672	0.0518	0.3869	0.2406	0.1497	0.09310	0.304	0.00211
	5/8	1.050	0.113	0.824	0.0687	0.679	0.5595	0.4610	0.3799	0.533	0.00371
	1	1.315	0.133	1.049	0.0874	1.100	1.154	1.210	1.270	0.864	0.00600
	1 1/4	1.660	0.140	1.380	0.1150	1.904	2.628	3.625	5.005	1.495	0.01040
	1 1/2	1.900	0.145	1.610	0.1342	2.592	4.173	6.718	10.82	2.036	0.01414
	2	2.375	0.154	2.067	0.1722	4.172	8.831	18.250	37.72	3.355	0.02330
	2 1/4	2.875	0.203	2.469	0.2057	6.096	15.051	37.161	91.75	4.788	0.03322
	3	3.500	0.216	3.068	0.2557	9.413	28.878	88.605	271.8	7.393	0.05130
	3 1/2	4.000	0.226	3.548	0.2957	12.59	44.663	158.51	562.2	9.886	0.06870
	4	4.500	0.237	4.026	0.3355	16.21	65.256	262.76	1058.	12.730	0.08840
	5	5.563	0.258	5.047	0.4206	25.47	128.56	648.72	3275.	20.006	0.1390
	6	6.625	0.280	6.065	0.5054	36.78	223.10	1352.8	8206.	28.891	0.2006
<b>Schedule 60</b>	8	8.625	0.322	7.981	0.6651	63.70	508.36	4057.7	32380.	50.017	0.1474
	10	10.75	0.365	10.02	0.8350	108.4	1006.0	10080.	101000.	78.855	0.5475
	12	12.75	0.406	11.938	0.9965	142.5	1701.3	20306.	242470.	111.93	0.7773
	14	14.0	0.438	13.124	1.0937	172.24	2260.5	29666.	389340.	135.28	0.9394
	16	16.0	0.500	18.000	1.250	225.0	3375.0	50625.	759375.	176.72	1.2272
	18	18.0	0.562	16.876	1.4063	284.8	4806.3	81111.	1368820.	223.68	1.5533
<b>Schedule 80</b>	20	20.0	0.593	18.814	1.5678	354.0	6659.5	125320.	2357244.	278.00	1.9305
	24	24.0	0.667	22.626	1.8855	511.9	11583.	261040.	5929784.	402.07	2.7921
	8	8.625	0.406	7.813	0.6511	61.04	476.93	3725.9	29113.	47.94	0.3329
	10	10.75	0.500	9.750	0.8125	95.06	926.86	9036.4	88110.	74.66	0.5185
	12	12.75	0.562	11.626	0.9688	135.16	1571.4	18268.	212399.	106.16	0.7372
	14	14.0	0.593	11.814	1.0678	164.20	2104.0	26962.	345480.	128.96	0.8956
<b>Schedule 80</b>	16	16.0	0.656	14.688	1.2240	215.74	3168.8	46544.	683618.	169.44	1.1766
	18	18.0	0.750	16.500	1.3750	272.25	4492.1	74120.	1222982.	213.83	1.4849
	20	20.0	0.812	18.376	1.5313	337.68	6205.2	114028.	2095342.	265.21	1.8417
	24	24.0	0.968	22.064	1.8387	486.82	10741.	236994.	5229036.	382.35	2.6552
	1/8	0.405	0.095	0.215	0.0179	0.0462	0.00994	0.002134	0.000459	0.036	0.00025
	1/4	0.540	0.119	0.302	0.0252	0.0912	0.0275	0.008317	0.002513	0.072	0.00050
<b>Schedule 80</b>	3/8	0.675	0.126	0.423	0.0353	0.1789	0.0757	0.03200	0.01354	0.141	0.00098
	1/2	0.840	0.147	0.546	0.0455	0.2981	0.1628	0.08886	0.04852	0.234	0.00163
	5/8	1.050	0.154	0.742	0.0618	0.5506	0.4085	0.3032	0.2249	0.433	0.00300
	1	1.315	0.179	0.957	0.0797	0.9158	0.8765	0.8387	0.8027	0.719	0.00499
	1 1/4	1.660	0.191	1.278	0.1065	1.633	1.087	2.6667	3.409	1.283	0.00891

Courtesy Crane Co., Technical Manual 410, Flow of Fluids.

**D-14**  
**(Continued). Commercial Wrought Steel Pipe Data**  
**(Based on ANSI B36.10 wall thicknesses)**

Nominal Pipe Size  Inches	Outside Diam- eter  Inches	Thick- ness  Inches	Inside Diameter		Inside Diameter Functions (In Inches)				Transverse Internal Area	
			d	D	d <sup>2</sup>	d <sup>3</sup>	d <sup>4</sup>	d <sup>5</sup>	a	A
			Inches	Feet					Sq. In.	Sq. Ft.
Schedule 80—cont.	1 1/4	1.900	0.200	1.500	0.1250	2.250	3.375	5.062	7.594	1.767 0.01225
	2	2.375	0.218	1.939	0.1616	3.760	7.290	14.136	27.41	2.953 0.02050
	2 1/2	2.875	0.276	2.323	0.1936	5.396	12.536	29.117	67.64	4.238 0.02942
	3	3.5	0.300	2.900	0.2417	8.410	24.389	70.728	205.1	6.605 0.04587
	3 1/2	4.0	0.318	3.364	0.2803	11.32	38.069	128.14	430.8	8.888 0.06170
	4	4.5	0.337	3.826	0.3188	14.64	56.006	214.33	819.8	11.497 0.07946
	5	5.563	0.375	4.813	0.4011	23.16	111.49	536.38	2583.	18.194 0.1263
	6	6.625	0.432	5.761	0.4801	33.19	191.20	1101.6	6346.	26.067 0.1810
	8	8.625	0.500	7.625	0.6354	58.14	443.32	3380.3	25775.	45.663 0.3171
	10	10.75	0.593	9.564	0.7970	91.47	874.82	8366.8	80020.	71.84 0.4989
	12	12.75	0.687	11.376	0.9480	129.41	1472.2	16747.	190523.	101.64 0.7058
	14	14.0	0.750	12.500	1.0417	156.25	1953.1	24414.	305176.	122.72 0.8522
	16	16.0	0.843	14.314	1.1928	204.89	2932.8	41980.	600904.	160.92 1.1175
	18	18.0	0.937	16.126	1.3438	260.05	4193.5	67626.	1090518.	204.24 1.4183
	20	20.0	1.031	17.938	1.4948	321.77	5771.9	103536.	1857248.	252.72 1.7550
	24	24.0	1.218	21.564	1.7970	465.01	10027.	2162347.	4662798.	365.22 2.5362
Schedule 100	8	8.625	0.593	7.439	0.6199	55.34	411.66	3062.	22781.	43.46 0.3018
	10	10.75	0.718	9.314	0.7762	86.75	807.99	7526.	69357.	68.13 0.4732
	12	12.75	0.843	11.064	0.9220	122.41	1354.4	14985.	165791.	96.14 0.6677
	14	14.0	0.937	12.126	1.0105	147.04	1783.0	21621.	262173.	115.49 0.8020
	16	16.0	1.031	13.938	1.1615	194.27	2707.7	37740.	526020.	152.58 1.0596
	18	18.0	1.156	15.688	1.3057	246.11	3861.0	60572.	950750.	193.30 1.3423
	20	20.0	1.281	17.438	1.4532	304.08	5302.6	92467.	1612438.	238.83 1.6585
	24	24.0	1.531	20.938	1.7448	438.40	9179.2	192195.	4024179.	344.32 2.3911
Schedule 120	4	4.50	0.438	3.624	0.302	13.133	47.595	172.49	625.1	10.315 0.07163
	5	5.563	0.500	4.563	0.3802	20.82	95.006	433.5	1978.	16.35 0.1136
	6	6.625	0.562	5.501	0.4584	30.26	166.47	915.7	5037.	23.77 0.1650
	8	8.625	0.718	7.189	0.5991	51.68	371.54	2671.	19202.	40.59 0.2819
	10	10.75	0.843	9.064	0.7553	82.16	744.66	6750.	61179.	64.53 0.4481
	12	12.75	1.000	10.750	0.8959	115.56	1242.3	13355.	143563.	90.76 0.6303
	14	14.0	1.093	11.814	0.9845	139.57	1648.9	19480.	230137.	109.62 0.7612
	16	16.0	1.218	13.564	1.1303	183.98	2495.5	33849.	459133.	144.50 1.0035
	18	18.0	1.375	15.250	1.2708	232.56	3546.6	54086.	824804.	182.66 1.2684
	20	20.0	1.500	17.000	1.4166	289.00	4913.0	83521.	1419857.	226.98 1.5762
	24	24.0	1.812	20.376	1.6980	415.18	8459.7	172375.	3512313.	326.08 2.2645
Schedule 140	8	8.625	0.812	7.001	0.5834	49.01	343.15	2402.	16819.	38.50 0.2673
	10	10.75	1.000	8.750	0.7292	76.56	669.92	5862.	51291.	60.13 0.4176
	12	12.75	1.125	10.500	0.8750	110.25	1157.6	12155.	127628.	86.59 0.6013
	14	14.0	1.250	11.500	0.9583	132.25	1520.9	17490.	201136.	103.87 0.7213
	16	16.0	1.438	13.124	1.0937	172.24	2260.5	29666.	389340.	135.18 0.9394
	18	18.0	1.562	14.876	1.2396	221.30	3292.0	48972.	728502.	173.80 1.2070
	20	20.0	1.750	16.5	1.3750	272.25	4492.1	74120.	1222981.	213.82 1.4849
	24	24.0	2.062	19.876	1.6563	395.06	7852.1	156069.	3102021.	310.28 2.1547
Schedule 160	1/2	0.840	0.187	0.466	0.0388	0.2172	0.1012	0.04716	0.02197	0.1706 0.00118
	3/4	1.050	0.218	0.614	0.0512	0.3770	0.2315	0.1421	0.08726	0.2961 0.00206
	1	1.315	0.250	0.815	0.0679	0.6642	0.5413	0.4412	0.3596	0.5217 0.00362
	1 1/4	1.660	0.250	1.160	0.0966	1.346	1.561	1.811	2.100	1.057 0.00734
	1 1/2	1.900	0.281	1.338	0.1115	1.790	2.395	3.205	4.288	1.406 0.00976
	2	2.375	0.343	1.689	0.1407	2.853	4.818	8.138	13.74	2.241 0.01556
	2 1/2	2.875	0.375	2.125	0.1771	4.516	9.596	20.19	43.33	3.546 0.02463
	3	3.50	0.438	2.624	0.2187	6.885	18.067	47.41	124.4	5.408 0.03755
	4	4.50	0.531	3.438	0.2865	11.82	40.637	139.7	480.3	9.283 0.06447
	5	5.563	0.625	4.313	0.3594	18.60	80.230	346.0	1492.	14.61 0.1015
	6	6.625	0.718	5.189	0.4324	26.93	139.72	725.0	3762.	21.15 0.1469
	8	8.625	0.906	6.813	0.5677	46.42	316.24	2155.	14679.	36.46 0.2532
	10	10.75	1.125	8.500	0.7083	72.25	614.12	5220.	44371.	56.75 0.3941
	12	12.75	1.312	10.126	0.8438	102.54	1038.3	10514.	106461.	80.53 0.5592
	14	14.0	1.406	11.188	0.9323	125.17	1400.4	15668.	175291.	98.31 0.6827
	16	16.0	1.593	12.814	1.0678	164.20	2104.0	26961.	345482.	128.96 0.8956
	18	18.0	1.781	14.438	1.2032	208.45	3009.7	43454.	627387.	163.72 1.1369
	20	20.0	1.968	16.064	1.3387	258.05	4145.3	66590.	1069715.	202.67 1.4074
	24	24.0	2.343	19.314	1.6095	373.03	7204.7	139152.	2687582.	292.98 2.0346

**D-14**  
**(Continued). Commercial Wrought Steel Pipe Data**  
**(Based on ANSI B36.10 wall thicknesses)**

Nominal Pipe Size Inches	Outside Diam- eter Inches	Thick- ness Inches	Inside Diameter		Inside Diameter Functions (In Inches)				Transverse Internal Area	
			d Inches	D Feet	d <sup>2</sup>	d <sup>3</sup>	d <sup>4</sup>	d <sup>5</sup>	a Sq. In.	A Sq. Ft.
<b>Standard Wall Pipe</b>										
1/8	0.405	0.068	0.269	0.0224	0.0724	0.0195	0.00524	0.00141	0.057	0.00040
1/4	0.540	0.088	0.364	0.0303	0.1325	0.0482	0.01756	0.00639	0.104	0.00072
3/8	0.675	0.091	0.493	0.0411	0.2430	0.1198	0.05905	0.02912	0.191	0.00133
1/2	0.840	0.109	0.622	0.0518	0.3869	0.2406	0.1497	0.0931	0.304	0.00211
5/8	1.050	0.113	0.824	0.0687	0.679	0.5595	0.4610	0.3799	0.533	0.00371
1	1.315	0.133	1.049	0.0874	1.100	1.154	1.210	1.270	0.864	0.00600
1 1/4	1.660	0.140	1.380	0.1150	1.904	2.628	3.625	5.005	1.495	0.01040
1 1/2	1.900	0.145	1.610	0.1342	2.592	4.173	6.718	10.82	2.036	0.01414
2	2.375	0.154	2.067	0.1722	4.272	8.831	18.250	37.72	3.355	0.02330
2 1/2	2.875	0.203	2.469	0.2057	6.096	15.051	37.161	91.75	4.788	0.03322
3	3.500	0.216	3.068	0.2557	9.413	28.878	88.605	271.8	7.393	0.05130
3 1/2	4.000	0.226	3.548	0.2957	12.59	44.663	158.51	562.2	9.886	0.06870
4	4.500	0.237	4.026	0.3355	16.21	65.256	262.76	1058.	12.730	0.08840
5	5.563	0.258	5.047	0.4206	25.47	128.56	648.72	3275.	20.006	0.1390
6	6.625	0.280	6.065	0.5054	36.78	223.10	1352.8	8206.	28.891	0.2006
8	8.625	0.277	8.071	0.6725	65.14	525.75	4243.0	34248.	51.161	0.3553
	8.625S	0.312	7.981	0.6651	63.70	508.36	4057.7	32380.	50.027	0.3474
10	10.75	0.279	10.192	0.8493	103.88	1058.7	10789.	109876.	81.585	0.5666
10	10.75	0.307	10.136	0.8446	102.74	1041.4	10555.	106987.	89.691	0.5604
	10.75S	0.365	10.020	0.8350	100.4	1006.0	10080.	101000.	78.855	0.5475
12	12.75	0.330	12.090	1.0075	146.17	1767.2	21366.	258300.	114.80	0.7972
	12.75S	0.375	12.000	1.000	144.0	1728.0	20736.	248800.	113.10	0.7854
<b>Extra Strong Pipe</b>										
1/8	0.405	0.095	0.215	0.0179	0.0462	0.00994	0.002134	0.000459	0.036	0.00025
1/4	0.540	0.119	0.302	0.0252	0.0912	0.0275	0.008317	0.002513	0.072	0.00050
3/8	0.675	0.126	0.423	0.0353	0.1789	0.0757	0.03201	0.01354	0.141	0.00098
1/2	0.840	0.147	0.546	0.0455	0.2981	0.1628	0.08886	0.04852	0.234	0.0163
5/8	1.050	0.154	0.742	0.0618	0.5506	0.4085	0.3032	0.2249	0.433	0.0300
1	1.315	0.129	0.957	0.0797	0.9158	0.8765	0.8387	0.8027	0.719	0.04499
1 1/4	1.660	0.191	1.278	0.1065	1.633	2.087	2.6667	3.409	1.283	0.0891
1 1/2	1.900	0.200	1.500	0.1250	2.250	3.375	5.062	7.594	1.767	0.01225
2	2.375	0.218	1.939	0.1616	3.760	7.290	14.136	27.41	2.953	0.02050
2 1/2	2.875	0.276	2.323	0.1936	5.396	12.536	29.117	67.64	4.238	0.02942
3	3.500	0.300	2.900	0.2417	8.410	24.389	70.728	205.1	6.605	0.04587
3 1/2	4.000	0.318	3.364	0.2803	11.32	38.069	128.14	430.8	8.888	0.06170
4	4.500	0.337	3.826	0.3188	14.64	56.006	214.33	819.8	11.497	0.07986
5	5.563	0.375	4.813	0.4011	23.16	111.49	536.6	2583.	18.194	0.1263
6	6.625	0.432	5.761	0.4801	33.19	191.20	1101.6	6346.	26.067	0.1810
8	8.625	0.500	7.625	0.6354	58.14	443.32	3380.3	25775.	45.663	0.3171
10	10.75	0.500	9.750	0.8125	95.06	926.86	9036.4	88110.	74.662	0.5185
12	12.75	0.500	11.750	0.9792	138.1	1622.2	19072.	223970.	108.434	0.7528
<b>Double Extra Strong Pipe</b>										
1/2	0.840	0.294	0.252	0.0210	0.0635	0.0160	0.004032	0.00102	0.050	0.00035
3/8	1.050	0.308	0.434	0.0362	0.1884	0.0817	0.03549	0.01540	0.148	0.00103
1	1.315	0.358	0.599	0.0499	0.3588	0.2149	0.1287	0.07711	0.282	0.00196
1 1/4	1.660	0.382	0.896	0.0747	0.8028	0.7193	0.6445	0.5775	0.630	0.00438
1 1/2	1.900	0.400	1.100	0.0917	1.210	1.331	1.4641	1.611	0.950	0.00660
2	2.375	0.436	1.503	0.1252	2.259	3.395	5.1031	7.670	1.774	0.01232
2 1/2	2.875	0.552	1.771	0.1476	3.136	5.554	9.8345	17.42	2.464	0.01710
3	3.500	0.600	2.300	0.1917	5.290	12.167	27.984	64.36	4.155	0.02885
3 1/2	4.000	0.636	2.728	0.2273	7.442	20.302	55.183	151.1	5.345	0.04059
4	4.500	0.674	3.152	0.2627	9.935	31.315	98.704	311.1	7.803	0.05419
5	5.563	0.750	4.063	0.3386	16.51	67.072	272.58	1107.	12.966	0.09006
6	6.625	0.864	4.897	0.4081	23.98	117.43	575.04	2816.	18.835	0.1308
8	8.625	0.875	6.875	0.5729	47.27	324.95	1234.4	15360.	37.122	0.2578

**D-15**  
**Stainless Steel Pipe Data**  
**(Based on ANSI B36.19 wall thicknesses)**

Nominal Pipe Size Inches	Outside Diam- eter Inches	Thick- ness Inches	Inside Diameter		Inside Diameter Functions (in Inches)				Transverse Internal Area	
			d Inches	D Feet	d <sup>2</sup>	d <sup>3</sup>	d <sup>4</sup>	d <sup>5</sup>	a Sq. In.	A Sq. Ft.

**Schedule 5 S**

1/2	0.840	0.065	0.710	0.0592	0.504	0.358	0.254	0.1804	0.396	0.00275
3/4	1.050	0.065	0.920	0.0767	0.846	0.779	0.716	0.659	0.664	0.00461
1	1.315	0.065	1.185	0.0988	1.404	1.664	1.972	2.337	1.103	0.00766
1 1/4	1.660	0.065	1.530	0.1275	2.341	3.582	5.480	8.384	1.839	0.01277
1 1/2	1.900	0.065	1.770	0.1475	3.133	5.545	9.815	17.37	2.461	0.01709
2	2.375	0.065	2.245	0.1871	5.040	11.31	25.40	57.03	3.958	0.02749
2 1/2	2.875	0.083	2.709	0.2258	7.339	19.88	53.86	145.9	5.764	0.04003
3	3.500	0.083	3.334	0.2778	11.12	37.06	123.6	411.9	8.733	0.06065
3 1/2	4.000	0.083	3.834	0.3195	14.70	56.36	216.1	828.4	11.545	0.08017
4	4.500	0.083	4.334	0.3612	18.78	81.41	352.8	1529.	14.750	0.1024
5	5.563	0.109	5.345	0.4454	28.57	152.7	816.2	4363.	22.439	0.1558
6	6.625	0.109	6.407	0.5339	41.05	263.0	1685.	10796.	32.241	0.2239
8	8.625	0.109	8.407	0.7006	70.68	594.2	4995.	41996.	55.512	0.3855
10	10.750	0.134	10.482	0.8375	109.9	1152.	12072.	126538.	86.315	0.5994
12	12.750	0.156	12.438	1.0365	154.7	1924.	23933.	297682.	121.50	0.8438

**Schedule 10 S**

1/4	0.405	0.049	0.307	0.0256	0.0942	0.0289	0.00888	0.00273	0.074	0.00051
3/4	0.540	0.065	0.410	0.0342	0.1681	0.0689	0.02826	0.01159	0.132	0.00092
7/8	0.675	0.065	0.545	0.0454	0.2970	0.1619	0.08322	0.04808	0.233	0.00162
1/2	0.840	0.083	0.674	0.0562	0.4543	0.3062	0.2064	0.1391	0.357	0.00248
3/4	1.050	0.083	0.884	0.0737	0.7815	0.6908	0.6107	0.5398	0.614	0.00426
1	1.315	0.109	1.097	0.0914	1.203	1.320	1.448	1.589	0.945	0.00656
1 1/4	1.660	0.109	1.442	0.1202	2.079	2.998	4.324	6.235	1.633	0.01134
1 1/2	1.900	0.109	1.682	0.1402	2.829	4.759	8.004	13.46	2.122	0.01543
2	2.375	0.109	2.157	0.1798	4.653	10.04	21.65	46.69	3.654	0.02538
2 1/2	2.875	0.120	2.635	0.2196	6.943	18.30	48.21	127.0	5.453	0.03787
3	3.500	0.120	3.260	0.2717	10.63	34.65	112.9	368.2	8.347	0.05796
3 1/2	4.000	0.120	3.760	0.3133	14.14	53.16	199.9	751.5	11.11	0.07712
4	4.500	0.120	4.260	0.3550	18.15	77.31	329.3	1403.	14.26	0.09899
5	5.563	0.134	5.195	0.4413	28.04	148.5	786.1	4162.	22.02	0.1529
6	6.625	0.134	6.357	0.5298	40.41	256.9	1633.	10382.	31.74	0.2204
8	8.625	0.148	8.329	0.6941	69.37	577.8	4813.	40083.	54.48	0.3784
10	10.750	0.165	10.420	0.8683	108.6	1131.	11789.	122840.	85.39	0.5923
12	12.750	0.180	12.390	1.0325	153.5	1902.	23566.	291982.	120.6	0.8372

**Schedule 40 S**

1/2 to 12	Values are the same, size for size, as those shown on the facing page for Standard Wall Pipe (heaviest weight on 8, 10, and 12-inch sizes).
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**Schedule 80 S**

1/2 to 12	Values are the same, size for size, as those shown on the facing page for Extra Strong Pipe.
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## D-16

### Properties of Pipe

Tabulated below are the most generally required data used in piping design. This table is believed to be the most comprehensive published up to this time. Many thicknesses traditionally included in such tables have been omitted because of their having become obsolete through disuse and lack of coverage by any Standard.

Sizes and thicknesses listed herein are covered by the following Standards:—

- 1) American National Standard Institute B36.10
- 2) American National Standard Institute B36.19
- 3) American Petroleum Institute Standard API 5L
- 4) American Petroleum Institute Standard API 5LX
- 5) New United States Legal Standard for Steel Plate Gauges.

Sizes and thicknesses to which no Standard designation applies are largely the more commonly used dimensions to which Taylor Forge Electric Fusion Welded Pipe is produced for a wide variety of applications including river crossings, penstocks, power plant and other piping.

All data is computed from the *nominal* dimensions listed and the effect of tolerances is not taken into account. Values are computed by application of the following formulas:

$$\text{Radius of Gyration: } R = \frac{\sqrt{D^2 + d^2}}{4}$$

$$\text{Moment of Inertia: } I = R^2 A$$

$$\text{Section Modulus: } Z = \frac{I}{0.5 D}$$

**ANSI** American National Standards Institute

Nominal		Design- nation	Wall Thick- ness	Inside Diam.  <b>D</b>	Weight per Foot	Wt. of Water per Ft. of Pipe	Sq. Ft. Outside Surface per Ft.	Sq. Ft. Inside Surface per Ft.	Trans- verse Area in. <sup>2</sup>	Area of Metal in. <sup>2</sup>	Moment of Inertia in. <sup>4</sup>	Section Modulus in. <sup>3</sup>	Radius of Gyra- tion in.
Pipe Size	Outside Diam.								<b>a</b>				<b>R</b>
$\frac{1}{8}$	<b>.405</b>	10S	.049	.307	.186	.0320	.106	.0804	.0740	.0548	.00090	.00440	.1270
		Std.	.068	.269	.244	.0246	.106	.0705	.0568	.0720	.00106	.00530	.1215
		X-Stg.	.095	.215	.314	.0157	.106	.0563	.0364	.0925	.00122	.00600	.1146
$\frac{1}{4}$	<b>.540</b>	10S	.065	.410	.330	.0570	.141	.1073	.1320	.0970	.00280	.01030	.1695
		Std.	.088	.364	.424	.0451	.141	.0955	.1041	.1250	.00331	.01230	.1628
		X-Stg.	.119	.302	.535	.0310	.141	.0794	.0716	.1574	.00378	.01395	.1547
$\frac{3}{8}$	<b>.675</b>	10S	.065	.545	.423	.1010	.177	.1427	.2333	.1245	.00590	.01740	.2160
		Std.	.091	.493	.567	.0827	.177	.1295	.1910	.1670	.00730	.02160	.2090
		X-Stg.	.126	.423	.738	.0609	.177	.1106	.1405	.2173	.00862	.02554	.1991
$\frac{1}{2}$	<b>.840</b>	10S	.083	.670	.671	.1550	.220	.1764	.3568	.1974	.01430	.03410	.2692
		Std.	.109	.622	.850	.1316	.220	.1637	.3040	.2503	.01710	.04070	.2613
		X-Stg.	.147	.546	1.087	.1013	.220	.1433	.2340	.3200	.02010	.04780	.2505
		160	.138	.464	1.310	.0740	.220	.1220	.1706	.3836	.02213	.05269	.2402
$\frac{5}{8}$	<b>1.050</b>	XX-Stg.	.294	.252	1.714	.0216	.220	.0660	.0499	.5043	.02424	.05772	.2192
		10S	.083	.884	.857	.2660	.275	.2314	.6138	.2522	.02970	.05660	.3430
		Std.	.113	.824	1.130	.2301	.275	.2168	.5330	.3326	.03704	.07055	.3337
		X-Stg.	.154	.742	1.473	.1875	.275	.1948	.4330	.4335	.04479	.08531	.3214
$\frac{3}{4}$	<b>1.315</b>	160	.219	.612	1.940	.1280	.275	.1607	.2961	.5698	.05270	.10038	.3041
		XX-Stg.	.308	.434	2.440	.0633	.275	.1137	.1479	.7180	.05792	.11030	.2840
		10S	.109	1.097	1.404	.4090	.344	.2872	.9448	.4129	.07560	.1150	.4282
$1\frac{1}{4}$	<b>1.660</b>	Std.	.133	1.049	1.678	.3740	.344	.2740	.8640	.4939	.08734	.1328	.4205
		X-Stg.	.179	.957	2.171	.3112	.344	.2520	.7190	.6388	.10560	.1606	.4066
		160	.250	.815	2.850	.2261	.344	.2134	.5217	.8364	.12516	.1903	.3868
$1\frac{1}{2}$	<b>1.660</b>	XX-Stg.	.358	.599	3.659	.1221	.344	.1570	.2818	1.0760	.14050	.2136	.3613
		10S	.109	1.442	1.806	.7080	.434	.3775	1.633	.5314	.1606	.1934	.5499
		Std.	.140	1.380	2.272	.6471	.434	.3620	1.495	.6685	.1947	.2346	.5397
$2\frac{1}{2}$	<b>1.660</b>		.191	1.278	2.996	.5553	.434	.3356	1.283	.8815	.2418	.2913	.5237
			.250	1.160	3.764	.4575	.434	.3029	1.057	1.1070	.2833	.3421	.5063
			.382	.896	5.214	.2732	.434	.2331	.6305	1.5340	.3411	.4110	.4716

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**D-16**  
**(Continued). Properties of Pipe**

<b>Nominal</b>		<b>Designation</b>	<b>Wall Thickness</b>	<b>Inside Diam.</b>	<b>d</b>	<b>Weight per Foot</b>	<b>Wt. of Water per Ft. of Pipe</b>	<b>Sq. Ft. Outside Surface per Ft.</b>	<b>Sq. Ft. Inside Surface per Ft.</b>	<b>Transverse Area in.<sup>2</sup></b>	<b>Area of Metal in.<sup>2</sup></b>	<b>Moment of Inertia in.<sup>4</sup></b>	<b>Section Modulus in.<sup>3</sup></b>	<b>Radius of Gyration in.</b>
<b>Pipe Size</b>	<b>Outside Diam.</b>													
<b>D</b>														
<b>1½</b>	<b>1.900</b>	10S Std.	.109	1.682	2.085	.9630	.497	.4403	2.221	.613	.2469	.2599	.6344	
		.145	1.610	2.717	.8820	.497	.4213	2.036	.800	.3099	.3262	.6226		
		X-Stg.	.200	1.500	3.631	.7648	.497	.3927	1.767	1.068	.3912	.4118	.6052	
		160	.281	1.337	4.862	.6082	.497	.3519	1.405	1.430	.4826	.5080	.5809	
		XX-Stg.	.400	1.100	6.408	.4117	.497	.2903	.950	1.885	.5678	.5977	.5489	
		10S Std.	.109	2.157	2.638	1.563	.622	.5647	3.654	.775	.5003	.4213	.8034	
<b>2</b>	<b>2.375</b>	.154	2.067	3.652	1.452	.622	.5401	3.355	1.075	.6657	.5606	.7871		
		X-Stg.	.218	1.939	5.022	1.279	.622	.5074	2.953	1.477	.8679	.7309	.7665	
		--	.250	1.875	5.673	1.196	.622	.4920	2.761	1.669	.9555	.8046	.7565	
		160	.344	1.687	7.450	.970	.622	.4422	2.240	2.190	1.162	.9790	.7286	
		XX-Stg.	.436	1.503	9.029	.769	.622	.3929	1.774	2.656	1.311	1.1040	.7027	
		10S Std.	.120	2.635	3.53	2.360	.753	.6900	5.453	1.038	.9878	.6872	.9755	
<b>2½</b>	<b>2.875</b>	.203	2.469	5.79	2.072	.753	.6462	4.788	1.704	1.530	1.064	.9474		
		X-Stg.	.276	2.323	7.66	1.834	.753	.6095	4.238	2.254	1.924	1.339	.9241	
		160	.375	2.125	10.01	1.535	.753	.5564	3.547	2.945	2.353	1.638	.8938	
		XX-Stg.	.552	1.771	13.69	1.067	.753	.4627	2.464	4.028	2.871	1.997	.8442	
		10S API	.120 .125	3.260 3.250	4.33 4.52	3.62 3.60	.916 .916	.853 .851	8.346 8.300	1.272 1.329	1.821 1.900	1.041 1.086	1.196 1.195	
		API	.156	3.188	5.58	3.46	.916	.835	7.982	1.639	2.298	1.313	1.184	
<b>3</b>	<b>3.500</b>	API Std.	.188	3.125	6.65	3.34	.916	.819	7.700	1.958	2.700	1.545	1.175	
		API	.216	3.068	7.57	3.20	.916	.802	7.393	2.228	3.017	1.724	1.164	
		API	.250	3.000	8.68	3.06	.916	.785	7.184	2.553	3.388	1.936	1.152	
		API	.281	2.938	9.65	2.94	.916	.769	6.780	2.842	3.819	2.182	1.142	
		X-Stg.	.300	2.900	10.25	2.86	.916	.761	6.605	3.016	3.892	2.225	1.136	
		160 - XX-Stg.	.438	2.624	14.32	2.34	.916	.687	5.407	4.214	5.044	2.882	1.094	
<b>3½</b>	<b>4.000</b>	.600	2.300	18.58	1.80	.916	.601	4.155	5.466	5.993	3.424	1.047		
		10S API	.120 .125	3.760 3.750	4.97 5.18	4.81 4.79	1.047 1.047	.984 .982	11.10 11.04	1.46 1.52	2.754 2.859	1.377 1.430	1.372 1.371	
		API Std.	.156	3.688	6.41	4.63	1.047	.966	10.68	1.88	3.485	1.743	1.360	
		API Std.	.188	3.624	7.71	4.48	1.047	.950	10.32	2.27	4.130	2.065	1.350	
		.226	3.548	9.11	4.28	1.047	.929	.989	9.89	2.68	4.788	2.394	1.337	
		.250	3.500	10.02	4.17	1.047	.916	.962	9.62	2.94	5.201	2.601	1.329	
<b>4</b>	<b>4.500</b>	API	.281	3.438	11.17	4.02	1.047	.900	9.28	3.29	5.715	2.858	1.319	
		X-Stg.	.318	3.364	12.51	3.85	1.047	.880	8.89	3.68	6.280	3.140	1.307	
		XX-Stg.	.636	2.728	22.85	2.53	1.047	.716	5.84	4.72	9.848	4.924	1.210	
		10S API	.120 .125	4.260 4.250	5.61 5.84	6.18 6.15	1.178 1.178	1.115 1.113	14.25 14.19	1.65 1.72	3.97 4.12	1.761 1.829	1.550 1.548	
		API Std.	.156	4.188	7.24	5.97	1.178	1.096	13.77	2.13	5.03	2.235	1.537	
		API Std.	.188	4.124	8.56	5.80	1.178	1.082	13.39	2.52	5.86	2.600	1.525	
<b>4</b>	<b>4.500</b>	API Std.	.219	4.062	10.02	5.62	1.178	1.063	12.96	2.94	6.77	3.867	1.516	
		.237	4.026	10.79	5.51	1.178	1.055	12.73	3.17	7.23	3.214	1.510		
		API	.250	4.000	11.35	5.45	1.178	1.049	12.57	3.34	7.56	3.360	1.505	
		API	.281	3.938	12.67	5.27	1.178	1.031	12.17	3.73	8.33	3.703	1.495	
		API	.312	3.876	14.00	5.12	1.178	1.013	11.80	4.11	9.05	4.020	1.482	
		X-Stg.	.337	3.826	14.98	4.98	1.178	1.002	11.50	4.41	9.61	4.271	1.477	
<b>120</b>	<b>.438</b>	.438	3.624	19.00	4.47	1.178	.949	10.32	5.59	11.65	5.177	1.444		
		.500	3.500	21.36	4.16	1.178	.916	9.62	6.28	12.77	5.676	1.425		
<b>160</b>	<b>.531</b>	3.438	22.60	4.02	1.178	.900	9.28	6.62	13.27	5.900	1.416			
		XX-Stg.	.674	3.152	27.54	3.38	1.178	.826	7.80	8.10	15.28	6.793	1.374	

**D-16**  
**(Continued). Properties of Pipe**

Nominal		Designation	Wall Thickness	Inside Diam.	Weight per Foot	Wt. of Water per Ft. of Pipe	Sq. Ft. Outside Surface per Ft.	Sq. Ft. Inside Surface per Ft.	Transverse Area in. <sup>2</sup>	Area of Metal in. <sup>2</sup>	Moment of Inertia in. <sup>4</sup>	Section Modulus in. <sup>3</sup>	Radius of Gyration in.
Pipe Size	Outside Diam.												
5	5.563	10S	.134	5.295	7.77	9.54	1.456	1.386	22.02	2.29	8.42	3.028	1.920
		API	.156	5.251	9.02	9.39	1.456	1.375	21.66	2.65	9.70	3.487	1.913
		API	.188	5.187	10.80	9.16	1.456	1.358	21.13	3.17	11.49	4.129	1.902
		API	.219	5.125	12.51	8.94	1.456	1.342	20.63	3.68	13.14	4.726	1.891
		Std.	.258	5.047	14.62	8.66	1.456	1.321	20.01	4.30	15.16	5.451	1.878
		API	.281	5.001	15.86	8.52	1.456	1.309	19.84	4.86	16.31	5.862	1.870
		API	.312	4.939	17.51	8.31	1.456	1.293	19.16	5.15	17.81	6.402	1.860
		API	.344	4.875	19.19	8.09	1.456	1.276	18.67	5.64	19.28	6.932	1.849
		X-Sig.	.375	4.813	20.78	7.87	1.456	1.260	18.19	6.11	20.67	7.431	1.839
		120	.500	4.563	27.10	7.08	1.456	1.193	16.35	7.95	25.74	9.253	1.799
6	6.625	160	.625	4.313	32.96	6.32	1.456	1.129	14.61	9.70	30.03	10.800	1.760
		XX-Sig.	.750	4.063	38.55	5.62	1.456	1.064	12.97	11.34	33.63	12.090	1.722
		12 Ga.	.104	6.417	7.25	14.02	1.734	1.680	32.34	2.13	11.33	3.42	2.31
		10S	.134	6.357	9.29	13.70	1.734	1.660	31.73	2.73	14.38	4.34	2.29
		8 Ga.	.164	6.297	11.33	13.50	1.734	1.649	31.14	3.33	17.38	5.25	2.28
		API	.188	6.249	12.93	13.31	1.734	1.639	30.70	3.80	19.71	5.95	2.28
		6 Ga.	.194	6.237	13.34	13.25	1.734	1.633	30.55	3.92	20.29	6.12	2.27
		API	.219	6.187	15.02	13.05	1.734	1.620	30.10	4.41	22.66	6.84	2.27
		API	.250	6.125	17.02	12.80	1.734	1.606	29.50	5.01	23.55	7.71	2.26
		API	.277	6.071	18.86	12.55	1.734	1.591	28.95	5.54	28.00	8.46	2.25
8	8.625	Std.	.280	6.065	18.97	12.51	1.734	1.587	28.90	5.58	28.14	8.50	2.24
		API	.312	6.001	21.05	12.26	1.734	1.571	28.28	6.19	30.91	9.33	2.23
		API	.344	5.937	23.09	12.00	1.734	1.554	27.68	6.79	33.51	10.14	2.22
		API	.375	5.875	25.10	11.75	1.734	1.540	27.10	7.37	34.20	10.90	2.21
		X-Sig.	.432	5.761	28.57	11.29	1.734	1.510	26.07	8.40	40.49	12.22	2.19
		--	.500	5.625	32.79	10.85	1.734	1.475	24.85	9.63	45.60	13.78	2.16
		120	.562	5.501	36.40	10.30	1.734	1.470	23.77	10.74	49.91	15.07	2.15
		160	.719	5.187	45.30	9.16	1.734	1.359	21.15	13.36	58.99	17.81	2.10
		XX-Sig.	.864	4.897	53.16	8.14	1.734	1.280	16.83	15.64	66.33	20.02	2.06
		12 Ga.	.104	8.417	9.47	24.1	2.26	2.204	55.6	2.78	25.3	5.86	3.01
10	8.625	10 Ga.	.134	8.357	12.16	23.8	2.26	2.168	54.8	3.57	32.2	7.44	3.00
		10S	.148	8.329	13.40	23.6	2.26	2.180	54.5	3.94	35.4	8.22	3.00
		8 Ga.	.164	8.297	14.83	23.4	2.26	2.172	54.1	4.36	39.1	9.06	2.99
		API	.188	8.249	16.90	23.2	2.26	2.161	53.5	5.00	44.5	10.30	2.98
		6 Ga.	.194	8.237	17.48	23.1	2.26	2.156	53.3	5.14	45.7	10.60	2.98
		API	.203	8.219	18.30	23.1	2.26	2.152	53.1	5.38	47.7	11.05	2.98
		API	.219	8.187	19.64	22.9	2.26	2.148	52.7	5.80	51.3	11.90	2.97
		3 Ga.	.239	8.147	21.42	22.6	2.26	2.133	52.1	6.30	55.4	12.84	2.96
		20	.250	8.125	22.40	22.5	2.26	2.127	51.8	6.58	57.7	13.39	2.96
		30	.277	8.071	24.70	22.2	2.26	2.115	51.2	7.26	63.3	14.69	2.95
12	8.625	API	.312	8.001	27.72	21.8	2.26	2.095	50.3	8.15	70.6	16.37	2.94
		Srd.	.322	7.981	28.55	21.6	2.26	2.090	50.0	9.40	72.5	16.81	2.94
		API	.344	7.937	30.40	21.4	2.26	2.078	49.5	8.94	76.8	17.81	2.93
		API	.375	7.875	33.10	21.1	2.26	2.062	48.7	9.74	83.1	19.27	2.92
		60	.406	7.813	35.70	20.8	2.26	2.045	47.9	10.48	88.8	20.58	2.91
		API	.438	7.749	38.33	20.4	2.26	2.029	47.2	11.27	94.7	21.97	2.90
		--	.500	7.625	43.39	19.8	2.26	2.006	45.6	12.76	105.7	24.51	2.88
		100	.594	7.437	50.90	18.8	2.26	1.947	43.5	14.96	121.4	28.14	2.85
		--	.625	7.373	53.40	18.5	2.26	1.931	42.7	15.71	126.5	29.33	2.84
		120	.719	7.187	60.70	17.6	2.26	1.882	40.6	17.84	140.6	32.61	2.81
140	8.625	140	.812	7.001	67.80	16.7	2.26	1.833	38.5	19.93	153.8	35.65	2.78
		XX-Sig.	.875	6.875	72.42	16.1	2.26	1.800	37.1	21.30	162.0	37.56	2.76
		160	.906	6.813	74.70	15.8	2.26	1.784	36.4	21.97	165.9	38.48	2.76

**D-16**  
**(Continued). Properties of Pipe**

Nominal		Outside Diam. <b>D</b>	Wall Thick- ness	Inside Diam. <b>d</b>	Weight per Foot	Wt. of Water per Ft. of Pipe	Sq. Ft. Outside Surface per Ft.	Sq. Ft. Inside Surface per Ft.	Trans- verse Area in. <sup>2</sup>	Area of Metal in. <sup>2</sup>	Moment of Inertia in. <sup>4</sup>	Section Modulus in. <sup>3</sup>	Radius of Gyra- tion in.
Pipe Size	Design- ation												
10	10.750	12 Ga.	.104	10.542	11.83	37.8	2.81	2.76	87.3	3.48	49.3	9.16	3.76
		10 Ga.	.134	10.482	15.21	37.4	2.81	2.74	86.3	4.47	63.0	11.71	3.75
		8 Ga.	.164	10.422	18.56	37.0	2.81	2.73	85.3	5.45	74.4	14.22	3.74
		10S	.165	10.420	18.65	36.9	2.81	2.73	85.3	5.50	76.8	14.29	3.74
		API	.188	10.374	21.12	36.7	2.81	2.72	84.5	6.20	86.3	16.10	3.74
		6 Ga.	.194	10.362	21.89	36.6	2.81	2.71	84.3	6.43	89.7	16.68	3.73
		API	.203	10.344	22.86	36.5	2.81	2.71	84.0	6.71	93.3	17.35	3.73
		API	.219	10.310	24.60	36.2	2.81	2.70	83.4	7.24	100.5	18.70	3.72
		3 Ga.	.239	10.272	28.05	35.9	2.81	2.69	82.9	7.89	109.2	20.32	3.72
		20	.250	10.250	28.03	35.9	2.81	2.68	82.6	8.24	113.6	21.12	3.71
		API	.279	10.192	31.20	35.3	2.81	2.66	81.6	9.18	125.9	23.42	3.70
		30	.307	10.136	34.24	35.0	2.81	2.65	80.7	10.07	137.4	25.37	3.69
		API	.344	10.062	38.26	34.5	2.81	2.63	79.5	11.25	152.3	28.33	3.68
		Std.	.365	10.020	40.48	34.1	2.81	2.62	78.9	11.91	160.7	29.90	3.67
		API	.438	9.874	48.28	33.2	2.81	2.58	76.6	14.19	188.8	35.13	3.65
		X-Stg.	.500	9.750	54.74	32.3	2.81	2.55	74.7	16.10	212.0	39.43	3.63
		80	.594	9.562	64.40	31.1	2.81	2.50	71.8	18.91	244.9	45.56	3.60
		100	.719	9.312	77.00	29.5	2.81	2.44	68.1	22.62	286.2	53.25	3.56
		--	.750	9.250	80.10	29.1	2.81	2.42	67.2	23.56	296.2	55.10	3.54
		120	.844	9.062	89.20	27.9	2.81	2.37	64.5	26.23	324.3	60.34	3.51
		140	1.000	8.750	104.20	26.1	2.81	2.29	60.1	30.63	367.8	68.43	3.46
		160	1.125	8.500	116.00	24.6	2.81	2.22	56.7	34.01	399.4	74.31	3.43
12	12.750	12 Ga.	.104	12.542	14.1	53.6	3.34	3.28	123.5	4.13	82.6	12.9	4.47
		10 Ga.	.134	12.482	18.1	53.0	3.34	3.27	122.4	5.31	105.7	16.6	4.46
		8 Ga.	.164	12.422	22.1	52.5	3.34	3.25	121.2	6.48	128.4	20.1	4.45
		10S	.180	12.390	24.2	52.2	3.34	3.24	120.6	7.11	140.4	22.0	4.44
		6 Ga.	.194	12.362	26.0	52.0	3.34	3.23	120.0	7.65	150.9	23.7	4.44
		API	.203	12.344	27.2	52.0	3.34	3.23	119.9	7.99	157.2	24.7	4.43
		API	.219	12.312	29.3	51.7	3.34	3.22	119.1	8.52	167.6	26.3	4.43
		3 Ga.	.239	12.272	32.0	51.3	3.34	3.21	118.3	9.39	183.8	28.8	4.42
		20	.250	12.250	33.4	51.3	3.34	3.12	118.0	9.84	192.3	30.2	4.42
		API	.281	12.188	37.4	50.6	3.34	3.19	116.7	11.01	214.1	33.6	4.41
		API	.312	12.126	41.5	50.1	3.34	3.17	115.5	12.19	236.0	37.0	4.40
		30	.330	12.090	43.8	49.7	3.34	3.16	114.8	12.88	248.5	39.0	4.39
		API	.344	12.062	45.5	49.7	3.34	3.16	114.5	13.46	259.0	40.7	4.38
		Std.	.375	12.000	49.6	48.9	3.34	3.14	113.1	14.58	279.3	43.8	4.37
		40	.406	11.938	53.6	48.5	3.34	3.13	111.9	15.74	300.3	47.1	4.37
		API	.438	11.874	57.5	48.2	3.34	3.11	111.0	16.95	321.0	50.4	4.35
		X-Stg.	.500	11.750	65.4	46.9	3.34	3.08	108.4	19.24	361.5	56.7	4.33
		60	.562	11.626	73.2	46.0	3.34	3.04	106.2	21.52	400.5	62.8	4.31
		--	.625	11.500	80.9	44.9	3.34	3.01	103.8	23.81	438.7	68.8	4.29
		80	.688	11.374	88.6	44.0	3.34	2.98	101.6	26.03	475.2	74.6	4.27
		--	.750	11.250	96.2	43.1	3.34	2.94	99.4	28.27	510.7	80.1	4.25
		100	.844	11.062	108.0	41.6	3.34	2.90	96.1	31.53	561.8	88.1	4.22
		--	.875	11.000	110.9	41.1	3.34	2.88	95.0	32.64	578.5	90.7	4.21
		120	1.000	10.750	125.5	39.3	3.34	2.81	90.8	36.91	641.7	100.7	4.17
		140	1.125	10.500	140.0	37.5	3.34	2.75	86.6	41.08	700.7	109.9	4.13
		--	1.250	10.250	153.6	35.8	3.34	2.68	82.5	45.16	755.5	118.5	4.09
		160	1.312	10.126	161.0	34.9	3.34	2.65	80.5	47.14	781.3	122.6	4.07
		--	1.375	10.000	167.2	34.0	3.34	2.62	78.5	49.14	807.2	124.6	4.05
		--	1.500	9.750	180.4	32.4	3.34	2.55	74.7	53.01	853.8	133.9	4.01

**D-16**  
**(Continued). Properties of Pipe**

Nominal Pipe Size		Outside Diam. <b>D</b>	Wall Thickness Designa- tion	Inside Diam. <b>d</b>	Weight per Foot	Wt. of Water per Ft. of Pipe	Sq. Ft. Outside Surface per Ft.	Sq. Ft. Inside Surface per Ft.	Trans- verse Area in. <sup>2</sup>	Area of Metal in. <sup>2</sup>	Moment of Inertia in. <sup>4</sup>	Section Modulus in. <sup>3</sup>	Radius of Gyra- tion in.	
14	14.000	10 Ga. 8 Ga. 6 Ga.	.134	13.732	20	64.2	3.67	3.59	148.1	5.84	140.4	20.1	4.90	
			.164	13.672	24	43.6	3.67	3.58	146.8	7.13	170.7	24.4	4.89	
			.194	13.612	29	63.1	3.67	3.56	143.5	8.41	200.6	28.7	4.88	
		API	.210	13.580	31	62.8	3.67	3.55	144.8	9.10	216.2	30.9	4.87	
			.219	13.562	32	62.6	3.67	3.55	144.5	9.48	225.1	32.2	4.87	
			.239	13.522	35	62.3	3.67	3.54	143.6	10.33	244.9	35.0	4.87	
		10 API 20	.250	13.500	37	62.1	3.67	3.54	143.0	10.82	256.0	36.6	4.86	
			.281	13.438	41	61.5	3.67	3.52	141.8	12.11	285.2	40.7	4.85	
			.312	13.375	46	60.8	3.67	3.50	140.5	13.44	314.9	45.0	4.84	
		API Std. 40	.344	13.312	50	60.3	3.67	3.48	139.2	14.76	344.3	49.2	4.83	
			.375	13.250	55	59.7	3.67	3.47	137.9	16.05	372.8	53.2	4.82	
			.438	13.124	63	58.5	3.67	3.44	135.3	18.66	429.6	61.4	4.80	
		X-Stg. 60 -- .625	.500	13.000	72	57.4	3.67	3.40	132.7	21.21	483.8	69.1	4.78	
			.594	12.812	85	55.9	3.67	3.35	129.0	24.98	562.4	80.3	4.74	
			--	12.750	89	55.3	3.67	3.34	127.7	26.26	588.5	84.1	4.73	
			.750	12.500	107	53.2	3.67	3.27	122.7	31.22	687.5	98.2	4.69	
			--	12.250	123	51.1	3.67	3.21	117.9	36.08	780.1	111.4	4.65	
			100	.938	12.124	131	50.0	3.67	3.17	115.5	38.47	820.5	117.2	4.63
		-- 120 1.125	1.000	12.000	139	49.0	3.67	3.14	113.1	40.84	868.0	124.0	4.61	
			1.094	11.812	151	47.5	3.67	3.09	109.6	44.32	929.8	132.8	4.58	
			1.125	11.750	155	47.0	3.67	3.08	108.4	45.50	950.3	135.8	4.57	
		140 -- 160 1.500	1.250	11.500	171	45.0	3.67	3.01	103.9	50.07	1027.5	146.8	4.53	
			1.375	11.250	186	43.1	3.67	2.94	99.4	54.54	1099.5	157.1	4.49	
			1.406	11.188	190	42.6	3.67	2.93	98.3	55.63	1116.9	159.6	4.48	
			--	11.000	200	41.2	3.67	2.88	95.0	58.90	1166.5	166.6	4.45	
16	16.000	10 Ga. 8 Ga. --	.134	15.732	23	84.3	4.19	4.12	194.4	6.68	210	26.3	5.61	
			.164	15.672	28	63.6	4.19	4.10	192.9	8.16	256	32.0	5.60	
			--	15.624	32	83.3	4.19	4.09	192.0	9.39	294	36.7	5.59	
		6 Ga. API 3 Ga.	.194	15.612	33	83.0	4.19	4.09	191.4	9.63	301	37.6	5.59	
			.219	15.562	37	82.5	4.19	4.07	190.2	10.86	338	42.3	5.58	
			.239	15.522	40	82.0	4.19	4.06	189.2	11.83	368	45.9	5.57	
		10 API 20	.250	15.500	42	82.1	4.19	4.06	189.0	12.40	385	48.1	5.57	
			.281	15.438	47	81.2	4.19	4.04	187.0	13.90	430	53.8	5.56	
			.312	15.375	52	80.1	4.19	4.03	185.6	15.40	474	59.2	5.55	
		API Std. API	.344	15.312	57	80.0	4.19	4.01	184.1	16.94	519	64.9	5.54	
			.375	15.250	63	79.1	4.19	4.00	182.6	18.41	562	70.3	5.53	
			.438	15.124	73	78.2	4.19	3.96	180.0	21.42	650	81.2	5.51	
		X-Stg. 60 --	.500	15.000	83	76.5	4.19	3.93	176.7	24.35	732	91.5	5.48	
			.625	14.750	103	74.1	4.19	3.86	170.9	30.19	893	111.7	5.44	
			.656	14.688	108	73.4	4.19	3.85	169.4	31.62	933	116.6	5.43	
		-- 80 --	.750	14.500	122	71.5	4.19	3.80	165.1	35.93	1047	130.9	5.40	
			.844	14.312	137	69.7	4.19	3.75	160.9	40.14	1157	144.6	5.37	
			.875	14.250	141	69.1	4.19	3.73	159.5	41.58	1192	149.0	5.35	
		-- 100 --	1.000	14.000	160	66.7	4.19	3.66	153.9	47.12	1331	166.4	5.31	
			1.031	13.938	165	66.0	4.19	3.65	152.6	48.49	1366	170.7	5.30	
			--	13.750	179	64.4	4.19	3.60	148.5	52.57	1463	182.9	5.27	
		120 -- 1.375	1.219	13.562	193	62.6	4.19	3.55	144.5	56.56	1556	194.5	5.24	
			--	1.250	13.500	197	62.1	4.19	3.53	143.1	57.92	1586	198.3	5.23
			--	13.250	215	59.8	4.19	3.47	137.9	63.17	1704	213.0	5.19	
		140 -- 160	1.438	13.124	224	58.6	4.19	3.44	135.3	65.79	1761	220.1	5.17	
			--	1.500	13.000	232	57.4	4.19	3.40	132.7	68.33	1816	227.0	5.15
			--	1.594	12.812	243	55.9	4.19	3.35	129.0	72.10	1893	236.6	5.12

**D-16**  
**(Continued). Properties of Pipe**

Nominal		Pipe Size	Designation	Outside Diam.	Wall Thick-ness	Inside Diam.	Weight per Foot	Wt. of Water per Ft. of Pipe	Sq. Ft. Outside Surface per Ft.	Sq. Ft. Inside Surface per Ft.	Trans-verse Area in. <sup>2</sup>	Area of Metal in. <sup>2</sup>	Moment of Inertia in. <sup>4</sup>	Section Modulus in. <sup>3</sup>	Radius of Gyra-tion in.	
Nominal	Outside Diam.	D	d	A	I	Z	R									
18	18.000	10 Ga.	.134	17.732	26	107.1	4.71	4.64	246.9	7.52	300	33.4	6.32			
		8 Ga.	.164	17.672	31	106.3	4.71	4.63	245.3	9.19	366	40.6	6.31			
		6 Ga.	.194	17.612	37	105.6	4.71	4.61	243.6	10.85	430	47.8	6.29			
		3 Ga.	.239	17.522	45	104.5	4.71	4.59	241.1	13.34	526	58.4	6.28			
		10	.250	17.500	47	104.6	4.71	4.58	241.0	13.96	550	61.1	6.28			
		API	.281	17.438	49	104.0	4.71	4.56	240.0	14.49	570	63.4	6.27			
		20	.312	17.375	59	102.5	4.71	4.55	237.1	17.36	679	75.5	6.25			
		API	.344	17.312	65	102.0	4.71	4.53	235.4	19.08	744	82.6	6.24			
		Std.	.375	17.250	71	101.2	4.71	4.51	233.7	20.76	807	89.6	6.23			
		API	.406	17.188	76	100.6	4.71	4.50	232.0	22.44	869	96.6	6.22			
		30	.438	17.124	82	99.5	4.71	4.48	229.5	24.95	963	107.0	6.21			
		X-Sig.	.500	17.000	93	98.2	4.71	4.45	227.0	27.49	1053	117.0	6.19			
		40	.562	16.876	105	97.2	4.71	4.42	224.0	30.85	1177	130.9	6.17			
		--	.625	16.750	116	95.8	4.71	4.39	220.5	34.15	1290	143.2	6.14			
		--	.750	16.500	138	92.5	4.71	4.32	213.8	40.64	1515	168.3	6.10			
		--	.875	16.250	160	89.9	4.71	4.25	207.4	47.07	1730	192.3	6.06			
		--	.938	16.124	171	88.5	4.71	4.22	204.2	50.23	1834	203.8	6.04			
		--	1.000	16.000	182	87.2	4.71	4.19	201.1	53.41	1935	215.0	6.02			
		--	1.125	15.750	203	84.5	4.71	4.12	194.8	59.64	2133	237.0	5.98			
		100	1.156	15.688	208	83.7	4.71	4.11	193.3	61.18	2182	242.3	5.97			
		--	1.250	15.500	224	81.8	4.71	4.06	188.7	65.78	2319	257.7	5.94			
		120	1.375	15.250	244	79.2	4.71	3.99	182.7	71.82	2498	277.5	5.90			
		--	1.500	15.000	265	76.6	4.71	3.93	176.7	77.75	2668	296.5	5.86			
		140	1.562	14.876	275	75.3	4.71	3.89	173.8	80.66	2730	305.5	5.84			
		160	1.781	14.438	309	71.0	4.71	3.78	163.7	90.75	3020	335.5	5.77			
20	20.000	10 Ga.	.134	19.732	28	132.6	5.24	5.17	305.8	8.36	413	41.3	7.02			
		8 Ga.	.164	19.672	35	131.8	5.24	5.15	303.9	10.22	503	50.3	7.01			
		6 Ga.	.194	19.612	41	131.0	5.24	5.13	302.1	12.07	592	59.2	7.00			
		3 Ga.	.239	19.522	50	129.8	5.24	5.11	299.3	14.84	725	72.5	6.99			
		10	.250	19.500	53	130.0	5.24	5.11	299.0	15.52	759	75.9	6.98			
		API	.281	19.438	59	128.6	5.24	5.09	296.8	17.41	846	84.6	6.97			
		API	.312	19.374	66	128.1	5.24	5.08	295.0	19.36	937	93.7	6.95			
		API	.344	19.312	72	127.0	5.24	5.06	292.9	21.24	1026	102.6	6.95			
		Std.	.375	19.250	79	126.0	5.24	5.04	291.1	23.12	1113	111.3	6.94			
		API	.406	19.188	85	125.4	5.24	5.02	289.2	24.99	1200	120.0	6.93			
		API	.438	19.124	92	125.1	5.24	5.01	288.0	26.95	1290	129.0	6.92			
		X-Sig.	.500	19.000	105	122.8	5.24	4.97	283.5	30.63	1457	145.7	6.90			
		40	.594	18.812	123	120.4	5.24	4.93	278.0	36.15	1704	170.4	6.86			
		--	.625	18.750	129	119.5	5.24	4.91	276.1	38.04	1787	178.7	6.85			
		--	.812	18.376	167	114.9	5.24	4.81	265.2	48.95	2257	225.7	6.79			
		--	.875	18.250	179	113.2	5.24	4.78	261.6	52.57	2409	240.9	6.77			
		--	1.000	18.000	203	110.3	5.24	4.71	254.5	59.69	2702	270.2	6.73			
		--	1.031	17.938	209	109.4	5.24	4.80	252.7	61.44	2771	277.1	6.72			
		--	1.125	17.750	227	107.3	5.24	4.65	247.4	66.71	2981	298.1	6.68			
		--	1.250	17.500	250	104.3	5.24	4.58	240.5	73.63	3249	324.9	6.64			
		100	1.281	17.438	256	103.4	5.24	4.56	238.8	75.34	3317	331.7	6.63			
		--	1.375	17.250	274	101.3	5.24	4.52	233.7	80.45	3508	350.8	6.60			
		120	1.500	17.000	297	98.3	5.24	4.45	227.0	87.18	3755	375.5	6.56			
		140	1.750	16.500	342	92.6	5.24	4.32	213.8	100.33	4217	421.7	6.48			
		160	1.969	16.062	379	87.9	5.24	4.21	202.7	111.49	4586	458.6	6.41			

**D-16**  
**(Continued). Properties of Pipe**

Nominal		Designation	Wall Thickness d	Inside Diam. d	Weight per Foot	Wt. of Water per Ft. of Pipe	Sq. Ft. Outside Surface per Ft.	Sq. Ft. Inside Surface per Ft.	Transverse Area in. <sup>2</sup>	Area of Metal in. <sup>2</sup>	Moment of Inertia in. <sup>4</sup>	Section Modulus in. <sup>3</sup>	Radius of Gyration in.	
Pipe Size	Outside Diam.													
22	22.000	8 Ga.	.164	21.672	38	159.9	5.76	5.67	348.9	11.25	671	61.0	7.72	
		6 Ga.	.194	21.612	45	159.0	5.76	5.66	366.8	13.29	790	71.8	7.71	
		3 Ga.	.239	21.522	56	157.7	5.76	5.63	363.8	16.34	967	87.9	7.69	
		API	.250	21.500	58	157.4	5.76	5.63	363.1	17.18	1010	91.8	7.69	
		API	.281	21.438	65	156.5	5.76	5.61	361.0	19.17	1131	102.8	7.68	
		API	.312	21.376	72	155.6	5.76	5.60	358.9	21.26	1250	113.6	7.67	
		API	.344	21.312	80	154.7	5.76	5.58	356.7	23.40	1373	124.8	7.66	
		API	.375	21.250	87	153.7	5.76	5.56	354.7	25.48	1490	135.4	7.65	
		API	.406	21.188	94	152.9	5.76	5.55	352.6	27.54	1607	146.1	7.64	
		API	.438	21.124	101	151.9	5.76	5.53	350.5	29.67	1725	156.8	7.62	
		API	.500	21.000	115	150.2	5.76	5.50	346.4	33.77	1953	177.5	7.61	
		--	.625	20.750	143	146.6	5.76	5.43	338.2	41.97	2400	218.2	7.56	
		--	.750	20.500	170	143.1	5.76	5.37	330.1	50.07	2829	257.2	7.52	
		--	.875	20.250	198	139.6	5.76	5.30	322.1	58.07	3245	295.0	7.47	
		--	1.000	20.000	224	136.2	5.76	5.24	314.2	65.97	3645	331.4	7.43	
		--	1.125	19.750	251	132.8	5.76	5.17	306.4	73.78	4029	366.3	7.39	
		--	1.250	19.500	277	129.5	5.76	5.10	298.6	81.48	4400	400.0	7.35	
		--	1.375	19.250	303	126.2	5.76	5.04	291.0	89.09	4758	432.6	7.31	
		--	1.500	19.000	329	122.9	5.76	4.97	283.5	96.60	5103	463.9	7.27	
24	24.000	8 Ga.	.164	23.672	42	190.8	6.28	6.20	440.1	12.28	872	72.7	8.43	
		6 Ga.	.194	23.612	49	189.8	6.28	6.18	437.9	14.51	1028	85.7	8.42	
		3 Ga.	.239	23.522	61	188.4	6.28	6.16	434.5	17.84	1260	105.0	8.40	
		10	.250	23.500	63	189.0	6.28	6.15	435.0	18.67	1320	110.0	8.40	
		API	.281	23.438	71	187.0	6.28	6.14	431.5	20.94	1472	122.7	8.38	
		API	.312	23.376	79	186.9	6.28	6.12	430.0	23.20	1630	136.0	8.38	
		API	.344	23.312	87	185.0	6.28	6.10	426.8	25.57	1789	149.1	8.36	
		Std.	.375	23.250	95	183.8	6.28	6.09	424.6	27.83	1942	161.9	8.33	
		API	.406	23.188	102	183.1	6.28	6.07	422.3	30.09	2095	174.6	8.34	
		API	.438	23.124	110	182.1	6.28	6.05	420.0	32.42	2252	187.7	8.33	
		X-Sig.	.500	23.000	125	181.0	6.28	6.02	416.0	36.90	2550	213.0	8.31	
		30	.562	22.876	141	178.5	6.28	5.99	411.0	41.40	2840	237.0	8.28	
		--	.625	22.750	156	175.9	6.28	5.98	406.5	45.90	3137	261.4	8.27	
		40	.688	22.624	171	174.2	6.28	5.92	402.1	50.30	3422	285.2	8.25	
		--	.750	22.500	186	172.1	6.28	5.89	397.6	54.78	3705	308.8	8.22	
		--	.875	22.250	216	168.6	6.28	5.82	388.8	63.57	4257	354.7	8.18	
		60	.969	22.062	238	165.8	6.28	5.78	382.3	70.04	4652	387.7	8.15	
		--	1.000	22.000	246	164.8	6.28	5.74	380.1	72.26	4788	399.0	8.14	
		--	1.125	21.750	275	161.1	6.28	5.69	371.5	80.85	5302	441.8	8.10	
		80	1.219	21.562	297	158.2	6.28	5.65	365.2	87.17	5673	472.8	8.07	
		--	1.250	21.500	304	157.4	6.28	5.63	363.1	89.34	5797	483.0	8.05	
		--	1.375	21.250	332	153.8	6.28	5.56	354.7	97.73	6275	522.9	8.01	
		--	1.500	21.000	361	150.2	6.28	5.50	346.4	106.03	6740	561.7	7.97	
		100	1.531	20.938	367	149.3	6.28	5.48	344.3	108.07	6847	570.6	7.96	
		--	120	1.812	20.376	429	141.4	6.28	5.33	326.1	126.30	7023	651.9	7.87
		140	2.062	19.876	484	134.4	6.28	5.20	310.3	142.10	8427	718.9	7.79	
		--	160	2.344	19.312	542	127.0	6.28	5.06	293.1	159.40	9457	788.1	7.70
26	26.000	8 Ga.	.164	25.672	45	224.4	6.81	6.72	517.6	13.31	1111	85.4	9.13	
		6 Ga.	.194	25.612	54	223.4	6.81	6.70	515.2	15.73	1310	100.7	9.12	
		3 Ga.	.239	25.522	66	221.8	6.81	6.68	511.6	19.34	1605	123.4	9.11	
		API	.250	25.500	67	221.4	6.81	6.68	510.7	19.85	1646	126.6	9.10	
		API	.281	25.438	77	220.3	6.81	6.66	508.2	22.70	1877	144.4	9.09	
		API	.312	25.376	84	219.2	6.81	6.64	505.8	25.18	2076	159.7	9.08	

**D-16**  
**(Continued). Properties of Pipe**

Nominal		Designation	Outside Diam.	Wall Thickness	Inside Diam. d	Weight per Foot	Wt. of Water per Ft. of Pipe	Sq. Ft. Outside Surface per Ft.	Sq. Ft. Inside Surface per Ft.	Transverse Area in. <sup>2</sup>	Area of Metal in. <sup>2</sup>	Moment of Inertia in. <sup>4</sup>	Section Modulus in. <sup>3</sup>	Radius of Gyration in.
Pipe Size	D													
26 cont.	26.000	API	.344	25.312	94	218.2	6.81	6.63	503.2	27.73	2280	175.4	9.07	
		API	.373	25.250	103	217.1	6.81	6.61	500.7	30.19	2478	190.6	9.06	
		API	.406	25.188	111	216.0	6.81	6.59	498.3	32.64	2673	205.6	9.05	
		API	.438	25.124	120	214.9	6.81	6.58	495.8	35.17	2874	221.1	9.04	
		API	.500	25.000	136	212.8	6.81	6.54	490.9	40.06	3259	250.7	9.02	
		--	.625	24.750	149	208.6	6.81	6.48	481.1	49.82	4013	308.7	8.98	
		--	.750	24.500	202	204.4	6.81	6.41	471.4	59.49	4744	364.9	8.93	
		--	.875	24.250	235	200.2	6.81	6.35	461.9	69.07	5458	419.9	8.89	
		--	1.000	24.000	267	196.1	6.81	6.28	452.4	78.54	6149	473.0	8.85	
		--	1.125	23.750	299	192.1	6.81	6.22	443.0	87.91	6813	524.1	8.80	
		--	1.375	23.250	362	184.1	6.81	6.09	424.6	106.37	8088	622.2	8.72	
		--	1.500	23.000	393	180.1	6.81	6.02	415.3	115.45	8695	688.8	8.68	
30	30.000	8 Ga.	.164	29.672	52	299.9	7.85	7.77	691.4	15.37	1711	114.0	10.55	
		6 Ga.	.194	29.612	62	298.6	7.85	7.75	688.6	18.17	2017	134.4	10.53	
		3 Ga.	.239	29.522	76	296.7	7.85	7.73	684.4	22.35	2474	165.0	10.52	
		API	.250	29.500	79	296.3	7.85	7.72	683.4	23.37	2585	172.3	10.52	
		API	.281	29.438	89	295.1	7.85	7.70	680.5	26.24	2897	193.1	10.51	
		10	.312	29.376	99	293.7	7.85	7.69	677.8	29.19	3201	213.4	10.50	
		API	.344	29.312	109	292.6	7.85	7.67	674.8	32.04	3524	235.0	10.49	
		API	.375	29.250	119	291.2	7.85	7.66	672.0	34.90	3823	254.8	10.48	
		API	.406	29.188	130	290.7	7.85	7.64	669.0	37.75	4132	275.5	10.46	
		API	.438	29.124	138	288.8	7.85	7.62	666.1	40.68	4442	296.2	10.45	
		20	.500	29.000	158	286.2	7.85	7.59	660.5	46.34	5033	335.5	10.43	
32	32.000	30	.625	28.750	194	281.3	7.85	7.53	649.2	57.68	6213	414.2	10.39	
		--	.750	28.500	234	276.6	7.85	7.46	637.9	68.92	7371	491.4	10.34	
		--	.875	28.250	272	271.8	7.85	7.39	620.7	80.06	8494	566.2	10.30	
		--	1.000	28.000	310	267.0	7.85	7.33	615.7	91.11	9591	639.4	10.26	
		--	1.125	27.750	347	262.2	7.85	7.26	604.7	102.05	10653	710.2	10.22	
		--	1.250	27.500	384	257.5	7.85	7.20	593.9	112.90	11682	778.8	10.17	
		--	1.375	27.250	421	252.9	7.85	7.13	583.1	123.65	12694	846.2	10.13	
		--	1.500	27.000	457	248.2	7.85	7.07	572.5	134.30	13673	911.5	10.09	
		API	.250	31.500	85	337.8	8.38	8.25	779.2	24.93	3141	196.3	11.22	
		API	.281	31.438	95	336.5	8.38	8.23	776.2	28.04	3525	220.3	11.21	
		API	.312	31.376	106	335.2	8.38	8.21	773.2	31.02	3891	243.2	11.20	
32	32.000	API	.344	31.312	116	333.8	8.38	8.20	770.0	34.24	4287	268.0	11.19	
		API	.375	31.250	127	332.5	8.38	8.18	766.9	37.25	4656	291.0	11.18	
		API	.406	31.188	137	331.2	8.38	8.16	764.0	40.29	5025	314.1	11.17	
		API	.438	31.124	143	329.8	8.38	8.15	760.8	43.43	5407	337.9	11.16	
		API	.500	31.000	168	327.2	8.38	8.11	754.7	49.48	6140	381.8	11.14	
		--	.625	30.750	209	321.9	8.38	8.05	742.5	61.59	7578	473.6	11.09	
		--	.750	30.500	250	316.7	8.38	7.98	730.5	73.63	8990	561.9	11.05	
		--	.875	30.250	291	311.5	8.38	7.92	718.6	85.53	10348	648.0	11.01	
		--	1.000	30.000	331	306.4	8.38	7.85	706.8	97.38	11680	730.0	10.95	
		--	1.125	29.750	371	301.3	8.38	7.79	695.0	109.0	13003	812.7	10.92	
		--	1.250	29.500	410	296.3	8.38	7.72	680.5	120.7	14398	899.9	10.88	
		--	1.375	29.250	450	291.2	8.38	7.66	671.9	132.2	15524	970.4	10.84	
		--	1.500	29.000	489	286.3	8.38	7.59	660.5	143.7	16752	1047.0	10.80	

**D-16**  
**(Continued). Properties of Pipe**

Nominal		Designation	Wall Thickness D	Inside Diam. d	Weight per Foot	Wt. of Water per Ft. of Pipe	Sq. Ft. Outside Surface per Ft.	Sq. Ft. Inside Surface per Ft.	Transverse Area in. <sup>2</sup>	Area of Metal in. <sup>2</sup>	Moment of Inertia in. <sup>4</sup>	Section Modulus in. <sup>3</sup>	Radius of Gyration in.
Pipe Size	Outside Diam.												
34	34.000	API	.250	33.500	.90	382.0	8.90	8.77	881.2	26.50	3773	221.9	11.93
		API	.281	33.438	101	380.7	8.90	8.75	878.2	29.77	4230	248.8	11.92
		API	.312	33.376	112	379.3	8.90	8.74	874.9	32.99	4680	275.3	11.91
		API	.344	33.312	124	377.8	8.90	8.72	871.6	36.36	5147	302.8	11.90
		API	.375	33.250	135	376.2	8.90	8.70	867.8	39.61	5597	329.2	11.89
		API	.406	33.188	146	375.0	8.90	8.69	865.0	42.88	6047	355.7	11.87
		API	.438	33.124	157	373.6	8.90	8.67	861.7	46.18	6501	382.4	11.86
		API	.500	33.000	179	370.8	8.90	8.64	858.3	52.62	7385	434.4	11.85
		--	.625	32.750	223	365.0	8.90	8.57	841.9	65.53	9124	536.7	11.80
		--	.750	32.500	266	359.5	8.90	8.51	829.3	78.34	10829	637.0	11.76
		--	.875	32.250	308	354.1	8.90	8.44	816.8	90.66	12442	731.9	11.71
		--	1.000	32.000	353	348.6	8.90	8.38	804.2	103.6	14114	830.2	11.67
		--	1.125	31.750	395	343.2	8.90	8.31	791.6	116.1	15703	923.7	11.63
		--	1.250	31.500	437	337.8	8.90	8.25	779.2	128.5	17246	1014.5	11.58
		--	1.375	31.250	479	332.4	8.90	8.18	766.9	140.9	18770	1104.1	11.54
		--	1.500	31.000	521	327.2	8.90	8.11	754.7	153.1	20247	1191.0	11.50
		--	.164	35.672	63	433.2	9.42	9.34	999.3	18.53	2975	165.3	12.67
		--	.194	35.612	74	431.8	9.42	9.32	996.0	21.83	3499	194.4	12.66
		--	.239	35.522	91	429.6	9.42	9.30	991.0	26.86	4293	238.5	12.64
		API	.250	35.500	96	429.1	9.42	9.29	989.7	28.11	4491	249.5	12.64
		API	.281	35.438	107	427.6	9.42	9.28	986.4	31.49	5023	279.1	12.63
		API	.312	35.376	119	426.1	9.42	9.26	982.9	34.95	5565	309.1	12.62
		API	.344	35.312	131	424.6	9.42	9.24	979.3	38.56	6127	340.4	12.60
		API	.375	35.250	143	423.1	9.42	9.23	975.8	42.01	6664	370.2	12.59
		API	.406	35.188	154	421.6	9.42	9.21	972.5	45.40	7191	399.5	12.58
		API	.438	35.124	166	420.1	9.42	9.19	968.9	48.93	7737	429.9	12.57
		API	.500	35.000	190	417.1	9.42	9.16	962.1	55.76	8785	488.1	12.55
		--	.625	34.750	236	411.1	9.42	9.10	948.3	69.50	10872	604.0	12.51
		--	.750	34.500	282	405.3	9.42	9.03	934.7	83.01	12898	716.5	12.46
		--	.875	34.250	329	399.4	9.42	8.97	921.2	96.60	14906	828.1	12.42
		--	1.000	34.000	374	393.6	9.42	8.90	907.9	109.9	16851	936.2	12.38
		--	1.125	33.750	419	387.8	9.42	8.83	894.5	123.3	18766	1042.6	12.34
		--	1.250	33.500	464	382.1	9.42	8.77	881.3	136.5	20624	1145.8	12.29
		--	1.375	33.250	509	376.4	9.42	8.70	868.2	149.6	22451	1247.3	12.25
		--	1.500	33.000	553	370.8	9.42	8.64	855.3	162.4	24237	1346.5	12.21
		--	.250	41.500	112	586.4	10.99	10.86	1352.6	32.82	7126	339.3	14.73
		--	.375	41.250	167	579.3	10.99	10.80	1336.3	49.08	10627	506.1	14.71
		--	.500	41.000	222	572.3	10.99	10.73	1320.2	65.18	14037	668.4	14.67
		--	.625	40.750	276	565.4	10.99	10.67	1304.1	81.28	17373	827.3	14.62
		--	.750	40.500	331	558.4	10.99	10.60	1288.2	97.23	20689	985.2	14.59
		--	.875	40.250	385	551.6	10.99	10.54	1272.3	113.0	23896	1137.9	14.54
		--	1.000	40.000	438	544.8	10.99	10.47	1256.6	128.8	27080	1289.5	14.50
		--	1.125	39.750	492	537.9	10.99	10.41	1240.9	144.5	30193	1437.8	14.45
		--	1.250	39.500	544	531.2	10.99	10.34	1225.3	160.0	33233	1582.5	14.41
		--	1.375	39.250	597	524.4	10.99	10.27	1209.9	175.5	36240	1725.7	14.37
		--	1.500	39.000	649	517.9	10.99	10.21	1194.5	190.8	39181	1865.7	14.33

## D-16

### Equation of Pipes

The table below gives the number of pipes of one size required to equal in delivery other larger pipes of same length and under same conditions. The upper portion above the diagonal line of stars pertains to "standard" steam and gas pipes, while the lower portion is for pipes of the ACTUAL internal diameters given. The figures given in the table opposite the intersection of any two sizes is the number of the smaller-sized pipes required to equal one of the larger. Thus, it requires 20 standard 2-inch pipes to equal one standard 7-inch pipe.

#### STANDARD STEAM AND GAS PIPES

Dia.	$\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{1}{2}$	2	$2\frac{1}{4}$	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	Dia.
$\frac{5}{8}$	***	2.27	4.88	15.8	31.7	52.9	96.9	205	377	620	918	1292	1767	2484	3014	3796	4904	5927	7321	8535	9717	$\frac{1}{2}$
$\frac{3}{4}$	2.60	** *	2.05	6.97	14.0	23.3	42.5	90.4	166	273	405	569	779	1096	1328	1608	2161	2615	3226	3761	4282	$\frac{3}{4}$
1	7.55	2.00	** *	3.45	6.82	11.4	20.9	44.1	81.1	133	198	278	380	536	649	815	1070	1263	1576	1837	2092	1
$1\frac{1}{2}$	24.2	9.30	3.20	* **	1.26	3.34	6.13	13.0	23.8	39.2	58.1	81.7	112	157	190	239	310	375	463	539	614	$1\frac{1}{2}$
2	54.3	21.0	7.25	2.26	* **	1.07	3.06	6.47	11.9	19.6	29.0	40.8	55.8	78.5	95.1	119	155	187	231	269	307	2
$2\frac{1}{2}$	102	39.4	13.6	4.23	1.87	* **	1.83	3.87	7.12	11.7	17.4	24.4	33.4	47.0	56.9	71.5	92.6	112	138	161	184	$2\frac{1}{2}$
3	170	65.4	22.6	7.03	3.11	1.65	* **	2.12	3.89	6.39	9.48	13.3	20.0	23.7	31.2	39.1	50.6	61.1	75.5	88.0	100	3
4	376	144	49.8	15.5	6.87	3.07	2.21	* **	1.84	3.02	4.48	6.30	8.81	12.1	14.7	18.5	23.9	28.0	35.7	41.6	47.4	4
5	686	263	90.9	28.3	12.5	6.70	4.03	1.83	* **	1.65	2.44	3.43	4.69	6.00	8.00	10.0	13.0	15.7	19.4	22.6	25.8	5
6	1116	429	148	46.0	20.4	10.9	6.56	2.97	1.63	* **	1.48	2.08	2.85	4.02	4.96	6.11	7.91	9.56	11.8	13.8	15.6	6
7	1707	656	226	70.5	31.2	16.6	10.0	4.54	2.49	1.51	* **	1.41	1.93	2.71	3.28	4.12	5.34	6.45	7.97	9.31	10.6	7
8	2435	936	322	101	44.5	23.8	14.3	6.48	3.54	2.18	1.43	* **	1.35	1.93	2.53	3.79	4.57	5.67	6.80	7.52	8	
9	3335	1281	440	137	60.8	32.5	19.5	8.85	4.85	2.98	1.95	1.37	* **	1.41	1.71	2.14	2.77	3.35	4.14	4.83	5.50	9
10	4393	1688	582	181	80.4	42.9	25.8	11.7	6.40	3.93	2.57	1.80	1.32	* **	1.21	1.52	1.97	2.38	2.94	3.43	3.91	10
11	5642	2168	747	237	103	55.1	33.1	15.0	8.22	5.05	3.31	2.32	1.70	1.28	* **	1.26	1.63	1.88	2.43	2.83	3.22	11
12	7087	2721	938	293	129	69.2	41.6	18.8	10.3	6.34	4.15	2.91	2.13	1.61	1.26	* **	1.30	1.57	1.93	2.26	2.58	12
13	8657	3326	1146	358	158	84.5	50.7	23.0	12.0	7.75	5.07	3.50	2.60	1.98	1.53	1.22	* **	1.21	1.49	1.74	1.98	13
14	10600	4070	1403	438	193	103	62.2	28.2	15.4	9.48	6.21	4.35	3.18	2.41	1.88	1.50	1.22	* **	1.24	1.44	1.64	14
15	12824	4927	1698	530	234	125	75.3	34.1	18.7	11.5	7.52	5.27	3.85	2.92	2.27	1.81	1.48	1.21	* **	1.17	1.35	15
16	14978	5758	1984	619	274	146	88.0	39.9	21.8	13.4	8.78	6.15	4.51	3.41	2.06	2.12	1.73	1.42	1.18	* **	1.14	16
17	17537	6738	2322	724	320	171	103	46.6	25.6	15.7	10.3	7.20	5.27	3.99	3.11	2.47	2.03	1.66	1.37	1.17	* **	17
18	20327	7810	2691	840	317	198	119	54.1	29.6	18.2	11.9	8.35	6.11	4.63	3.60	2.87	2.35	1.92	1.59	1.36	1.16	18
20	26676	10249	3532	1102	487	260	157	70.9	38.9	23.9	15.6	10.9	8.02	6.07	4.73	3.76	3.08	2.52	2.08	1.78	1.52	20
24	42624	16326	5644	1761	778	416	250	113	62.1	38.2	25.0	17.5	12.8	9.70	7.55	6.01	4.92	4.09	3.32	2.84	2.43	24
30	75453	28090	9990	3117	1378	736	443	201	110	67.6	44.2	31.0	22.7	17.2	13.4	10.7	8.72	7.14	5.88	6.03	4.30	30
36	120100	46143	15802	4961	2193	1172	705	319	175	108	70.4	49.3	36.1	27.3	21.3	16.9	13.9	11.3	9.37	8.01	6.85	36
42	177724	68282	23524	7341	3245	1734	1044	473	259	159	104	73.0	53.4	40.5	31.5	25.1	20.5	16.8	13.9	11.9	10.1	42
48	249351	95818	33020	10301	4554	2434	1465	663	363	223	146	102	75.0	56.8	44.2	35.2	28.8	23.5	19.4	16.6	14.2	48
Dia.	$\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{1}{2}$	2	$2\frac{1}{4}$	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	

By permission, Buffalo Tank Div., Bethlehem Steel Corp.

## D-18

(Continued). Circumferences and Areas of Circles  
(Advancing of eighths)

Dia.	Circum.	Area*	Dia.	Circum.	Area*	Dia.	Circum.	Area*
29 $\frac{5}{8}$	93.070	689.30	36.	113.097	1017.9	42 $\frac{3}{8}$	132.732	1402.0
30 $\frac{1}{8}$	93.182	689.13	37.	113.490	1026.0	43 $\frac{1}{8}$	133.126	1410.3
30 $\frac{5}{8}$	93.855	700.98	38.	113.883	1032.1	44 $\frac{3}{8}$	133.516	1418.6
31.	94.248	708.86	39.	114.275	1039.2	45 $\frac{1}{8}$	133.910	1427.0
31 $\frac{5}{8}$	94.640	712.76	40.	114.668	1046.3	46 $\frac{3}{8}$	134.303	1435.4
31 $\frac{1}{8}$	95.033	718.69	41.	115.061	1053.5	47 $\frac{1}{8}$	134.696	1443.8
31 $\frac{5}{8}$	95.426	724.61	42.	115.454	1060.7	48.	135.058	1452.2
31 $\frac{1}{8}$	95.819	730.62	43.	115.846	1068.0	49.	135.451	1460.7
31 $\frac{5}{8}$	96.211	738.62	44.	116.239	1075.2	50.	135.844	1469.1
31 $\frac{1}{8}$	96.604	742.64	45.	116.632	1082.5	51.	136.237	1477.6
31 $\frac{5}{8}$	96.997	748.69	46.	117.024	1089.8	52.	136.630	1485.3
31 $\frac{1}{8}$	97.389	754.77	47.	117.417	1097.1	53.	137.023	1494.7
31 $\frac{5}{8}$	97.782	760.87	48.	117.810	1104.5	54.	137.416	1503.3
31 $\frac{1}{8}$	98.175	766.98	49.	118.202	1111.8	55.	137.809	1511.9
31 $\frac{5}{8}$	98.567	773.14	50.	118.595	1119.2	56.	138.202	1520.5
31 $\frac{1}{8}$	99.950	779.31	51.	119.381	1134.5	57.	138.595	1529.1
31 $\frac{5}{8}$	99.353	785.51	52.	119.773	1141.6	58.	139.088	1537.9
31 $\frac{1}{8}$	99.746	791.73	53.	120.166	1149.1	59.	139.481	1546.5
31 $\frac{5}{8}$	100.138	797.98	54.	120.559	1156.6	60.	140.184	1554.0
32.	100.531	804.25	55.	121.314	1164.2	61.	140.586	1572.8
32 $\frac{1}{8}$	100.924	810.54	56.	121.737	1171.7	62.	141.372	1590.4
32 $\frac{5}{8}$	101.316	816.86	57.	122.129	1179.3	63.	142.157	1608.8
32 $\frac{1}{8}$	101.709	823.21	58.	122.522	1194.8	64.	142.550	1617.0
32 $\frac{5}{8}$	102.102	829.58	59.	122.915	1202.3	65.	143.335	1634.9
32 $\frac{1}{8}$	102.494	835.97	60.	123.308	1210.6	66.	144.121	1652.9
32 $\frac{5}{8}$	102.887	842.39	61.	123.700	1217.7	67.	144.513	1661.9
32 $\frac{1}{8}$	103.280	848.53	62.	124.093	1225.4	68.	144.906	1670.9
33.	103.673	855.30	63.	124.486	1232.1	69.	145.299	1680.0
33 $\frac{1}{8}$	104.066	861.79	64.	124.878	1239.2	70.	145.691	1689.1
33 $\frac{5}{8}$	104.458	868.31	65.	125.271	1241.0	71.	146.084	1698.2
33 $\frac{1}{8}$	104.851	874.85	66.	125.664	1248.8	72.	146.477	1707.4
33 $\frac{5}{8}$	105.243	881.41	67.	126.057	1255.5	73.	146.868	1716.5
33 $\frac{1}{8}$	105.636	888.00	68.	126.450	1262.3	74.	147.262	1725.7
33 $\frac{5}{8}$	106.029	894.62	69.	126.843	1269.2	75.	147.656	1734.9
33 $\frac{1}{8}$	106.421	901.26	70.	127.235	1276.2	76.	148.048	1744.2
34.	106.814	907.92	71.	127.627	1283.0	77.	148.440	1753.5
34 $\frac{1}{8}$	107.207	914.51	72.	128.020	1304.2	78.	148.833	1762.7
34 $\frac{5}{8}$	107.600	921.32	73.	128.413	1312.2	79.	149.226	1772.1
34 $\frac{1}{8}$	107.993	928.06	80.	129.805	1320.3	81.	149.618	1781.4
34 $\frac{5}{8}$	108.385	934.82	81.	129.198	1328.3	82.	150.011	1790.8
34 $\frac{1}{8}$	108.778	941.61	82.	129.591	1336.4	83.	150.404	1800.1
34 $\frac{5}{8}$	109.170	948.42	83.	130.181	1360.0	84.	150.796	1809.6
34 $\frac{1}{8}$	109.563	955.26	84.	130.574	1377.2	85.	151.189	1819.0
35.	109.956	962.11	85.	130.967	1344.5	86.	151.582	1828.6
35 $\frac{1}{8}$	110.348	969.00	86.	130.360	1352.7	87.	151.975	1837.9
35 $\frac{5}{8}$	110.741	975.91	87.	130.753	1360.8	88.	152.367	1847.6
35 $\frac{1}{8}$	111.134	982.84	88.	131.146	1369.0	89.	152.759	1857.0
35 $\frac{5}{8}$	111.527	989.60	90.	131.539	1377.2	91.	153.152	1866.4
35 $\frac{1}{8}$	111.919	996.78	91.	131.932	1385.4	92.	153.545	1875.8
35 $\frac{5}{8}$	112.312	1003.8	92.	132.325	1393.7	93.	153.937	1885.2
35 $\frac{1}{8}$	112.705	1010.8	94.	132.340	1393.7	95.	154.330	1894.6

\*Approximate area, sufficiently accurate for practical purposes, including estimating.

## D-18

(Continued). Circumferences and Areas of Circles  
(Advancing of eighths)

Dia.	Circum.	Area*	Dia.	Circum.	Area*	Dia.	Circum.	Area*
48 $\frac{1}{8}$	152.700	1857.0	55.	172.788	2375.8	61 $\frac{3}{4}$	192.423	2948.5
48 $\frac{5}{8}$	153.153	1866.6	56.	173.180	2386.6	62 $\frac{1}{4}$	192.815	2953.5
48 $\frac{1}{8}$	153.545	1876.1	57.	173.573	2397.3	63 $\frac{3}{4}$	193.208	2959.6
49.	153.938	1885.7	58.	174.358	2410.2	64 $\frac{1}{4}$	193.599	2964.8
49 $\frac{1}{8}$	154.331	1895.4	59.	174.751	2420.1	65 $\frac{3}{4}$	194.086	2969.9
49 $\frac{5}{8}$	154.723	1905.0	60.	175.144	2441.1	66 $\frac{1}{4}$	194.479	2974.0
49 $\frac{1}{8}$	155.116	1914.7	61.	175.536	2452.0	67.	194.779	2979.1
49 $\frac{5}{8}$	155.509	1924.4	62.	176.929	2463.0	68.	195.171	2983.3
49 $\frac{1}{8}$	155.902	1934.2	69.	177.322	2474.0	69.	195.564	2988.5
50.	157.080	1983.5	70.	177.600	2485.2	70.	196.350	2992.6
50 $\frac{1}{8}$	157.472	1973.3	71.	177.893	2518.3	71.	196.742	2996.3
50 $\frac{5}{8}$	157.863	1993.2	72.	178.285	2529.4	72.	197.135	3004.0
50 $\frac{1}{8}$	158.256	1993.1	73.	178.678	2640.6	73.	197.920	3117.2
51.	160.221	2042.8	74.	180.642	2500.7	74.	200.277	3121.0
51 $\frac{1}{8}$	160.614	2052.8	75.	181.034	2605.0	75.	200.669	3204.4
51 $\frac{5}{8}$	161.007	2062.9	76.	181.427	2610.4	76.	201.062	3217.0
51 $\frac{1}{8}$	161.399	2072.0	77.	181.820	2630.7	77.	201.455	3220.8
51 $\frac{5}{8}$	161.792	2082.1	78.	182.212	2642.1	78.	201.847	3224.2
51 $\frac{1}{8}$	162.185	2092.3	79.	182.605	2653.5	79.	202.240	3234.8
52.	163.363	2123.7	80.	183.390	2676.4	80.	202.633	3280.1
52 $\frac{1}{8}$	163.756	2133.9	81.	183.783	2687.6	81.	203.418	3292.8
52 $\frac{5}{8}$	164.148	2144.2	82.	184.176	2699.3	82.	203.811	3305.6
52 $\frac{1}{8}$	164.541	2154.5	83.	184.569	2710.9	83.	204.204	3318.3
52 $\frac{5}{8}$	164.934	2164.8	84.	184.961	2722.4	84.	204.596	3331.1
52 $\frac{1}{8}$	165.326	2175.1	85.	185.354	2734.0	85.	204.989	3343.9
52 $\frac{5}{8}$	165.719	2185.4	86.	185.747	2745.6	86.	205.382	3356.7
52 $\frac{1}{8}$	166.112	2195.8	87.	186.139	2757.2	87.	205.774	3369.6
53.	166.504	2206.2	88.	186.532	2768.6	88.	206.167	3382.4
53 $\frac{1}{8}$	166.897	2216.0	89.	186.925	2780.5	89.	206.560	3395.3
53 $\frac{5}{8}$	167.290	2227.0	90.	187.317	2792.2	90.	206.952	3408.2
53 $\frac{1}{8}$	167.683	2237.5	91.	187.710	2803.9	91.	207.345	3421.2
53 $\frac{5}{8}$	168.075	2248.0	92.	188.103	2815.7	92.	207.738	3434.2
53 $\frac{1}{8}$	168.468	2258.5	93.	188.496	2827.4	93.	208.131	3447.2
53 $\frac{5}{8}$	168.861	2268.1	94.	188.888	2839.2	94.	208.523	3460.2
53 $\frac{1}{8}$	169.253	2278.6	95.	189.281	2851.0	95.	208.916	3473.2
54.	169.646	2290.2	96.	189.674	2862.9	96.	209.300	3486.3
54 $\frac{1}{8}$	170.030	2300.8	97.	190.066	2874.6	97.	209.701	3499.4
54 $\frac{5}{8}$	170.421	2311.5	98.	190.459	2886.8	98.	210.094	3512.5
54 $\frac{1}{8}$	170.814	2322.1	99.	190.852	2898.6	99.	210.487	3525.7
54 $\frac{5}{8}$	171.207	2332.8	100.	191.244	2910.5	100.	210.870	3538.8
54 $\frac{1}{8}$	171.600	2343.5	101.	191.637	2922.5	101.	211.272	3552.0
54 $\frac{5}{8}$	172.002	2353.0	102.	192.030	2934.6	102.	211.665	3565.2

\*Approximate area, sufficiently accurate for practical purposes, including estimating.

## D-18

**(Continued). Circumferences and Areas of Circles  
(Advancing of eighths)**

Dia.	Circum.	Area*	Dia.	Circum.	Area*	Dia.	Circum.	Area*
1	.01009	.00018	216	6.6750	3.5466	516	10.886	22.691
2	.00178	.00077	214	6.6722	3.5383	514	10.882	22.221
3	.14726	.00173	212	7.0686	3.9781	512	11.270	23.758
4	.10635	.00307	210	7.2849	4.2000	510	11.746	24.301
5	.29452	.00600	208	7.4613	4.4301	508	11.871	24.850
6	.30370	.01237	206	7.6576	4.6664	506	11.803	25.408
7	.19087	.01917	204	7.5640	4.9087	504	11.064	25.987
8	.58005	.02761	202	8.0103	5.1572	502	11.201	26.535
9	.85722	.03758	200	8.2487	5.4119	500	11.457	27.109
10	.78540	.04808	198	8.4430	5.6727	498	11.653	27.688
11	.88357	.06313	196	8.8857	6.2126	6	18.880	28.274
12	.98175	.07870	194	9.0321	6.4918	16	11.242	29.465
13	1.07100	.08281	192	9.2284	6.7771	14	11.635	30.880
14	1.1781	.11045	190	11.1045	7.0086	12	20.028	31.918
15	1.2703	.12962	188	12.962	7.3062	10	20.420	33.183
16	1.3744	.15033	186	15.033	7.6099	8	21.208	34.472
17	1.4738	.17257	184	17.257	8.9175	6	21.598	35.785
18	1.5708	.19035	182	19.035	10.014	4	7.0798	37.122
19	1.6680	.22106	180	22.106	10.210	2	8.2958	
20	1.7671	.24550	178	24.550	10.407	0	8.6179	
21	1.8653	.27088	176	27.088	10.603	8	22.384	
22	1.9635	.30680	174	30.680	10.709	6	23.189	
23	2.0617	.33824	172	33.824	10.906	4	42.718	
24	2.1598	.37122	170	37.122	11.192	2	23.502	
25	2.2580	.40574	168	40.574	11.388	0	44.179	
26	2.3562	.44179	166	44.179	11.781	10	24.347	
27	2.4544	.47937	164	47.937	11.977	8	47.173	
28	2.5535	.51849	162	51.849	12.174	6	24.740	
29	2.6507	.56014	160	56.014	12.370	4	48.707	
30	2.7480	.60132	158	60.132	12.566	2	11.046	
31	2.8471	.64504	156	64.504	12.763	0	12.763	
32	2.9452	.69029	154	69.029	12.959	8	27.096	
33	3.0434	.73708	152	73.708	13.155	6	27.489	
34	1.	3.1416	150	75.534	13.352	4	27.882	
35	3.3370	.85000	148	85.000	13.548	2	14.007	
36	3.5343	.91440	146	91.440	13.744	0	28.274	
37	3.7306	1.1075	144	11.075	13.941	8	65.817	
38	3.9270	1.2272	142	12.272	14.137	6	28.667	
39	4.1233	1.3530	140	13.530	14.334	4	65.397	
40	4.3197	1.4548	138	14.548	14.530	2	29.060	
41	4.5160	1.6230	136	16.230	14.726	0	67.201	
42	4.7124	1.7671	134	17.671	14.923	10	30.238	
43	4.9087	1.8175	132	18.175	15.119	8	72.728	
44	5.1051	2.0739	130	20.739	15.315	6	18.190	
45	5.3014	2.2365	128	22.365	15.512	4	31.023	
46	5.4978	2.4053	126	24.053	15.708	2	31.416	
47	5.6911	2.5502	124	25.502	15.900	0	31.809	
48	5.8905	2.7612	122	27.612	16.101	10	32.201	
49	6.0868	2.9483	120	29.483	16.297	8	32.987	
50	2.	6.2832	118	31.416	16.493	6	33.379	
51	6.4795	3.3410	116	33.410	16.690	4	34.165	

\*Approximate area, sufficiently accurate for practical purposes, including estimating.

## D-18

**(Continued). Circumferences and Areas of Circles  
(Advancing of eighths)**

Dia.	Circum.	Area*	Dia.	Circum.	Area*	Dia.	Circum.	Area*	Dia.	Circum.	Area*
11	34.658	95.033	1714	54.192	233.71	2314	79.527	433.74	24	75.398	452.39
12	34.950	97.205	1724	54.585	237.10	24220	74.220	438.36	25	75.791	457.11
13	35.343	99.402	1734	54.978	240.53	24.813	74.613	443.01	26	76.184	461.86
14	35.736	101.62	1744	55.371	243.98	25.006	75.006	447.89	27	76.576	466.64
15	36.128	103.87	1754	55.763	247.45	26.05	75.398	452.39	28	76.969	471.44
16	36.521	106.14	1764	56.156	250.95	27	77.302	478.26	29	77.754	481.11
17	36.914	108.43	1774	56.549	254.47	28	78.147	485.98	30	78.540	490.87
18	37.306	110.75	1784	56.941	258.02	29	79.325	500.74	31	79.933	506.70
19	37.699	113.10	1794	57.327	261.59	30	80.111	510.71	32	80.860	515.72
20	38.092	115.47	1804	57.727	265.18	33	80.866	515.72	34	81.269	525.84
21	38.485	117.86	1814	58.119	268.80	35	81.681	530.93	36	82.074	536.05
22	38.877	120.28	1824	58.512	272.45	37	82.467	541.19	38	83.860	556.76
23	39.270	122.72	1834	58.805	276.12	39	83.408	562.00	40	84.330	562.00
24	39.663	125.19	1844	59.298	279.81	41	84.228	572.58	42	85.216	577.87
25	40.055	127.68	1854	59.690	283.53	43	85.608	583.21	44	86.001	588.57
26	40.448	130.19	1864	60.083	287.27	45	86.394	593.96	46	87.178	604.81
27	40.841	132.73	1874	60.476	291.04	47	87.736	609.37	48	88.572	610.27
28	41.233	135.30	1884	60.868	294.83	49	88.330	614.81	50	89.143	621.26
29	41.626	137.89	1894	61.261	298.65	51	89.535	627.94	52	90.321	643.55
30	42.019	140.50	1904	61.654	302.49	53	90.928	649.18	54	91.719	654.64
31	42.412	143.14	1914	62.046	306.35	55	92.323	664.80	56	93.117	677.71
32	42.804	145.80	1924	62.439	310.24	57	93.905	681.55	58	94.706	696.52
33	43.187	148.46	1934	62.833	314.16	59	95.527	706.36	60	96.315	721.23
34	43.582	151.20	1944	63.226	318.03	61	97.244	731.04	62	97.932	745.91
35	43.982	153.94	1954	63.617	321.90	63	98.951	745.91	64	99.668	760.78
36	44.375	156.70	1964	64.010	324.77	65	100.385	760.78	66	101.113	775.65
37	44.768	159.48	1974	64.403	328.06	67	102.008	775.65	68	102.742	790.52
38	45.160	162.30	1984	64.795	330.06	69	103.645	790.52	70	104.330	804.39
39	45.553	165.18	1994	65.188	334.16	71	105.372	804.39	72	106.059	818.26
40	45.946	167.99	2004	65.581	342.25	73	107.785	818.26	74	108.516	832.13
41	46.338	170.87	2014	66.976	348.33	75	109.403	832.13	76	110.140	846.00
42	46.731	173.78	2024	67.368	354.40	77	111.117	846.00	78	111.857	860.87
43	47.124	176.71	2034	67.759	360.46	79	112.844	860.87	80	113.555	874.74
44	47.517	179.67	2044	68.152	366.53	81	114.394	874.74	82	115.091	888.61
45	47.909	182.65	2054	68.544	372.60	83	115.832	888.61	84	116.529	902.48
46	48.302	185.66	2064	68.937	378.67	85	117.277	902.48	86	117.925	916.35
47	48.895	188.69	2074	69.329	384.74	87	118.724	916.35	88	119.371	930.22
48	49.087	191.75	2084	69.722	387.53	89	120.169	930.22	90	120.816	944.09
49	49.480	194.83	2094	69.882	397.81	91	121.506	944.09	92	122.153	957.78
50	49.878	197.93	2104	70.864	406.49	93	122.853	957.78	94	123.491	971.55
51	50.265	201.06	2114	70.956	410.97	95	124.188	971.55	96	124.835	985.32
52	50.658	204.22	2124	70.293	415.48	97	125.535	985.32	98	126.180	999.09
53	51.051	207.39	2134	70.666	420.00	99	126.928	999.09	100	127.571	101.86
54	51.444	210.60	2144	71.079	424.58	101	128.321	101.86	102	128.911	103.63
55	51.836	213.82	2154	71.471	430.96	103	129.713	103.63	104	130.304	105.39
56	52.229	217.03	2164	71.864	437.44	105	131.096	105.39	106	131.789	107.17
57	52.622	220.25	2174	72.257	443.92	107	132.482	107.17	108	133.175	108.94
58	53.014	223.45	2184	72.649	450.40	109	133.868	108.94	110	134.561	110.71
59	53.407	226.68	2194	73.042	456.88	111	135.264				

**D-18**

(Continued). Circumferences and Areas of Circles  
(Advancing of eighths)

Dia.	Circum.	Area*	Dia.	Circum.	Area*	Dia.	Circum.	Area*
67 $\frac{1}{8}$	212.058	3578.5	73 $\frac{1}{4}$	231.692	4271.8	80.	251.327	5026.5
68 $\frac{1}{8}$	212.450	3591.7	74 $\frac{1}{4}$	232.088	4286.3	81.	251.720	5042.3
69 $\frac{1}{8}$	212.843	3605.0	74 $\frac{1}{4}$	232.478	4300.8	81.	252.113	5056.0
70 $\frac{1}{8}$	213.236	3618.3	74 $\frac{1}{4}$	232.871	4315.4	81.	252.506	5073.8
71 $\frac{1}{8}$	213.628	3631.7	74 $\frac{1}{4}$	233.263	4328.9	81.	252.898	5099.6
72 $\frac{1}{8}$	214.021	3645.0	74 $\frac{1}{4}$	233.650	4344.5	81.	253.291	5104.4
73 $\frac{1}{8}$	214.414	3658.4	74 $\frac{1}{4}$	234.049	4359.2	81.	253.684	5121.2
74 $\frac{1}{8}$	214.806	3671.8	74 $\frac{1}{4}$	234.441	4373.8	81.	254.076	5137.1
75 $\frac{1}{8}$	215.199	3685.3	74 $\frac{1}{4}$	234.834	4388.5	81.	254.469	5153.0
76 $\frac{1}{8}$	215.592	3698.7	74 $\frac{1}{4}$	235.227	4403.1	81.	254.862	5168.0
77 $\frac{1}{8}$	215.984	3712.2	75.	235.619	4417.9	81.	255.254	5184.9
78 $\frac{1}{8}$	216.377	3726.7	75.	236.012	4432.6	81.	256.647	5200.8
79 $\frac{1}{8}$	216.770	3739.3	74 $\frac{1}{4}$	236.405	4447.4	81.	256.183	5228.8
80 $\frac{1}{8}$	217.163	3752.8	74 $\frac{1}{4}$	236.798	4462.2	81.	256.525	5248.9
81 $\frac{1}{8}$	217.555	3766.4	74 $\frac{1}{4}$	237.190	4477.0	81.	257.181	5264.9
82 $\frac{1}{8}$	217.948	3780.0	74 $\frac{1}{4}$	237.583	4491.8	82.	257.611	5281.0
83 $\frac{1}{8}$	218.341	3793.7	74 $\frac{1}{4}$	237.976	4506.7	82.	258.003	5297.1
84 $\frac{1}{8}$	218.733	3807.3	74 $\frac{1}{4}$	238.368	4521.5	82.	258.396	5313.3
85 $\frac{1}{8}$	219.126	3821.0	74 $\frac{1}{4}$	238.761	4536.5	82.	258.789	5329.4
86 $\frac{1}{8}$	219.510	3834.7	76.	239.154	4551.4	82.	259.181	5345.6
87 $\frac{1}{8}$	219.891	3848.5	74 $\frac{1}{4}$	239.548	4566.4	82.	259.574	5361.8
88 $\frac{1}{8}$	220.304	3862.3	74 $\frac{1}{4}$	240.939	4581.3	82.	260.907	5378.1
89 $\frac{1}{8}$	220.807	3876.0	74 $\frac{1}{4}$	240.382	4596.3	82.	260.339	5394.3
90 $\frac{1}{8}$	221.090	3889.8	74 $\frac{1}{4}$	240.725	4611.4	83.	260.752	5410.6
91 $\frac{1}{8}$	221.482	3903.6	74 $\frac{1}{4}$	241.117	4626.4	83.	261.145	5426.9
92 $\frac{1}{8}$	221.875	3917.5	74 $\frac{1}{4}$	241.510	4641.5	83.	261.538	5443.3
93 $\frac{1}{8}$	222.268	3931.4	74 $\frac{1}{4}$	241.903	4656.6	83.	261.930	5459.6
94 $\frac{1}{8}$	222.660	3945.3	77.	242.296	4671.8	83.	262.323	5476.0
95 $\frac{1}{8}$	223.058	3959.2	74 $\frac{1}{4}$	242.688	4686.9	83.	262.716	5492.4
96 $\frac{1}{8}$	223.446	3973.1	74 $\frac{1}{4}$	243.081	4702.1	83.	263.108	5508.8
97 $\frac{1}{8}$	223.838	3987.1	74 $\frac{1}{4}$	243.473	4717.3	83.	263.501	5525.3
98 $\frac{1}{8}$	224.231	4001.1	74 $\frac{1}{4}$	243.866	4732.5	84.	263.894	5541.8
99 $\frac{1}{8}$	224.624	4015.2	74 $\frac{1}{4}$	244.259	4747.8	84.	264.286	5558.3
100 $\frac{1}{8}$	225.017	4029.2	74 $\frac{1}{4}$	244.652	4763.1	84.	264.679	5574.8
101 $\frac{1}{8}$	225.409	4043.3	74 $\frac{1}{4}$	245.044	4778.4	84.	265.072	5591.4
102 $\frac{1}{8}$	225.802	4057.4	78.	245.437	4793.7	84.	265.465	5607.9
103 $\frac{1}{8}$	226.195	4071.5	74 $\frac{1}{4}$	245.830	4809.0	84.	265.857	5624.5
104 $\frac{1}{8}$	226.587	4085.7	74 $\frac{1}{4}$	246.222	4824.4	84.	266.250	5641.2
105 $\frac{1}{8}$	226.980	4099.8	74 $\frac{1}{4}$	246.615	4839.8	84.	266.643	5657.8
106 $\frac{1}{8}$	227.373	4114.0	74 $\frac{1}{4}$	247.008	4855.2	84.	267.428	5670.9
107 $\frac{1}{8}$	227.765	4128.2	74 $\frac{1}{4}$	247.400	4870.7	84.	268.006	5684.5
108 $\frac{1}{8}$	228.158	4142.5	74 $\frac{1}{4}$	247.793	4886.2	84.	268.909	5698.3
109 $\frac{1}{8}$	228.551	4156.8	74 $\frac{1}{4}$	248.071	4902.7	84.	269.318	5714.5
110 $\frac{1}{8}$	228.944	4171.1	79.	248.186	4917.2	84.	269.906	5724.7
111 $\frac{1}{8}$	229.336	4185.4	74 $\frac{1}{4}$	248.579	4932.7	84.	270.570	5738.3
112 $\frac{1}{8}$	229.729	4199.7	74 $\frac{1}{4}$	249.364	4948.3	84.	270.876	5747.1
113 $\frac{1}{8}$	230.122	4214.1	74 $\frac{1}{4}$	249.757	4963.9	84.	271.284	5759.6
114 $\frac{1}{8}$	230.514	4228.5	74 $\frac{1}{4}$	250.140	4979.5	84.	271.691	5770.8
115 $\frac{1}{8}$	230.907	4242.9	74 $\frac{1}{4}$	250.542	4995.2	84.	272.098	5780.8
116 $\frac{1}{8}$	231.300	4257.4	74 $\frac{1}{4}$	250.935	5010.9	84.	272.505	5795.7

\*Approximate area, sufficiently accurate for practical purposes, including estimating.

**D-18**

(Continued). Circumferences and Areas of Circles  
(Advancing of eighths)

Dia.	Circum.	Area*	Dia.	Circum.	Area*	Dia.	Circum.	Area*
86 $\frac{1}{4}$	270.902	5842.8	92 $\frac{1}{4}$	290.597	6720.1	98 $\frac{1}{4}$	310.232	7658.9
87 $\frac{1}{4}$	271.355	5850.8	93 $\frac{1}{4}$	290.990	6738.2	99 $\frac{1}{4}$	310.625	7678.3
88 $\frac{1}{4}$	271.748	5876.6	94 $\frac{1}{4}$	291.383	6756.4	99 $\frac{1}{4}$	311.018	7697.7
89 $\frac{1}{4}$	272.140	5893.5	95 $\frac{1}{4}$	291.775	6774.7	99 $\frac{1}{4}$	311.410	7717.1
90 $\frac{1}{4}$	272.533	5910.6	96 $\frac{1}{4}$	292.168	6792.9	99 $\frac{1}{4}$	311.803	7736.6
91 $\frac{1}{4}$	272.926	5927.6	97 $\frac{1}{4}$	292.561	6811.2	99 $\frac{1}{4}$	312.196	7756.1
92 $\frac{1}{4}$	273.319	5944.7	98 $\frac{1}{4}$	292.954	6829.5	99 $\frac{1}{4}$	312.588	7775.6
93 $\frac{1}{4}$	273.711	5961.8	99 $\frac{1}{4}$	293.346	6847.8	99 $\frac{1}{4}$	312.981	7795.2
94 $\frac{1}{4}$	274.407	5988.0	100 $\frac{1}{4}$	294.132	6864.5	100 $\frac{1}{4}$	313.374	7814.8
95 $\frac{1}{4}$	274.798	6004.0	101 $\frac{1}{4}$	294.917	6921.3	100 $\frac{1}{4}$	313.767	7834.4
96 $\frac{1}{4}$	275.675	6047.6	102 $\frac{1}{4}$	295.310	6938.8	100 $\frac{1}{4}$	314.16	7854
97 $\frac{1}{4}$	276.067	6061.9	103 $\frac{1}{4}$	295.702	6958.2	100 $\frac{1}{4}$	314.55	7873
98 $\frac{1}{4}$	276.460	6082.1	104 $\frac{1}{4}$	296.095	6976.7	100 $\frac{1}{4}$	314.95	7893
99 $\frac{1}{4}$	276.853	6099.4	105 $\frac{1}{4}$	296.488	6995.3	100 $\frac{1}{4}$	315.34	7913
100 $\frac{1}{4}$	277.246	6116.7	106 $\frac{1}{4}$	296.881	7013.8	100 $\frac{1}{4}$	315.73	7933
101 $\frac{1}{4}$	277.638	6134.1	107 $\frac{1}{4}$	297.273	7032.4	100 $\frac{1}{4}$	316.12	7952
102 $\frac{1}{4}$	278.031	6151.4	108 $\frac{1}{4}$	297.666	7051.0	100 $\frac{1}{4}$	316.52	7972
103 $\frac{1}{4}$	278.424	6168.8	109 $\frac{1}{4}$	298.059	7069.6	100 $\frac{1}{4}$	316.91	8002
104 $\frac{1}{4}$	278.816	6186.2	110 $\frac{1}{4}$	298.451	7088.2	100 $\frac{1}{4}$	317.30	8012
105 $\frac{1}{4}$	279.209	6203.7	111 $\frac{1}{4}$	298.844	7106.9	100 $\frac{1}{4}$	317.69	8032
106 $\frac{1}{4}$	279.602	6221.1	112 $\frac{1}{4}$	299.237	7125.6	100 $\frac{1}{4}$	318.09	8052
107 $\frac{1}{4}$	279.994	6238.6	113 $\frac{1}{4}$	299.629	7144.3	100 $\frac{1}{4}$	318.48	8071
108 $\frac{1}{4}$	280.387	6256.1	114 $\frac{1}{4}$	300.022	7163.0	100 $\frac{1}{4}$	318.87	8091
109 $\frac{1}{4}$	280.780	6273.7	115 $\frac{1}{4}$	300.415	7181.8	100 $\frac{1}{4}$	319.27	8111
110 $\frac{1}{4}$	281.173	6291.2	116 $\frac{1}{4}$	300.807	7200.6	100 $\frac{1}{4}$	319.66	8131
111 $\frac{1}{4}$	281.565	6308.8	117 $\frac{1}{4}$	301.200	7219.4	100 $\frac{1}{4}$	320.05	8151
112 $\frac{1}{4}$	282.361	6344.1	118 $\frac{1}{4}$	301.593	7238.2	100 $\frac{1}{4}$	320.44	8171
113 $\frac{1}{4}$	282.743	6361.7	119 $\frac{1}{4}$	301.986	7257.1	100 $\frac{1}{4}$	320.84	8191
114 $\frac{1}{4}$	283.136	6379.4	120 $\frac{1}{4}$	302.376	7276.0	100 $\frac{1}{4}$	321.23	8211
115 $\frac{1}{4}$	283.529	6397.1	121 $\frac{1}{4}$	302.771	7294.9	100 $\frac{1}{4}$	321.62	8231
116 $\frac{1}{4}$	283.921	6414.9	122 $\frac{1}{4}$	303.164	7313.8	100 $\frac{1}{4}$	322.01	8252
117 $\frac{1}{4}$	284.314	6422.6	123 $\frac{1}{4}$	303.556	7332.8	100 $\frac{1}{4}$	322.41	8272
118 $\frac{1}{4}$	284.707	6430.4	124 $\frac{1}{4}$	303.949	7351.8	100 $\frac{1}{4}$	322.80	8292
119 $\frac{1}{4}$	285.100	6448.2	125 $\frac{1}{4}$	304.342	7370.8	100 $\frac{1}{4}$	323.19	8312
120 $\frac{1}{4}$	285.492	6468.0	126 $\frac{1}{4}$	304.734	7389.8	100 $\frac{1}{4}$	323.58	8332
121 $\frac{1}{4}$	285.885	6480.9	127 $\frac{1}{4}$	305.127	7408.9	100 $\frac{1}{4}$	323.98	8352
122 $\frac{1}{4}$	286.278	6501.8	128 $\frac{1}{4}$	305.520	7428.0	100 $\frac{1}{4}$	324.37	8372
123 $\frac{1}{4}$	286.670	6521.8	129 $\frac{1}{4}$	305.913	7447.1	100 $\frac{1}{4}$	324.76	8393
124 $\frac{1}{4}$	287.063	6539.7	130 $\frac{$					

### D-18

**(Continued). Circumferences and Areas of Circles  
(Advancing of eighths)**

Dia.	Circum.	Area*	Dia.	Circum.	Area*	Dia.	Circum.	Area*
105.	329.87	8659	111 $\frac{1}{8}$	349.50	9720	117 $\frac{1}{4}$	369.14	10844
$\frac{1}{8}$	330.28	8679	$\frac{1}{8}$	349.90	9742	$\frac{1}{8}$	369.53	10887
$\frac{1}{4}$	330.65	8700	$\frac{1}{4}$	360.29	9764	$\frac{1}{4}$	369.92	10890
$\frac{1}{2}$	331.05	8721	$\frac{1}{2}$	360.68	9786	$\frac{1}{2}$	370.32	10913
$\frac{3}{4}$	331.44	8741	$\frac{3}{4}$	361.07	9808	$\frac{3}{4}$	370.71	10936
$\frac{5}{8}$	331.83	8762	$\frac{5}{8}$	361.47	9830	$\frac{5}{8}$	371.11	10960
$\frac{7}{8}$	332.22	8783	$\frac{7}{8}$	361.86	9852	$\frac{7}{8}$	371.49	10983
106.	333.01	8825	$\frac{1}{8}$	362.25	9874	$\frac{1}{8}$	371.89	11007
$\frac{1}{8}$	333.40	8845	$\frac{1}{8}$	362.65	9897	$\frac{1}{8}$	372.28	11030
$\frac{1}{4}$	333.80	8866	$\frac{1}{4}$	363.04	9919	$\frac{1}{4}$	372.67	11053
$\frac{1}{2}$	334.19	8887	$\frac{1}{2}$	363.43	9941	$\frac{1}{2}$	373.07	11076
$\frac{3}{4}$	334.58	8908	$\frac{3}{4}$	363.82	9963	$\frac{3}{4}$	373.46	11099
$\frac{5}{8}$	334.97	8929	$\frac{5}{8}$	364.22	9985	$\frac{5}{8}$	373.85	11122
$\frac{7}{8}$	335.37	8950	$\frac{7}{8}$	364.61	10007	$\frac{7}{8}$	374.24	11146
107.	336.15	8992	$\frac{1}{8}$	365.00	10029	$\frac{1}{8}$	374.64	11169
$\frac{1}{8}$	336.54	9014	$\frac{1}{8}$	365.39	10052	$\frac{1}{8}$	375.03	11193
$\frac{1}{4}$	336.94	9035	$\frac{1}{4}$	365.18	10097	$\frac{1}{4}$	375.42	11216
$\frac{1}{2}$	337.33	9056	$\frac{1}{2}$	365.57	10119	$\frac{1}{2}$	375.81	11240
$\frac{3}{4}$	337.72	9077	$\frac{3}{4}$	365.96	10141	$\frac{3}{4}$	376.21	11263
$\frac{5}{8}$	338.12	9098	$\frac{5}{8}$	366.36	10163	$\frac{5}{8}$	376.60	11287
$\frac{7}{8}$	338.51	9119	$\frac{7}{8}$	367.75	10185	$\frac{7}{8}$	376.99	11310
911	339.90	9140	$\frac{1}{8}$	368.14	10207	$\frac{1}{8}$	377.39	11334
$\frac{1}{8}$	340.29	9161	$\frac{1}{8}$	368.54	10230	$\frac{1}{8}$	377.78	11357
$\frac{1}{4}$	340.69	9183	$\frac{1}{4}$	368.93	10253	$\frac{1}{4}$	378.17	11381
$\frac{1}{2}$	340.08	9204	$\frac{1}{2}$	369.32	10275	$\frac{1}{2}$	378.56	11404
$\frac{3}{4}$	340.47	9225	$\frac{3}{4}$	369.71	10297	$\frac{3}{4}$	378.96	11428
$\frac{5}{8}$	340.86	9246	$\frac{5}{8}$	370.11	10320	$\frac{5}{8}$	379.35	11451
$\frac{7}{8}$	341.26	9268	$\frac{7}{8}$	370.50	10342	$\frac{7}{8}$	379.74	11475
$\frac{1}{8}$	341.65	9289	$\frac{1}{8}$	370.89	10363	$\frac{1}{8}$	380.13	11499
$\frac{1}{8}$	342.04	9310	$\frac{1}{8}$	361.28	10387	$\frac{1}{8}$	380.53	11522
$\frac{1}{4}$	342.43	9331	$\frac{1}{4}$	361.68	10410	$\frac{1}{4}$	380.92	11546
$\frac{1}{2}$	342.83	9353	$\frac{1}{2}$	362.07	10432	$\frac{1}{2}$	381.31	11570
$\frac{3}{4}$	343.22	9374	$\frac{3}{4}$	362.46	10455	$\frac{3}{4}$	381.70	11594
$\frac{5}{8}$	343.61	9396	$\frac{5}{8}$	362.86	10477	$\frac{5}{8}$	382.10	11618
$\frac{7}{8}$	344.01	9417	$\frac{7}{8}$	363.25	10500	$\frac{7}{8}$	382.49	11642
$\frac{1}{8}$	344.40	9439	$\frac{1}{8}$	363.64	10522	$\frac{1}{8}$	382.88	11666
$\frac{1}{8}$	344.79	9460	$\frac{1}{8}$	364.03	10545	$\frac{1}{8}$	383.28	11690
$\frac{1}{4}$	345.18	9491	$\frac{1}{4}$	364.43	10568	$\frac{1}{4}$	383.67	11714
$\frac{1}{2}$	345.58	9503	$\frac{1}{2}$	364.82	10590	$\frac{1}{2}$	384.06	11738
$\frac{3}{4}$	345.97	9525	$\frac{3}{4}$	365.21	10613	$\frac{3}{4}$	384.45	11762
$\frac{5}{8}$	346.36	9546	$\frac{5}{8}$	366.00	10636	$\frac{5}{8}$	384.85	11786
$\frac{7}{8}$	346.75	9568	$\frac{7}{8}$	366.39	10682	$\frac{7}{8}$	385.63	11834
$\frac{1}{8}$	347.15	9589	$\frac{1}{8}$	366.78	10705	$\frac{1}{8}$	386.02	11868
$\frac{1}{8}$	347.54	9611	$\frac{1}{8}$	367.18	10728	$\frac{1}{8}$	386.42	11882
$\frac{1}{4}$	347.93	9633	$\frac{1}{4}$	367.57	10751	$\frac{1}{4}$	386.81	11907
$\frac{1}{2}$	348.33	9655	$\frac{1}{2}$	367.96	10774	$\frac{1}{2}$	387.20	11931
111.	348.72	9677	$\frac{1}{2}$	368.35	10798	$\frac{1}{2}$	387.60	11956
$\frac{3}{4}$	349.11	9698	$\frac{3}{4}$	368.75	10821	$\frac{3}{4}$	387.99	11980

### D-18

**(Continued). Circumferences and Areas of Circles  
(Advancing of eighths)**

Dia.	Circum.	Area*	Dia.	Circum.	Area*	Dia.	Circum.	Area*
123 $\frac{1}{8}$	389.38	12004	130.	408.41	13273	136 $\frac{1}{4}$	428.04	14580
$\frac{1}{8}$	388.77	12028	$\frac{1}{8}$	408.80	13299	$\frac{1}{8}$	428.44	14607
$\frac{1}{4}$	389.17	12053	$\frac{1}{4}$	409.19	13324	$\frac{1}{4}$	428.83	14638
$\frac{3}{4}$	390.56	12076	$\frac{3}{4}$	409.58	13350	$\frac{3}{4}$	429.22	14660
124.	390.56	12076	131.	410.87	13375	137.	430.40	14741
$\frac{1}{8}$	390.95	12101	$\frac{1}{8}$	411.56	13478	$\frac{1}{8}$	430.79	14768
$\frac{1}{4}$	391.32	12223	$\frac{1}{4}$	411.94	13504	$\frac{1}{4}$	431.19	14795
$\frac{3}{4}$	391.69	12248	$\frac{3}{4}$	412.34	13529	$\frac{3}{4}$	431.58	14822
125.	392.70	12272	$\frac{1}{8}$	412.73	13555	$\frac{1}{8}$	432.30	14849
$\frac{1}{8}$	393.09	12297	$\frac{1}{8}$	413.12	13581	$\frac{1}{8}$	432.76	14908
$\frac{1}{4}$	393.49	12321	$\frac{1}{4}$	413.51	13607	$\frac{1}{4}$	433.15	14930
$\frac{3}{4}$	393.88	12346	$\frac{3}{4}$	414.30	13639	138.	433.54	14957
126.	394.27	12370	$\frac{1}{8}$	414.69	13685	$\frac{1}{8}$	433.93	14984
$\frac{1}{8}$	394.66	12395	133.	415.08	13711	$\frac{1}{8}$	434.33	15012
$\frac{1}{4}$	395.06	12419	$\frac{1}{4}$	415.48	13737	$\frac{1}{4}$	434.72	15030
$\frac{3}{4}$	395.45	12444	$\frac{3}{4}$	415.87	13763	$\frac{3}{4}$	435.50	15067
127.	395.84	12469	$\frac{1}{8}$	416.26	13815	$\frac{1}{8}$	436.29	15148
$\frac{1}{8}$	396.23	12494	$\frac{1}{8}$	416.66	13841	$\frac{1}{8}$	436.68	15178
$\frac{1}{4}$	396.63	12518	$\frac{1}{4}$	417.05	13867	139.	437.08	15203
$\frac{3}{4}$	397.02	12543	$\frac{3}{4}$	417.44	13893	$\frac{3}{4}$	437.47	15230
128.	397.41	12568	$\frac{1}{8}$	417.83	13919	$\frac{1}{8}$	437.86	15268
$\frac{1}{8}$	397.81	12593	133.	418.23	13946	$\frac{1}{8}$	438.25	15285
$\frac{1}{4}$	398.20	12618	$\frac{1}{4}$	418.62	13972	$\frac{1}{4}$	438.65	15313
$\frac{3}{4}$	398.59	12643	$\frac{3}{4}$	419.01	13999	$\frac{3}{4}$	439.04	15340
129.	398.98	12668	$\frac{1}{8}$	419.39	14021	$\frac{1}{8}$	439.43	15367
$\frac{1}{8}$	399.38	12693	$\frac{1}{8}$	419.70	14051	$\frac{1}{8}$	439.82	15394
$\frac{1}{4}$	399.77	12718	$\frac{1}{4}$	420.09	14077	$\frac{1}{4}$	440.22	15422
$\frac{3}{4}$	400.16	12743	$\frac{3}{4}$	420.48	14103	$\frac{3}{4}$	440.61	15449
130.	400.55	12768	$\frac{1}{8}$	420.87	14130	$\frac{1}{8}$	441.00	15477
$\frac{1}{8}$	401.34	12818	$\frac{1}{8}$	421.26	14156	$\frac{1}{8}$	441.40	15504
$\frac{1}{4}$	401.73	12843	$\frac{1}{4}$	421.66	14183	$\frac{1}{4}$	441.79	15532
$\frac{3}{4}$	402.13	12868	$\frac{3}{4}$	422.05	14209	$\frac{3}{4}$	442.18	15559
131.	402.52	12893	$\frac{1}{8}$	422.44	14236	$\frac{1}{8}$	442.57	15587
$\frac{1}{8}$	402.91	12919	$\frac{1}{8}$	422.84	14262	$\frac{1}{8}$	442.97	15615
$\frac{1}{4}$	403.30	12944	$\frac{1}{4}$	423.23	14288	141.	443.36	15642
$\frac{3}{4}$	403.67	12970	$\frac{3}{4}$	423.62	14314	$\frac{3}{4}$	443.75	15670
132.	404.09	12995	$\frac{1}{8}$	424.51	14341	$\frac{1}{8}$	444.14	15697
$\frac{1}{8}$	404.48	13020	135.	424.90	14367	$\frac{1}{8}$	444.54	15725
$\frac{1}{4}$	404.87	13045	$\frac{1}{4}$	425.29	14394	$\frac{1}{4}$	444.93	15753
$\frac{3}{4}$	405.27	13070	$\frac{3}{4}$	425.68	14420	$\frac{3}{4}$	445.32	15781
133.	405.66	13096	$\frac{1}{8}$	426.08	14447	$\frac{1}{8}$	445.72	15809
$\frac{1}{8}$	406.05	13121	$\frac{1}{8}$	426.44	14473	$\frac{1}{8}$	446.11	15837
$\frac{1}{4}$	406.44	13147	$\frac{1}{4}$	426.84	14500	142.	446.50	15865
$\frac{3}{4}$	406.84	13172	$\frac{3}{4}$	427.23	14527	$\frac{3}{4}$	446.80	15893
134.	407.23	13198	$\frac{1}{8}$	427.60	14553	$\frac{1}{8}$	447.20	15921
$\frac{1}{8}$	407.62	13223	136.	428.00	14583	$\frac{1}{8}$	447.59	15951
$\frac{1}{4}$	408.02	13248	$\frac{1}{4}$	428.38	14600	142.	448.11	15987
$\frac{3}{4}$	408.44	13272	$\frac{3}{4}$	428.77	14627	$\frac{3}{4}$	448.50	16025
135.	408.84	13300	$\frac{1}{8}$	429.16	14653	$\frac{1}{8}$	448.86	16063
$\frac{1}{8}$	409.24	13326	$\frac{1}{8}$	429.55	14680	$\frac{1}{8}$	449.20	16101
$\frac{1}{4}$ </td								

**D-18**

(Continued). Circumferences and Areas of Circles  
(Advancing of eighths)

Dia.	Circum.	Area*	Dia.	Circum.	Area*	Dia.	Circum.	Area*
142 56	447.68	16049	148 56	467.81	17379	156.	486.95	18889
56	448.07	15977	56	467.71	17408	56	487.34	16900
56	448.46	16005	56	468.10	17437	56	487.73	18930
56	448.86	16033	149.	468.49	17466	56	488.13	18961
56	449.25	16061	56	468.88	17496	56	488.52	18991
56	449.64	16089	56	469.28	17525	56	488.91	19022
56	450.03	16117	56	469.67	17555	56	489.30	19053
56	450.43	16145	56	470.06	17584	56	489.69	19083
56	450.82	16173	56	470.46	17614	156.	490.09	19113
56	451.21	16201	56	470.86	17643	56	490.48	19144
56	451.61	16229	56	471.24	17672	56	490.88	19174
56	452.00	16258	150.	471.63	17702	56	491.27	19205
56	452.39	16286	56	472.03	17731	56	491.66	19235
56	452.78	16314	56	472.42	17761	56	492.05	19265
56	453.18	16342	56	472.81	17790	56	492.44	19297
56	453.57	16371	56	473.20	17820	56	492.84	19328
56	453.96	16399	56	473.59	17849	157.	493.23	19359
56	454.35	16428	56	473.99	17879	56	493.62	19390
56	454.75	16456	56	474.38	17908	56	494.02	19421
56	455.14	16485	151.	474.77	17938	56	494.41	19452
56	455.53	16513	56	475.17	17967	56	494.80	19483
56	455.93	16542	56	475.56	17997	56	495.19	19514
56	456.32	16570	56	475.95	18026	56	495.59	19545
56	456.71	16599	56	476.35	18056	56	495.98	19576
56	457.10	16627	56	476.74	18086	158.	496.37	19607
56	457.50	16656	56	477.13	18116	56	496.77	19638
56	457.89	16684	56	477.52	18146	56	497.16	19669
56	458.28	16713	152.	477.92	18175	56	497.55	19701
56	458.67	16742	56	478.31	18205	56	497.94	19732
56	459.07	16770	56	478.70	18235	56	498.34	19763
56	459.46	16799	56	479.09	18265	56	498.73	19794
56	459.85	16827	56	479.49	18295	159.	499.12	19825
56	460.24	16856	56	479.88	18326	56	499.51	19856
56	460.64	16885	56	480.27	18355	56	499.91	19887
56	461.03	16914	56	480.67	18385	56	500.30	19919
56	461.42	16943	153.	481.06	18415	56	500.69	19950
56	461.82	16972	56	481.46	18446	56	501.09	19982
56	462.21	17000	56	481.84	18476	56	501.48	20013
56	462.60	17029	56	482.24	18507	56	501.87	20044
56	462.99	17058	56	482.63	18537	56	502.26	20075
56	463.39	17087	56	483.02	18567	150.	502.66	20106
56	463.78	17116	56	483.41	18597	56	503.05	20138
56	464.17	17145	56	483.81	18627	56	503.44	20169
56	464.56	17174	154.	484.20	18658	56	503.83	20201
56	464.96	17203	56	484.59	18688	56	504.22	20232
56	465.35	17232	56	484.99	18718	56	504.62	20264
56	465.74	17262	56	485.38	18749	56	505.01	20295
56	466.14	17291	56	485.77	18778	56	505.41	20327
56	466.53	17321	56	486.16	18809	161.	506.80	20358
56	466.92	17350	56	486.56	18839	56	506.19	20390

\*Approximate area, sufficiently accurate for practical purposes, including estimating.

**D-18**

(Continued). Circumferences and Areas of Circles  
(Advancing of eighths)

Dia.	Circum.	Area*	Dia.	Circum.	Area*	Dia.	Circum.	Area*
161 56	506.58	20421	167 56	526.22	22035	173 56	545.85	23711
56	506.98	20453	56	526.61	22068	56	546.25	23746
56	507.37	20484	56	527.00	22101	56	547.42	23848
56	507.76	20516	56	527.40	22134	174.	547.82	23882
56	508.15	20548	56	527.79	22167	56	548.21	23917
56	508.55	20580	168.	528.18	22200	56	548.60	23951
56	508.94	20612	56	528.57	22238	56	549.00	23985
56	509.33	20644	56	528.97	22266	56	549.39	24019
56	509.73	20676	56	529.36	22299	56	549.78	24053
56	510.12	20707	56	529.75	22332	56	550.17	24087
56	510.51	20739	56	530.15	22366	56	550.57	24122
56	510.90	20771	56	530.54	22399	176.	551.32	24156
56	511.30	20803	56	530.93	22432	56	551.74	24191
56	511.69	20835	169.	531.32	22465	56	552.14	24225
56	512.08	20867	56	531.72	22499	56	552.53	24260
56	512.47	20899	56	532.11	22532	56	552.93	24294
56	512.87	20931	56	532.50	22566	56	553.31	24333
56	513.26	20964	56	532.89	22599	56	553.71	24368
56	513.65	20996	56	533.28	22632	56	554.09	24407
56	514.04	21028	56	533.68	22665	56	554.49	24457
56	514.44	21060	56	534.07	22698	56	554.88	24496
56	514.83	21092	170.	534.47	22731	56	555.24	24536
56	515.22	21124	56	534.86	22765	56	555.62	24575
56	515.62	21157	56	535.25	22798	56	556.00	24610
56	516.01	21189	56	535.64	22832	56	556.38	24640
56	516.40	21222	56	536.04	22865	56	556.76	24675
56	516.77	21254	56	536.43	22899	56	557.14	24710
56	517..	21287	56	536.82	22932	177.	557.51	24748
56	517.97	21351	171.	537.31	22966	56	557.88	24780
56	518.36	21383	56	538.00	23033	56	558.26	24815
56	518.76	21418	56	538.39	23066	56	558.64	24850
56	519.15	21448	56	538.78	23100	56	559.02	24885
56	519.54	21481	56	539.18	23133	56	559.39	24920
56	519.94	21513	56	539.57	23167	56	559.76	24955
56	520.33	21546	56	539.98	23201	178.	560.33	24990
56	520.72	21578	56	540.36	23235	56	560.68	25026
56	521.11	21610	172.	540.75	23268	56	561.17	25060
56	521.51	21642	56	541.14	23303	56	561.56	25095
56	521.90	21675	56	541.53	23336	56	561.96	25130
56	522.29	21707	56	541.93	23370	56	562.35	25165
56	522.68	21740	56	542.32	23404	56	562.74	25200
56	523.08	21772	56	542.71	23438	56	563.13	25236
56	523.47	21805	56	543.10	23472	179.	563.58	25271
56	523.88	21838	56	543.89	23506	56	564.04	25307
56	524.26	21871	178.	544.28	23540	56	564.51	25343
56	524.65	21904	56	544.68	23575	56	564.87	25377
56	525.04	21937	56	545.43	23609	56	565.10	25412
56	525.43	21969	56	546.07	23643	56	565.48	254412
56	525.83	22002	56	546.46	23677	56	565.85	25471

\*Approximate area, sufficiently accurate for practical purposes, including estimating.

## D-18

**(Continued). Circumferences and Areas of Circles  
(Advancing of eighths)**

Dia.	Circum.	Area*	Dia.	Circum.	Area*	Dia.	Circum.	Area*
180.	565.49	25447	186 1/8	585.12	27245	192 1/2	604.76	29108
181.	565.88	25482	187 1/8	585.52	27281	193 1/2	605.15	29141
182.	566.27	25518	188 1/8	585.91	27318	194 1/2	605.54	29179
183.	566.67	25553	189 1/8	586.30	27354	195 1/2	605.94	29217
184.	567.06	25589	190 1/8	586.69	27391	196 1/2	606.33	29255
185.	567.45	25624	191 1/8	587.09	27428	197 1/2	606.72	29283
186.	567.84	25660	192 1/8	587.48	27465	198 1/2	607.11	29311
187.	568.24	25696	193 1/8	587.87	27501	199 1/2	607.51	29339
188.	568.63	25730	194 1/8	588.27	27538	200 1/2	607.90	29407
189.	569.02	25765	195 1/8	588.66	27574	201 1/2	608.29	29445
190.	569.42	25801	196 1/8	589.05	27611	202 1/2	608.68	29483
191.	569.81	25836	197 1/8	589.44	27648	203 1/2	609.08	29521
192.	570.20	25872	198 1/8	589.84	27685	204 1/2	609.47	29559
193.	570.59	25908	199 1/8	590.23	27722	205 1/2	609.86	29597
194.	570.99	25944	200 1/8	590.62	27759	206 1/2	610.26	29636
195.	571.38	25980	201 1/8	591.01	27796	207 1/2	610.65	29674
196.	571.77	26016	202 1/8	591.41	27833	208 1/2	611.05	29713
197.	572.16	26051	203 1/8	591.80	27870	209 1/2	611.43	29751
198.	572.55	26087	204 1/8	592.19	27907	210 1/2	611.83	29789
199.	572.95	26122	205 1/8	592.58	27944	211 1/2	612.29	29827
200.	573.34	26158	206 1/8	592.98	27981	212 1/2	612.68	29865
201.	573.74	26194	207 1/8	593.37	28018	213 1/2	613.06	29903
202.	574.13	26230	208 1/8	593.76	28055	214 1/2	613.00	29908
203.	574.52	26266	209 1/8	594.16	28092	215 1/2	613.79	29980
204.	574.91	26302	210 1/8	594.55	28130	216 1/2	614.18	30019
205.	575.31	26338	211 1/8	594.94	28167	217 1/2	614.57	30057
206.	575.70	26374	212 1/8	595.33	28205	218 1/2	614.97	30096
207.	576.09	26410	213 1/8	595.73	28242	219 1/2	615.36	30134
208.	576.48	26446	214 1/8	596.12	28279	220 1/2	615.75	30172
209.	576.88	26482	215 1/8	596.51	28316	221 1/2	616.15	30210
210.	577.27	26518	216 1/8	596.90	28353	222 1/2	616.54	30249
211.	577.66	26554	217 1/8	597.29	28390	223 1/2	616.93	30287
212.	578.05	26590	218 1/8	597.68	28428	224 1/2	617.32	30326
213.	578.45	26626	219 1/8	598.08	28465	225 1/2	617.72	30364
214.	578.84	26663	220 1/8	598.47	28503	226 1/2	618.11	30403
215.	579.23	26699	221 1/8	598.86	28540	227 1/2	618.50	30442
216.	579.63	26736	222 1/8	599.25	28578	228 1/2	618.89	30481
217.	580.02	26772	223 1/8	599.64	28615	229 1/2	619.29	30519
218.	580.41	26808	224 1/8	600.04	28652	230 1/2	619.68	30558
219.	580.80	26844	225 1/8	600.44	28689	231 1/2	620.08	30596
220.	581.20	26880	226 1/8	600.83	28727	232 1/2	620.47	30635
221.	581.59	26916	227 1/8	601.22	28764	233 1/2	620.86	30674
222.	581.98	26953	228 1/8	601.62	28802	234 1/2	621.25	30713
223.	582.37	26989	229 1/8	602.01	28839	235 1/2	621.64	30752
224.	582.77	27026	230 1/8	602.40	28877	236 1/2	622.04	30791
225.	583.16	27062	231 1/8	602.79	28915	237 1/2	622.44	30830
226.	583.55	27099	232 1/8	603.19	28953	238 1/2	622.83	30869
227.	583.94	27136	233 1/8	603.58	28990	239 1/2	623.22	30908
228.	584.34	27172	234 1/8	603.97	29028	240 1/2	623.62	30947
229.	584.73	27208	235 1/8	604.36	29065	241 1/2	624.01	30986

\*Approximate area, sufficiently accurate for practical purposes, including estimating.

## D-18

**(Continued). Circumferences and Areas of Circles  
(Advancing of eighths)**

Dia.	Circum.	Area*									
198 1/8	624.40	31025	204 1/8	643.63	32966	211 1/8	662.88	34967	218 1/8	683.28	35008
199 1/8	624.79	31064	205 1/8	644.03	33006	212 1/8	683.67	35050	219 1/8	684.07	35091
200 1/8	625.18	31103	210 1/8	644.43	33046	217 1/8	684.46	35133	226 1/8	684.85	35174
201 1/8	625.58	31142	211 1/8	644.82	33087	223 1/8	685.24	35216	232 1/8	685.63	35257
202 1/8	625.97	31181	212 1/8	645.21	33127	228 1/8	685.61	35291	237 1/8	686.02	35299
203 1/8	626.36	31220	213 1/8	645.61	33168	234 1/8	686.43	35340	242 1/8	686.82	35382
204 1/8	626.76	31260	214 1/8	646.00	33208	239 1/8	687.21	35423	247 1/8	687.61	35465
205 1/8	627.15	31299	215 1/8	646.39	33249	244 1/8	688.00	35507	252 1/8	688.39	35549
206 1/8	627.54	31338	216 1/8	646.78	33289	249 1/8	688.78	35591	257 1/8	689.16	35633
207 1/8	627.94	31377	217 1/8	647.17	33329	254 1/8	689.57	35674	262 1/8	690.96	35718
208 1/8	628.32	31416	218 1/8	647.57	33369	259 1/8	690.86	35758	267 1/8	691.50	35800
209 1/8	628.72	31455	219 1/8	647.96	33410	264 1/8	692.35	35843	272 1/8	693.50	35926
210 1/8	629.11	31493	220 1/8	648.35	33450	269 1/8	693.89	35984	277 1/8	694.87	36010
211 1/8	629.51	31534	221 1/8	648.75	33491	274 1/8	695.25	36062	282 1/8	695.50	36187
212 1/8	629.90	31574	222 1/8	649.14	33531	279 1/8	696.67	36179	287 1/8	697.05	36221
213 1/8	630.29	31613	223 1/8	649.53	33572	284 1/8	697.40	36263	292 1/8	697.74	36347
214 1/8	630.68	31652	224 1/8	650.31	33654	289 1/8	698.16	36420	297 1/8	698.55	36500
215 1/8	631.10	31731	225 1/8	650.71	33694	294 1/8	699.06	36580	302 1/8	699.57	36662
216 1/8	631.50	31770	226 1/8	651.10	33735	299 1/8	699.96	36730	307 1/8	699.96	36804
217 1/8	632.06	31810	227 1/8	651.50	33775	304 1/8	700.75	36880	312 1/8	701.14	36964
218 1/8	632.56	31849	228 1/8	651.89	33816	309 1/8	701.50	36984	317 1/8	701.83	37062
219 1/8	633.05	31889	229 1/8	652.28	33857	314 1/8	702.28	37099	322 1/8	702.97	37197
220 1/8	633.43	31928	230 1/8	652.67	33898	319 1/8	703.07	37193	327 1/8	703.84	37286
221 1/8	633.83	31968	231 1/8	653.06	33939	324 1/8	703.85	37282	332 1/8	704.57	37374
222 1/8	634.22	32007	232 1/8	653.45	33980	329 1/8	704.50	37467	337 1/8	705.30	37568
223 1/8	634.60	32047	233 1/8	653.85	34020	334 1/8	705.11	37557	342 1/8	705.71	37650
224 1/8	635.00	32086	234 1/8	654.25	34061	339 1/8	706.10	37630	347 1/8	706.30	37737
225 1/8	635.40	32126	235 1/8	654.64	34102	344 1/8	706.50	37719	352 1/8	707.39	37817
226 1/8	635.79	32165	236 1/8	655.04	34143	349 1/8	707.18	37804	357 1/8	707.78	37894
227 1/8	636.18	32206	237 1/8	655.42	34184	354 1/8	707.67	37984	362 1/8	708.28	38002
228 1/8	636.57	32246	238 1/8	655.82	34225	359 1/8	708.07	38098	367 1/8	708.57	38187
229 1/8	637.04	32286	239 1/8	656.22	34266	364 1/8	708.46	38196	372 1/8	709.14	38284
230 1/8	637.44	32326	240 1/8	656.59	34307	369 1/8	709.54	38347	377 1/8	709.85	38437
231 1/8	637.82	32366	241 1/8	656.99	34348	374 1/8	709.85	38437	382 1/8	710.25	38520
232 1/8	638.21	32405	242 1/8	657.39	34389	379 1/8	710.25	38520	387 1/8	710.64	38600
233 1/8	638.54	32445	243 1/8	657.78	34431	384 1/8	710.64	38632	392 1/8	711.04	38717
234 1/8	638.89	32485	244 1/8	658.17	34472	389 1/8	711.42	38717	397 1/8	711.82	38814
235 1/8	639.22	32525	245 1/8	658.56	34513	394 1/8	712.22	38851	402 1/8	712.82	38950
236 1/8	639.72	32565	246 1/8	658.96	34554	399 1/8	713.62	38985	407 1/8	713.82	39044
237 1/8	640.11	32605	247 1/8	660.35	34595	404 1/8	714.50	39098	412 1/8	714.88	3919

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## Capacities of Cylinders and Spheres

Diam. in Foot	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	12 Gallon Barrels per Foot of Cylinder	Sphere Surface in Cu. Ft.	Diam. in Foot	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Cu. Ft.	Sphere Volume in Cu. Ft.	
.16	.0002	.00143	.000034	.000077	.000002	.215	4.9087	.36.720	.87428	.19.635	.8.1812
.16	.0008	.00574	.000137	.000307	.000018	.216	5.1572	.38.579	.91854	.20.629	.8.8103
.16	.0031	.02295	.000546	.01227	.000128	.216	5.4119	.40.484	.96390	.21.648	.9.4708
.16	.0069	.05184	.00123	.02761	.000431	.217	5.8727	.42.434	.1.0103	.22.691	.10.164
.16	.0123	.09180	.00219	.04909	.00102	.217	5.9396	.44.431	.1.0578	.23.758	.10.889
.16	.0192	.14344	.00342	.07670	.00200	.218	6.2126	.46.474	.1.1068	.24.850	.11.849
.16	.0276	.20655	.00492	.11045	.00345	.218	6.4918	.48.562	.1.1682	.25.957	.12.443
.16	.0376	.28114	.00669	.16033	.00548	.219	6.7771	.50.696	.1.2071	.27.109	.13.272
.16	.0491	.36720	.00874	.19635	.00818						
.16	.0621	.46474	.01107	.24850	.01165	3	7.0886	.82.877	.1.2590	.28.274	.14.137
.16	.0767	.57375	.01388	.30690	.01598	3	7.3662	.65.103	.1.3120	.29.466	.15.039
.16	.0928	.69424	.01653	.37122	.02127	3	7.6599	.57.376	.1.3661	.30.680	.15.979
.16	.1104	.82620	.01967	.44179	.02761	3	7.9788	.59.693	.1.4213	.31.819	.16.957
.16	.1296	.96964	.02309	.51849	.03511	3	8.2958	.62.057	.1.4778	.33.183	.17.974
.16	.1503	.12465	.02677	.60132	.04385	3	8.6179	.64.466	.1.5349	.34.472	.19.031
.16	.1726	.19099	.03074	.69029	.05393	3	8.9482	.66.822	.1.5934	.35.785	.20.128
.16	.1953	.24688	.03487	.78540	.06545	3	9.2680	.69.424	.1.6529	.37.122	.21.268
.16	.2217	.35681	.03948	.88664	.07850	3	9.6211	.71.971	.1.7136	.38.485	.22.449
.16	.2485	.48589	.04426	.99402	.09319	3	10.321	.77.204	.1.8382	.41.282	.24.942
.16	.2769	.70712	.04932	.1.1075	.10960	3	11.045	.82.620	.1.9671	.44.179	.27.612
.16	.3068	.2950	.05464	.1.2272	.12783	3	11.793	.88.220	.2.1005	.47.173	.30.466
.16	.3382	.2.8302	.06024	.1.3530	.14798						
.16	.3712	.2.7769	.06612	.1.4849	.17014	4	12.566	.94.003	.2.3382	.50.268	.33.510
.16	.4057	.3.0351	.07227	.1.6230	.19442	4	13.384	.99.970	.2.3802	.53.456	.36.781
.16	.4418	.3.3048	.07869	.1.7671	.22089	4	14.186	.106.12	.2.5257	.58.745	.40.194
.16	.4784	.3.5859	.08538	.1.9175	.24967	4	15.033	.112.45	.2.6776	.60.132	.43.846
.16	.5185	.3.8785	.09235	.2.0739	.2.0805	4	15.904	.118.97	.2.8327	.63.817	.47.713
.16	.5591	.4.1826	.09959	.2.2365	.31481	4	16.800	.125.67	.2.9922	.67.201	.51.800
.16	.6013	.4.4982	.10710	.2.4063	.35077	4	17.721	.132.56	.3.1562	.70.862	.66.115
.16	.6450	.4.8252	.11489	.2.6802	.38971	4	18.686	.139.63	.3.2428	.74.682	.70.683
.16	.6903	.5.1637	.12295	.2.7612	.43143						
.16	.7371	.5.6137	.13128	.2.9483	.47603	5	19.635	.146.88	.3.4971	.78.540	.65.450
1	.7854	.8.8762	.13989	.3.1416	.52360	5	20.629	.154.32	.3.8742	.82.516	.70.482
1	.8866	.8.6325	.15792	.3.5466	.62804	5	21.648	.161.93	.3.8556	.86.590	.75.768
1	.9940	.7.4358	.17704	.3.9761	.74551	5	22.691	.169.74	.4.0414	.90.763	.81.308
1	1.1075	.8.2849	.19726	.4.4301	.87680	5	23.758	.177.27	.4.2315	.95.033	.87.114
1	1.2272	.9.1800	.21857	.4.9087	.1.0227	5	24.850	.185.89	.4.4261	.98.402	.93.189
1	1.3530	.10.121	.24097	.5.4119	.1.1838	5	25.967	.194.25	.4.6250	.103.87	.98.541
1	1.4849	.11.108	.26447	.5.9396	.1.3612	5	27.109	.202.79	.4.8282	.108.43	.106.17
1	1.6230	.12.141	.28906	.6.4916	.1.5853						
1	1.7671	.13.219	.31474	.7.0686	.1.7671	6	28.274	.211.51	.5.0369	.113.10	.113.10
1	1.9175	.14.344	.34152	.7.8699	.1.9974	6	29.465	.220.41	.5.2479	.117.86	.120.31
1	2.0739	.15.514	.36938	.8.2958	.2.2468	6	30.680	.229.50	.5.4643	.122.72	.127.83
1	2.2385	.16.731	.39835	.8.9462	.2.5161	6	31.919	.238.77	.5.6850	.127.68	.135.66
1	2.4053	.17.993	.42840	.9.6211	.2.8062	6	33.183	.248.23	.8.9102	.132.73	.143.79
1	2.5802	.19.301	.45955	.10.321	.3.1177	6	34.472	.257.87	.8.1397	.137.88	.152.28
1	2.7612	.20.655	.49178	.11.045	.3.4615	6	35.785	.267.69	.8.3735	.143.14	.161.03
1	2.9483	.22.055	.52512	.11.793	.3.6082	6	37.122	.277.69	.6.6118	.148.49	.170.14
2	3.1416	.23.501	.55954	.12.586	.4.1866	7	38.485	.287.88	.6.8544	.153.94	.178.89
2	3.3410	.24.992	.59606	.13.364	.4.5939	7	39.871	.298.26	.7.1014	.159.48	.188.39
2	3.5466	.26.530	.63167	.14.186	.5.0243	7	41.282	.308.81	.7.3527	.166.13	.199.63
2	3.7683	.28.114	.66937	.16.033	.5.4808	7	42.718	.319.56	.7.6085	.170.87	.210.03
2	3.9781	.29.743	.70817	.15.904	.5.9641	7	44.179	.330.48	.7.8686	.176.71	.220.89
2	4.2000	.31.418	.74806	.16.800	.6.4761	7	45.664	.341.59	.8.1330	.182.65	.232.12
2	4.4301	.33.140	.78904	.17.721	.7.0144	7	47.173	.352.88	.8.4019	.188.69	.243.73
2	4.6864	.34.907	.83112	.18.665	.7.6829	7	48.707	.364.30	.8.6751	.194.83	.265.71

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## (Continued). Capacities of Cylinders and Spheres

Diam. in Foot	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	12 Gallon Barrels per Foot of Cylinder	Sphere Surface in Cu. Ft.	Diam. in Foot	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Cu. Ft.	Sphere Volume in Cu. Ft.	
8	.50.265	.376.01	8.9527	.201.06	268.08	18	.254.47	.1903.6	45.323	.1017.9	.3053.6
8	51.849	.387.85	8.2346	.207.39	268.85	18	261.59	.1958.0	40.591	.1046.3	.3182.8
8	53.456	.399.88	9.5209	.213.82	294.01	18	273.80	.2010.8	47.876	.1075.2	.3318.2
8	55.088	.412.09	9.8110	.220.35	307.58	18	276.12	.2065.5	49.178	.1104.5	.3451.5
8	56.745	.424.48	10.107		226.98	21	281.56	.2120.9	50.499	.1134.1	.3591.4
8	58.426	.437.08	10.406		233.71	21	283.53	.2120.9	50.499	.1134.1	.3591.4
8	60.132	.449.82	10.710		240.53	21	291.04	.2177.1	51.836	.1164.2	.3735.0
8	61.802	.462.76	11.018		247.45	21	298.85	.2234.0	53.191	.1194.8	.3982.4
9	63.617	.475.89	11.331		254.47	21	314.16	.2350.1	55.954	.1226.6	.4188.8
9	65.397	.488.20	11.648		261.58	21	322.06	.2409.2	57.362	.1288.2	.4347.8
9	67.201	.502.70	11.969		268.80	21	330.06	.2469.0	58.787	.1320.3	.4510.9
9	69.029	.515.37	12.295		276.12	21	338.16	.2529.6	60.229	.1352.7	.4677.9
9	70.882	.530.24	12.625		283.53	21	346.36	.2591.0	61.589	.1385.4	.4849.0
9	72.759	.548.59	13.041		303.36	21	354.05	.2654.2	63.167	.1418.6	.5024.3
10	78.540	.587.52	13.989		311.16	21	363.60	.2723.6	65.054	.1453.0	.5387.4
10	82.516	.617.26	14.897		330.06	21	380.13	.2843.6	67.705	.1520.5	.5575.3
10	86.590	.647.74	15.422		346.36	21	388.62	.2908.8	69.262	.1555.3	.5767.5
10	90.673	.680.56	16.166		363.08	21	397.51	.2974.3	70.817	.1600.4	.5984.1
11	98.033	.710.90	19.926		380.13	21	406.49	.3040.8	72.399	.1626.0	.6165.1
11	100.402	.743.58	17.704		397.61	21	415.48	.3106.0	73.999	.1661.9	.6370.6
11	103.87	.776.90	18.800		415.48	21	424.56	.3178.9	75.617	.1698.2	.6580.6
11	108.43	.811.14	19.313		433.74	21	434.74	.3244.6	77.292	.1734.9	.6795.2
12	113.10	.846.03	20.143		452.39	21	443.01	.3314.0	78.804	.1772.1	.7014.4
12	117.86	.881.65	20.092		471.44	21	452.39	.3384.1	80.574	.1809.6	.7238.2
12	122.72	.918.00	21.857		490.87	21	462.39	.3458.0	82.261	.1847.6	.7466.8
12	127.08	.955.08	22.740		510.71	21	471.44	.3526.6	83.966	.1885.7	.7700.1
12	132.73	.992.01									

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(Continued). Capacities of Cylinders and Spheres

Diam. in Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.	Diam. in Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.
30	706.86	5287.7	125.90	2827.4	14137	42	1385.4	10304	246.78	6541.8	38792
30 1/4	718.69	5376.2	128.00	2874.8	14494	42 1/4	1402.0	10488	249.70	6607.9	39489
30 1/2	730.62	5465.4	130.13	2922.5	14886	42 1/2	1418.6	10512	252.67	6674.5	40194
30 3/4	742.64	5555.6	132.27	2970.0	15224	42 3/4	1435.4	10737	255.65	6741.5	40808
31	754.77	5646.1	134.43	3019.1	15699	43	1452.2	10863	258.65	6808.8	41630
31 1/4	766.99	5737.5	136.61	3068.0	15979	43 1/4	1469.1	10990	261.66	6876.6	42360
31 1/2	779.31	5829.7	138.80	3117.2	16366	43 1/2	1486.2	11117	264.70	6944.7	43099
31 3/4	791.73	5922.6	141.01	3166.9	16758	43 3/4	1503.3	11245	267.75	6913.2	43846
32	804.25	6016.2	143.24	3217.0	17157	44	1520.5	11374	270.82	6982.1	44602
32 1/4	816.86	6110.6	145.49	3267.5	17603	44 1/4	1537.9	11504	273.90	6151.4	45367
32 1/2	829.58	6205.7	147.75	3318.3	17974	44 1/2	1555.3	11634	277.01	6221.1	46140
32 3/4	842.39	6301.5	150.04	3369.8	18392	44 3/4	1572.8	11765	280.13	6291.2	46922
33	855.30	6398.1	152.34	3421.2	18817	45	1580.4	11897	283.27	6361.7	47713
33 1/4	868.31	6495.4	154.65	3473.2	19247	45 1/4	1608.2	12030	286.42	6432.6	48513
33 1/2	881.41	6593.4	156.95	3525.7	19685	45 1/2	1626.0	12163	289.60	6503.8	49321
33 3/4	894.62	6692.2	168.34	3578.5	20129	45 3/4	1643.9	12297	292.79	6575.5	50139
34	907.92	6791.7	161.71	3631.7	20580	46	1661.9	12432	296.00	6647.6	50965
34 1/4	921.32	6892.0	164.09	3685.3	21037	46 1/4	1680.0	12567	299.22	6720.1	51800
34 1/2	934.82	6992.9	166.50	3739.3	21501	46 1/2	1698.2	12704	302.47	6792.9	52645
34 3/4	948.42	7094.7	168.92	3793.7	21972	46 3/4	1716.5	12841	305.73	6866.1	53499
35	962.11	7197.1	171.36	3848.5	22449	47	1734.9	12978	309.01	6939.8	54362
35 1/4	975.91	7300.3	173.82	3903.6	22934	47 1/4	1753.5	13117	312.30	7013.8	55234
35 1/2	989.80	7404.2	176.29	3959.2	23425	47 1/2	1772.1	13258	315.62	7088.2	56116
35 3/4	1003.8	7508.9	178.78	4015.2	23924	47 3/4	1790.8	13306	318.98	7163.0	57006
36	1017.9	7614.2	181.29	4071.5	24429	48	1809.6	13356	322.30	7238.2	57906
36 1/4	1032.1	7720.4	183.82	4128.2	24942	48 1/4	1828.5	13678	325.68	7313.8	58815
36 1/2	1046.3	7827.2	186.36	4185.4	25461	48 1/2	1847.5	13820	329.06	7398.9	59734
36 3/4	1060.7	7934.8	188.92	4242.9	25988	48 3/4	1866.5	13963	332.45	7466.2	60663
37	1075.2	8043.1	191.50	4300.8	26522	49	1885.7	14106	335.86	7543.0	61601
37 1/4	1089.8	8152.2	194.10	4358.2	27063	49 1/4	1905.0	14251	339.30	7620.1	62549
37 1/2	1104.5	8262.0	196.71	4417.9	27612	49 1/2	1924.4	14396	342.75	7697.7	63506
37 3/4	1119.2	8372.5	199.35	4477.0	28168	49 3/4	1943.9	14541	346.23	7778.6	64473
38	1134.1	8483.8	201.99	4536.5	28731	50	1963.8	14688	349.71	7854.0	66450
38 1/4	1149.1	8595.8	204.60	4596.3	29302	50 1/4	1983.2	14835	353.22	7932.7	66437
38 1/2	1164.2	8708.5	207.35	4656.6	29880	50 1/2	2003.0	14983	356.74	8011.8	67433
38 3/4	1179.3	8822.0	210.05	4717.3	30468	50 3/4	2022.6	15132	360.28	8091.4	68439
39	1194.6	8938.2	212.77	4778.4	31059	51	2042.8	15281	363.84	8171.3	69456
39 1/4	1210.0	9051.1	215.50	4839.8	31600	51 1/4	2062.9	15432	367.42	8251.8	70489
39 1/2	1225.4	9166.8	218.26	4901.7	32269	51 1/2	2083.1	15682	371.01	8332.3	71619
39 3/4	1241.0	9283.2	221.03	4963.9	32886	51 3/4	2103.3	15734	374.52	8413.4	72665
40	1256.6	9400.3	223.82	5028.5	33510	52	2123.7	15887	378.25	8494.9	73622
40 1/4	1272.4	9518.2	226.62	5089.6	34143	52 1/4	2144.2	16040	381.90	8576.7	74689
40 1/2	1288.2	9636.8	229.45	5153.0	34783	52 1/2	2164.8	16193	385.56	8659.0	75766
40 3/4	1304.2	9756.1	232.29	5216.8	35431	52 3/4	2185.4	16348	389.24	8741.7	76854
41	1320.3	9876.2	235.16	5281.0	36087	53	2206.2	16503	392.94	8824.7	77952
41 1/4	1336.4	9997.0	238.02	5345.8	36751	53 1/4	2227.0	16659	396.65	8908.2	79060
41 1/2	1352.7	10119.1	240.92	5410.6	37423	53 1/2	2248.0	16818	400.39	8992.0	80179
41 3/4	1369.0	10241.1	243.83	5476.0	38104	53 3/4	2269.1	16974	404.14	9076.3	81308

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(Continued). Capacities of Cylinders and Spheres

Diam. in Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.	Diam. in Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.
54	2290.2	17132	407.91	9160.9	82448	68	3421.2	25592	609.34	13685	180633
54 1/4	2311.8	17291	411.69	9245.9	83598	68 1/4	3447.2	25787	613.97	13789	182260
54 1/2	2323.8	17451	415.49	9331.3	84759	68 1/2	3473.2	25982	618.61	13893	183980
54 3/4	2334.3	17611	419.32	9417.1	85931	68 3/4	3499.4	26177	623.27	13998	185723
55	2358.8	17772	423.15	9503.3	87114	69	3625.7	26374	627.95	14103	187479
55 1/4	2379.5	17934	427.01	9589.9	88307	69 1/4	3652.0	26571	632.64	14208	189249
55 1/2	2419.2	18087	430.88	9676.9	89511	69 1/2	3678.5	26769	637.35	14314	191031
55 3/4	2441.1	18260	434.77	9764.3	90728	69 3/4	3704.0	26967	642.08	14420	192827
56	2463.0	18246	438.68	9852.0	91982	68	3831.7	27167	646.83	14527	194636
56 1/4	2485.0	18589	442.61	9940.2	93189	68 1/4	3868.4	27367	651.59	14634	196459
56 1/2	2507.2	18755	446.55	10029	94437	68 1/2	3905.3	27588	656.38	14741	198266
56 3/4	2529.4	18921	450.53	10118	95697	68 3/4	3931.2	27769	661.18	14849	197044
57	2551.8	19088	454.49	10207	96987	69	3973.8	27972	666.99	14957	197007
57 1/4	2574.2	19256	458.48	10297	98248	69 1/4	4003.4	28175	670.83	15066	197883
57 1/2	2596.7	19425	462.50	10387	99541	69 1/2	4035.7	28379	675.68	15178	197773
57 3/4	2619.4	19594	466.53	10477	100845	68 3/4	4062.0	28563	680.55	15284	197787
58	2642.1	19784	470.67	10668	102160	70	4084.5	28788	685.44	15394	198594
58 1/4	2664.9	19935	474.64	10660	103487	70 1/4	4116.0	28994	690.34	15504	198125
58 1/2	2687.8	20106	478.72	10751	104825	70 1/2	4147.6	29190	695.27	15615	198470
58 3/4	2710.9	20279	482.82	10843	106175	70 3/4	4181.4	29409	700.21	15725	198429
59	2734.0	20452	486.94	10936	107538	71	4201.5	29617	705.16	15837	197402
59 1/4	2757.2	20626	490.82	11029	108809	71 1/4	4238.1	29826	710.14	15948	198398
59 1/2	2780.5	20800	495.23	11122	110293	71 1/2	4261.2	30035	715.13	16061	191389
59 3/4	2803.9	20975	499.40	11216	111690	71 3/4	4284.3	30246	720.14	16173	193404
60	2827.4	21151	503.89	11310	113097	72	4307.5	30457	726.17	16286	194332
60 1/4	2851.0	21327	507.79	11404	114517	72 1/4	4339.8	30659	730.21	16399	197475
60 1/2	2874.8	21505	512.02	11499	115948	72 1/2	4372.8	30881	735.27	16513	199532
60 3/4	2898.6	21683	516.26	11594							

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(Continued). Capacities of Cylinders and Spheres

Diam. In Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	12 Gallon Barrels per Foot of Cylinder	Sphere Surface In Sq. Ft.	Sphere Volume In Cu. Ft.	Diam. In Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	12 Gallon Barrels per Foot of Cylinder	Sphere Surface In Sq. Ft.	Sphere Volume In Cu. Ft.
78	4778.4	35745	851.06	19113	248475	90	6361.7	47589	1133.1	25447	381704
78 1/4	4809.0	35974	856.53	19236	250872	90 1/4	6397.1	47854	1139.4	25588	384893
78 1/2	4839.8	36204	862.01	19359	253284	90 1/2	6432.6	48119	1145.7	25730	388101
78 3/4	4870.7	36435	867.51	19483	256712	90 3/4	6468.2	48385	1162.0	25873	391326
79	4901.7	36667	873.02	19607	258155	91	6503.9	48652	1158.4	26016	394669
79 1/4	4932.7	36899	878.56	19731	260613	91 1/4	6539.7	48920	1164.8	26159	397830
79 1/2	4963.8	37133	884.11	19856	263087	91 1/2	6576.5	49189	1171.2	26302	401109
79 3/4	4995.2	37367	889.69	19981	265577	91 3/4	6611.6	49458	1177.6	26446	404406
80	5026.5	37601	895.27	20108	268083	92	6647.6	49728	1184.0	26590	407720
80 1/4	5058.0	37837	900.87	20232	270604	92 1/4	6683.8	49998	1190.4	26735	411053
80 1/2	5089.6	38073	906.49	20358	273141	92 1/2	6720.1	50270	1196.9	26880	414404
80 3/4	5121.2	38310	912.13	20485	275693	92 3/4	6756.4	50542	1203.4	27026	417773
81	5153.0	38547	917.79	20612	278262	93	6792.8	50814	1209.9	27172	421160
81 1/4	5184.9	38785	923.46	20738	280846	93 1/4	6829.5	51088	1216.4	27318	424666
81 1/2	5216.8	39024	929.15	20867	283447	93 1/2	6866.1	51362	1222.9	27465	427980
81 3/4	5248.9	39264	934.86	20998	286053	93 3/4	6902.8	51637	1229.5	27612	431432
82	5281.0	39505	940.59	21124	288696	94	6939.8	51813	1236.0	27759	434893
82 1/4	5313.3	39746	946.33	21263	291344	94 1/4	6976.7	52190	1242.6	27807	438372
82 1/2	5345.6	39988	952.09	21382	294009	94 1/2	7013.8	52467	1249.2	28055	441870
82 3/4	5378.1	40231	957.87	21512	296690	94 3/4	7051.0	52745	1255.8	28204	445386
83	5410.6	40474	963.67	21642	299387	95	7088.2	53024	1262.6	28353	449920
83 1/4	5443.3	40718	969.48	21773	302100	95 1/4	7126.6	53303	1269.1	28502	452474
83 1/2	5476.0	40963	975.32	21904	304830	95 1/2	7163.0	53563	1275.8	28652	456048
83 3/4	5508.8	41209	981.16	22035	307576	95 3/4	7200.6	53864	1282.5	28802	459637
84	5541.8	41455	987.03	22167	310339	96	7238.2	54146	1289.2	28953	463247
84 1/4	5574.8	41702	992.92	22299	313118	96 1/4	7276.0	54428	1295.9	29104	466876
84 1/2	5607.9	41950	998.82	22432	315914	96 1/2	7313.8	54711	1302.6	29265	470523
84 3/4	5641.2	42199	1004.7	22568	318728	96 3/4	7351.6	54995	1309.4	29407	474189
85	5674.5	42448	1010.7	22698	321555	97	7389.8	55280	1316.2	29559	477874
85 1/4	5707.9	42688	1016.5	22832	324401	97 1/4	7428.0	55565	1323.0	29712	481579
85 1/2	5741.6	42949	1022.6	22966	327263	97 1/2	7466.2	55851	1329.8	29865	485302
85 3/4	5775.1	43201	1028.6	23100	330142	97 3/4	7504.9	56136	1336.8	30018	489045
86	5808.8	43453	1034.6	23235	333038	98	7543.0	56425	1343.5	30172	492807
86 1/4	5842.6	43706	1040.6	23371	335951	98 1/4	7581.5	56714	1350.3	30326	496588
86 1/2	5876.5	43960	1046.7	23508	338881	98 1/2	7620.1	57003	1357.2	30481	500388
86 3/4	5910.6	44214	1052.7	23642	341828	98 3/4	7658.9	57292	1364.1	30635	504208
87	5944.7	44469	1058.8	23779	344791	99	7697.7	57583	1371.0	30791	508047
87 1/4	5978.9	44725	1064.9	23916	347772	99 1/4	7736.6	57874	1377.9	30946	511908
87 1/2	6013.2	44982	1071.0	24053	350770	99 1/2	7775.6	58168	1384.9	31103	515784
87 3/4	6047.6	45235	1077.1	24190	353785	99 3/4	7814.8	58458	1391.9	31269	519882
88	6082.1	45497	1083.3	24328	356818	100	7854.0	58752	1398.9	31416	523699
88 1/4	6116.7	45756	1089.4	24457	359866	100 1/4	7893.3	59048	1405.9	31573	527536
88 1/2	6151.4	46016	1095.6	24606	362935	100 1/2	7932.7	59341	1412.9	31731	531492
88 3/4	6186.2	46276	1101.8	24745	366018	100 3/4	7972.2	59636	1419.9	31889	536468
89	5221.1	46537	108.0	24886	369121	101	8011.8	59933	1427.0	32047	539454
89 1/4	5256.1	46799	1114.3	25025	372240	101 1/4	8051.8	60230	1434.0	32206	543480
89 1/2	5291.2	47062	1120.6	25166	375337	101 1/2	8091.4	60528	1441.1	32385	547516
89 3/4	5326.4	47325	1126.8	25306	378531	101 3/4	8131.3	60826	1448.2	32525	551572

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(Continued). Capacities of Cylinders and Spheres

Diam. In Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	12 Gallon Barrels per Foot of Cylinder	Sphere Surface In Sq. Ft.	Sphere Volume In Cu. Ft.	Diam. In Feet	Cu. Ft. per Foot of Cylinder	Gallons per Foot of Cylinder	12 Gallon Barrels per Foot of Cylinder	Sphere Surface In Sq. Ft.	Sphere Volume In Cu. Ft.
102	8171.3	61125	1455.4	32685	556647	114	10207	76354	1818.0	40828	775735
102 1/4	8211.4	61425	1462.8	32846	559743	114 1/4	10282	76689	1825.9	41007	780849
102 1/2	8251.6	61726	1469.7	33006	563859	114 1/2	10297	77025	1833.9	41187	785986
102 3/4	8291.9	62028	1476.8	33168	567994	114 3/4	10342	77362	1841.9	41367	791146
103	8332.3	62330	1484.0	33329	572161	115	10387	77699	1850.0	41548	796328
103 1/4	8372.8	62633	1491.3	33491	576327	115 1/4	10432	78038	1858.0	41728	801533
103 1/2	8413.4	62936	1498.5	33654	580523	115 1/2	10477	78376	1866.1	41910	806760
103 3/4	8454.1	63241	1505.7	33816	584740	115 3/4	10523	78716	1874.2	42091	812010
104	8494.8	63646	1513.0	33979	588977	116	10568	79057	1882.3	42273	817203
104 1/4	8535.8	63852	1520.3	34143	593235	116 1/4	10614	79398	1890.4	42466	822679
104 1/2	8576.7	64189	1527.6	34307	597613	116 1/2	10660	79739	1898.6	42638	827897
104 3/4	8617.8	64466	1534.9	34471	601812	116 3/4	10705	80062	1906.7	42822	833238
105	8658.0	64774	1542.6	34636	606131	117	10751	80425	1914.9	43005	838603
105 1/4	8700.3	65083	1549.0	34801	610471	117 1/4	10797	80709	1923.1	43189	843990
105 1/2	8741.7	65392	1557.0	34987	614831	117 1/2	10843	81114	1931.3	43374	849400
105 3/4	8783.2	65703	1564.3	35133	619213	117 3/4	10890	81460	1939.5	43568	854833
106	8824.7	66014	1571.8	35299	623615	118	10936	81806	1947.8	43744	850290
106 1/4	8866.4	66325	1579.2	36406	628037	118 1/4	10982	82153	1956.0	43929	856769
106 1/2	8908.2	66638	1586.8	36533	632481	118 1/2	11029	82501	1964.3	44115	861272
106 3/4	8950.1	66951	1594.1	36800	636845	118 3/4	11075	82849	1972.6	44301	867988
107	8992.0	67265	1601.6	35988	641431	119	11122	83199	1980.9	44488	862347
107 1/4	9034.1	67580	1609.0	36136	645938	119 1/4	11169	83548	1989.2	44675	867920
107 1/2	9076.3	67895	1616.6	36305	650465	119 1/2	11216	83899	1997.6	44863	893516
107 3/4	9118.5	68211	1624.1	36474	655014	119 3/4	11263	84251	2006.0	45051	891916
108	9160.8	68528	1631.6	36604	659584	120	11310	84603	2014.3	45239	904779
108 1/4	9203.3	68846	1638.2	36813	664175	120 1/4	11357	84956	2022.8	45428	910445
108 1/2	9245.8	69164	1645.8	36984	668787	120 1/2	11404	85309	2031.2	45617	916136
108 3/4	9286.0	69483	1654.4	37184	673421	120 3/4	11452	85664			

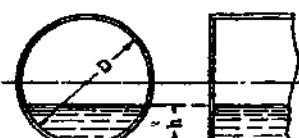
**D-19**  
**(Concluded). Capacities of Cylinders and Spheres**

Diam. in Feet	Fl. Cu. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.	Diam. in Feet	Cu. Fl. per Foot of Cylinder	Gallons per Foot of Cylinder	42 Gallon Barrels per Foot of Cylinder	Sphere Surface in Sq. Ft.	Sphere Volume in Cu. Ft.
126	12469	93274	2220.8	49876	1047394	138	14957	111887	2664.0	59828	1376055
126 1/4	12519	93645	2229.6	50074	1053641	138 1/4	15011	112293	2673.6	60045	1383547
126 1/2	12558	94016	2238.5	50273	1059913	138 1/2	15066	112699	2683.3	60263	1391067
126 3/4	12618	94388	2247.3	50471	1066209	138 3/4	15120	113107	2693.0	60481	1398613
127	12668	94761	2256.2	50671	1072531	139	15175	113514	2702.7	60699	1406187
127 1/4	12718	95134	2265.1	50870	1078877	139 1/4	15229	113923	2712.5	60917	1413788
127 1/2	12758	95508	2274.0	51071	1085248	139 1/2	15284	114333	2722.2	61136	1421416
127 3/4	12818	95883	2282.9	51271	1091645	139 3/4	15339	114743	2732.0	61356	1429072
128	12868	96259	2291.9	51472	1098066	140	15394	115154	2741.8	61575	1436755
128 1/4	12918	96635	2300.8	51673	1104513	140 1/4	15449	115565	2751.6	61795	1444466
128 1/2	12959	97013	2309.8	51875	1110985	140 1/2	15504	115978	2761.4	62016	1452204
128 3/4	13019	97390	2318.8	52077	1117481	140 3/4	15559	116391	2771.2	62237	1459970
129	13070	97769	2327.8	52279	1124004	141	15615	116805	2781.1	62458	1467763
129 1/4	13121	98148	2336.9	52482	1130551	141 1/4	15670	117219	2790.9	62680	1475584
129 1/2	13171	98528	2345.9	52685	1137124	141 1/2	15725	117634	2800.8	62902	1483433
129 3/4	13222	98909	2355.0	52889	1143723	141 3/4	15781	118050	2810.7	63124	1491310
130	13273	99291	2364.1	53093	1150347	142	15837	118467	2820.6	63347	1499214
130 1/4	13324	99673	2373.2	53297	1156996	142 1/4	15893	118885	2830.6	63570	1507146
130 1/2	13376	100056	2382.3	53502	1163677	142 1/2	15948	119303	2840.5	63794	1515107
130 3/4	13427	100440	2391.4	53707	1170371	142 3/4	16005	119722	2850.5	64018	1523095
131	13478	100824	2400.6	53913	1177098	143	16061	120142	2860.5	64242	1531111
131 1/4	13530	101209	2409.7	54119	1183850	143 1/4	16117	120562	2870.5	64467	1539156
131 1/2	13581	101595	2418.9	54325	1190627	143 1/2	16173	120983	2880.5	64692	1547228
131 3/4	13633	101982	2428.1	54532	1197431	143 3/4	16230	121405	2890.6	64918	1555329
132	13685	102369	2437.4	54739	1204260	144	16286	121828	2900.7	65144	1563458
132 1/4	13737	102757	2446.6	54947	1211116	144 1/4	16343	122251	2910.7	65370	1571615
132 1/2	13789	103146	2455.9	55155	1217997	144 1/2	16399	122675	2920.8	65597	1579800
132 3/4	13841	103536	2465.1	55363	1224904	144 3/4	16466	123100	2931.0	65824	1588014
133	13893	103926	2474.4	55572	1231838	145	16513	123526	2941.1	66052	1596256
133 1/4	13945	104317	2483.7	55781	1238797	145 1/4	16570	123952	2951.2	66280	1604527
133 1/2	13998	104709	2493.1	55990	1245783	145 1/2	16627	124379	2961.4	66508	1612826
133 3/4	14050	105102	2502.4	56200	1252795	145 3/4	16684	124807	2971.5	66737	1621154
134	14103	105495	2511.8	56410	1259833	146	16742	125235	2981.8	66966	1629511
134 1/4	14155	105889	2521.2	56621	1266898	146 1/4	16799	125665	2992.0	67196	1637896
134 1/2	14208	106284	2530.6	56832	1273988	146 1/2	16856	126095	3002.3	67426	1646310
134 3/4	14261	106679	2540.0	57044	1281106	146 3/4	16914	126525	3012.5	67656	1654752
135	14314	107075	2549.4	57256	1288249	147	16972	126957	3022.8	67887	1663224
135 1/4	14367	107472	2558.9	57468	1295420	147 1/4	17029	127389	3033.1	68118	1671724
135 1/2	14420	107870	2568.3	57680	1302616	147 1/2	17087	127822	3043.4	68349	1680253
135 3/4	14473	108268	2577.8	57893	1309840	147 3/4	17145	128256	3053.7	68581	1688811
136	14527	108667	2587.3	58107	1317090	148	17203	128690	3064.0	68813	1697398
136 1/4	14580	109067	2596.8	58321	1324366	148 1/4	17262	129125	3074.4	69046	1706015
136 1/2	14634	109468	2606.4	58535	1331670	148 1/2	17320	129561	3084.8	69279	1714660
136 3/4	14687	109869	2615.9	58750	1339000	148 3/4	17378	129998	3095.2	69513	1723334
137	14741	110271	2625.5	58965	1346357	149	17437	130435	3105.6	69746	1732038
137 1/4	14795	110674	2635.1	59180	1353741	149 1/4	17495	130873	3116.0	69981	1740771
137 1/2	14849	111078	2644.7	59396	1361152	149 1/2	17554	131312	3126.5	70215	1749533
137 3/4	14903	111482	2654.3	59612	1368590	149 3/4	17613	131751	3136.9	70450	1758325
						150	17671	132192	3147.4	70686	1767146

**D-20**  
**Tank Capacities, Horizontal Cylindrical**  
**Contents of Tanks with Flat Ends**  
**When Filled to Various Depths**

Diameter of tank inches	Full tank	Depth of liquid, in inches = k															
		3"	6"	9"	12"	15"	18"	21"	24"	27"	30"	33"	36"	39"	42"	45"	48"
12"	5.88	1.15	2.94	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
18"	13.22	1.45	3.96	6.61	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
24"	23.50	1.70	4.60	8.05	11.75	15"	18"	21"	.....	.....	.....	.....	.....	.....	.....	.....	.....
30"	36.72	1.91	5.23	9.27	13.72	18.36	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
36"	52.88	2.12	5.79	10.34	15.47	20.85	26.34	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
42"	71.97	2.28	6.31	11.31	16.97	23.07	29.47	35.09	24"	27"	30"	.....	.....	.....	.....	.....	.....
48"	94.01	2.45	6.78	12.20	18.38	25.10	32.20	39.54	47.00	.....	.....	.....	.....	.....	.....	.....	.....
54"	118.08	2.60	7.22	13.64	19.68	26.07	34.72	42.30	51.08	59.49	.....	.....	.....	.....	.....	.....	.....
60"	146.89	2.75	7.61	13.82	20.91	28.72	37.06	45.82	54.87	64.11	73.44	33"	36"	39"	.....	.....	.....
66"	177.72	2.90	8.04	14.56	22.07	30.37	39.28	48.65	58.39	68.41	78.59	88.26	.....	.....	.....	.....	.....
72"	211.52	3.02	8.42	15.26	23.17	31.82	41.36	51.32	61.78	72.45	83.41	94.54	105.76	.....	.....	.....	.....
78"	248.24	3.15	8.78	15.94	24.21	33.41	43.34	53.86	64.87	76.27	87.97	99.98	111.97	124.13	42"	45"	48"
84"	287.90	3.36	9.12	16.57	25.34	34.85	45.24	56.20	67.87	79.91	92.30	104.98	117.95	130.87	143.95	.....	.....
90"	330.49	3.43	9.16	17.20	26.20	36.21	47.05	58.61	70.75	83.39	96.43	109.81	123.45	137.28	151.23	165.25	.....
96"	376.02	3.50	9.79	17.80	27.13	37.52	48.81	60.84	73.52	86.73	100.39	114.44	128.79	143.40	158.17	173.06	188.01
102"	424.50	3.61	10.10	18.37	28.01	39.00	50.49	62.99	76.18	89.94	104.20	118.89	133.92	149.25	164.81	180.53	196.37
108"	476.10	3.71	10.30	18.94	28.90	40.03	52.14	65.79	78.74	93.04	107.87	123.17	138.87	154.89	171.19	187.71	204.37
114"	530.25	3.78	10.74	19.49	29.75	41.25	53.73	67.10	81.24	96.05	111.43	127.31	143.63	160.33	177.33	194.68	212.05
120"	587.54	3.91	10.98	20.02	30.57	42.39	55.26	69.06	83.65	98.95	114.87	131.32	148.25	165.58	183.27	201.34	219.46

Contents in U.S. gallons per 1 foot of length.  
By permission, The Permutit Co., Inc., Data Book, 1953.

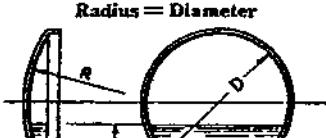


To ascertain the contents of a tank over one-half full: Let  $k$  = depth of unfilled portion. Find from the table the quantity corresponding to a depth  $k$ . Subtract this quantity from the contents of a full tank.

**D-21**

**Tank Capacities, Horizontal Cylindrical**  
**Contents of Standards Dished Heads**  
**When Filled to Various Depths**

Diameter of head inches	Full head	Depth of liquid, in inches = k															
		3"	6"	9"	12"	15"	18"	21"	24"	27"	30"	33"	36"	39"	42"	45"	48"
12"	0.40	0.05	0.20	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
18"	1.36	0.07	0.32	0.68	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
24"	3.22	0.08	0.41	0.95	1.61	15"	18"	21"	.....	.....	.....	.....	.....	.....	.....	.....	.....
30"	6.30	0.10	0.49	1.18	2.10	3.15	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
36"	10.98	0.11	0.56	1.39	2.54	3.92	5.44	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
42"	17.29	0.12	0.63	1.59	2.94	4.64	6.57	8.64	24"	27"	30"	.....	.....	.....	.....	.....	.....
48"	25.79	0.13	0.68	1.75	3.31	5.29	7.62	10.19	12.88	.....	.....	.....	.....	.....	.....	.....	.....
54"	36.72	0.14	0.74	1.90	3.64	5.91	8.60	11.65	14.95	18.36	.....	.....	.....	.....	.....	.....	.....
60"	50.37	0.14	0.82	2.07	3.98	6.49	9.54	13.03	16.87	20.96	25.18	33"	36"	39"	.....	.....	.....
66"	67.04	0.15	0.83	2.19	4.25	6.98	10.35	14.39	18.68	23.43	28.42	33.52	.....	.....	.....	.....	.....
72"	87.84	0.16	0.88	2.12	4.52	7.47	11.15	15.48	20.38	25.74	31.46	37.43	43.52	.....	.....	.....	.....
78"	110.66	0.17	0.93	2.44	4.79	7.97	11.94	16.05	22.02	27.97	34.39	41.16	48.20	55.33	42"	45"	48"
84"	138.22	0.18	0.98	2.50	5.07	8.44	12.60	17.75	21.60	30.11	37.19	44.75	52.67	60.83	69.11	.....	.....
90"	170.01	0.18	1.00	2.68	5.11	8.91	13.44	18.86	25.12	32.18	39.90	48.22	56.99	66.14	75.52	85.00	.....
96"	206.32	0.20	1.07	2.83	5.59	9.36	14.14	19.00	26.80	34.17	42.52	51.53	61.13	71.22	81.66	92.34	103.16
102"	247.48	0.22	1.14	3.01	5.89	9.87	14.92	21.01	28.11	36.18	45.19	54.91	65.31	76.29	87.72	99.56	111.59
108"	293.77	0.20	1.13	3.03	6.04	10.21	15.50	21.93	29.47	38.03	47.56	57.97	69.14	81.05	93.53	106.47	119.76
114"	345.51	0.21	1.16	3.12	6.25	10.55	16.06	22.30	30.70	39.73	49.81	60.88	72.85	85.61	99.05	113.07	127.56
120"	402.27	0.21	1.19	3.23	6.47	10.93	16.58	23.70	31.96	41.43	52.04	63.73	76.40	89.95	104.32	119.39	135.04



To ascertain the contents of a head over one-half full: Let  $k$  = depth of unfilled portion. Find from the table the quantity corresponding to a depth  $k$ . Subtract this quantity from the contents of a full head.

Contents in U.S. gallons for one head only. This table is only approximate, but close enough for practical use.  
By permission, The Permutit Co., Inc., Data Book, 1953.

## D-22

### Miscellaneous Formulas

(Courtesy of Chicago Bridge and iron Co.)

#### 1. Area of Roofs.

##### Umbrella Roofs:

$D$  = diameter of tank in feet.

$$\text{Surface area in } \left\{ \begin{array}{l} = 0.842 D^2 \text{ (when radius = diameter)} \\ = 0.892 D^2 \text{ (when radius = } 0.5 \text{ diameter)} \end{array} \right.$$

##### Couloidal Roofs:

$$\text{Surface area in } \left\{ \begin{array}{l} = 0.787 D^2 \text{ (when pitch is } \frac{3}{4} \text{ in 12)} \\ = 0.792 D^2 \text{ (when pitch is } 1\frac{1}{2} \text{ in 12)} \end{array} \right.$$

#### 2. Average weights.

Steel — 490 pounds per cubic foot—specific gravity 7.85

Wrought iron — 485 pounds per cubic foot—specific gravity 7.77

Cast iron — 450 pounds per cubic foot—specific gravity 7.21

1 cubic foot air or gas at 32° F., 760 mm. barometer = molecular weight  $\times$  0.0027855 pounds.

#### 3. Expansion in steel pipe = 0.78 inch per 100 lineal feet per 100 degrees Fahr. change in temperature = 0.412 inch per mile per degree Fahr. temperature change.

#### 4. Linear coefficients of expansion per degree increase in temperature:

	Per Degree Fahrenheit	Per Degree Centigrade
<b>STRUCTURAL STEEL—A-7</b>		
70° to 200° F.	0.0000067	—
21.1° to 93° C.	—	0.0000121
<b>STAINLESS STEEL—TYPE 304</b>		
32° to 932° F.	0.0000102	—
0° to 500° C.	—	0.0000184
<b>ALUMINUM</b>		
-76° to 68° F.	0.0000120	—
-60° to 20° C.	—	0.0000216

#### 5. To determine the net thickness of shells for horizontal cylindrical pressure tanks:

$$T = \frac{6PD}{S}$$

$P$  = working pressure in pounds per square inch

$D$  = diameter of cylinder in feet

$S$  = allowable unit working stress in pounds per square inch

$T$  = Net thickness in inches

Resulting net thickness must be corrected to gross or actual thickness by dividing by joint efficiency.

#### 6. To determine the net thickness of heads for cylindrical pressure tanks:

##### (6a) Ellipsoidal or Bumped Heads:

$$T = \frac{6PD}{S}$$

$T$ ,  $P$  and  $D$  as in formula 5

##### (6b) Dished or Basket Heads:

$$T = \frac{10.6P(MR)}{S}$$

$T$ ,  $S$  and  $P$  as in formula 5

$MR$  = principal radius of head in feet

Resulting net thickness of heads is both net and gross thickness if one piece seamless heads are used, otherwise net thickness must be corrected to gross thickness as above.

Formulas 5 and 6 must often be modified to comply with various engineering codes, and state and municipal regulations. Calculated gross plate thicknesses are sometimes arbitrarily increased to provide an additional allowance for corrosion.

#### 7. Heads for Horizontal Cylindrical Tanks:

*Hemi-ellipsoidal Heads* have an ellipsoidal cross section, usually with minor axis equal to one half the major axis—that is, depth =  $\frac{1}{4} D$ , or more.

*Dished or Basket Heads* consist of a spherical segment normally dished to a radius equal to the inside diameter of the tank cylinder (or within a range of 6 inches plus or minus) and connected to the straight cylindrical flange by a "knuckle" whose inside radius is usually not less than 6 per cent of the inside diameter of the cylinder nor less than 3 times the thickness of the head plate. Basket heads closely approximate hemi-ellipsoidal heads.

*Bumped Heads* consist of a spherical segment joining the tank cylinder directly without the transition "knuckle." The radius =  $D$ , or less. This type of head is used only for pressures of 10 pounds per square inch or less, excepting where a compression ring is placed at the junction of head and shell.

##### Surface Area of Heads:

###### (7a) Hemi-ellipsoidal Heads:

$$S = \pi R^2 [1 + K^2(2-K)]$$

$S$  = surface area in square feet

$R$  = radius of cylinder in feet

$K$  = ratio of the depth of the head (not including the straight flange) to the radius of the cylinder

The above formula is not exact but is within limits of practical accuracy.

###### (7b) Dished or Basket Heads:

Formula (7a) gives surface area within practical limits.

###### (7c) Bumped Heads:

$$S = \pi R^2 (1 + K^2)$$

$S$ ,  $R$ , and  $K$  as in formula (7a)

##### Volume of Heads:

###### (7d) Hemi-ellipsoidal Heads:

$$V = \frac{1}{2} \pi K R^4$$

$R$  = radius of cylinder in feet

$K$  = ratio of the depth of the head (not including the straight flange) to the radius of the cylinder

###### (7e) Dished or Basket Heads:

Formula (7d) gives volume within practical limits.

###### (7f) Bumped Heads:

$$V = \frac{1}{2} \pi R^4 (1 + \frac{1}{3} K^2)$$

$V$ ,  $K$  and  $R$  as in formula (7d)

Note:  $K$  in above formulas may be determined as follows:

Hemi-ellipsoidal heads— $K$  is known

$$\text{Dished Heads—} K = M - \sqrt{(M-1)(M+1-2m)}$$

$$\text{Bumped Heads—} K = [M - \sqrt{M^2-1}]$$

$MR$  = principal radius of head in feet

$mR$  = radius of knuckle in feet

$R$  = radius of cylinder in feet

$$M = \frac{MR}{R} \quad m = \frac{mR}{R}$$

For bumped heads  $m = v$

#### 8. Total volume or length of shell in cylindrical tank with ellipsoidal or hemispherical heads:

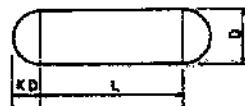
$V$  = Total volume

$L$  = Length of cylindrical shell

$KD$  = Depth of head

$$V = \frac{\pi D^2}{4} (L + 1\frac{1}{3} KD)$$

$$L = (V \div \frac{\pi D^2}{4}) - 1\frac{1}{3} KD$$

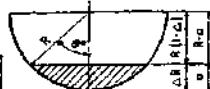


**D-22**  
**(Continued). Miscellaneous Formulas**

9. Volume or contents of partially filled horizontal cylindrical tanks:

(9a) Tank cylinder or shell (straight portion only)

$$Q = RL \left[ \left( \frac{\pi \theta}{180} \right) - \sin \theta \cos \theta \right]$$



$Q$  = partially filled volume or contents in cubic feet

$R$  = radius of cylinder in feet

$L$  = length of straight portion of cylinder in feet

The straight portion or flange of the heads must be considered a part of the cylinder. The length of flange depends upon the diameter of tank and thickness of head but ranges usually between 2 and 4 inches.

$a = \Delta R$  = depth of liquid in feet

$$\Delta = \frac{a}{R} = \text{a ratio}$$

$$\cos \theta = 1 - \Delta, \text{ or } \frac{R-a}{R}$$

$\theta$  = degrees

(9b) Hemi-ellipsoidal Heads:

$$Q = \frac{1}{2} V \Delta^2 (1 - \frac{1}{4} \Delta)$$

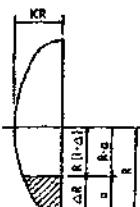
$Q$  = partially filled volume or contents in cubic feet

$V$  = total volume of one head per formula (7d)

$$\Delta = \frac{a}{R} = \text{a ratio}$$

$a = \Delta R$  = depth of liquid in feet

$R$  = radius of cylinder in feet



(9c) Dished or Basket Heads:

Formula (9b) gives partially filled volume within practical limits, and formula (7d) gives  $V$  within practical limits.

(9d) Bumped Heads:

Formula (9b) gives partially filled volume within practical limits, and formula (7f) gives  $V$ .

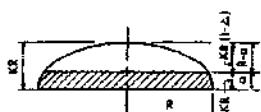
Note: To obtain the volume or quantity of liquid in partially filled tanks, add the volume per formula (9a) for the cylinder or straight portion to twice (for 2 heads) the volume per formula (9b), (9c) or (9d) for the type of head concerned.

10. Volume or contents of partially filled hemi-ellipsoidal heads with major axis vertical:

$Q$  = Partially filled volume or contents in cubic feet

$V$  = Total volume of one head per formula (7d)

$R$  = Radius of cylinder in feet

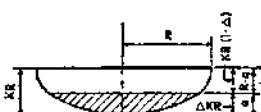


(10a) Upper Head:

$$Q = 1\frac{1}{2} V \Delta (1 - \frac{1}{2} \Delta^2)$$

$$\Delta = \frac{a}{KR} = \text{a ratio}$$

$a = \Delta KR$  = depth of liquid in feet



(10b) Lower Head:

$$Q = 1\frac{1}{2} V \Delta^2 (1 - \frac{1}{2} \Delta)$$

$$\Delta = \frac{a}{KR} = \text{a ratio}$$

$a = \Delta KR$  = depth of liquid in feet

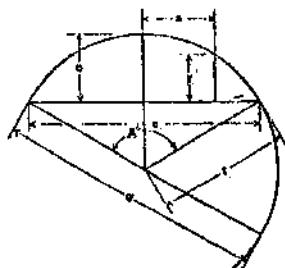
**D-23**  
**Decimal Equivalent in Inches, Feet and Millimeters**

In. Equiv. for Decimal of In.	Decimals	Millimeter Equiv. for Decimal of In.	In. Equiv. for Decimal of Ft.
1/64	.0156	0.397	3/64
1/32	.0313	0.794	7/32
3/64	.0469	1.191	9/64
1/16	.0625	1.588	3/16
5/64	.0781	1.984	13/64
3/32	.0938	2.381	11/32
7/64	.1094	2.778	15/64
1/8	.1250	3.175	11/16
9/64	.1406	3.572	13/16
5/32	.1563	3.969	17/32
11/64	.1719	4.366	21/32
3/16	.1875	4.763	21/4
13/64	.2031	5.159	25/32
7/32	.2188	5.556	25/4
15/64	.2344	5.953	213/16
1/4	.2500	6.350	3
17/64	.2656	6.747	33/16
9/32	.2813	7.144	35/32
19/64	.2969	7.541	39/16
5/16	.3125	7.938	37/4
21/64	.3281	8.334	315/16
11/32	.3438	8.731	41/8
23/64	.3594	9.128	49/16
7/8	.3750	9.525	45/8
25/64	.3906	9.922	411/16
13/32	.4063	10.319	47/8
27/64	.4219	10.716	51/16
7/16	.4375	11.113	53/8
29/64	.4531	11.509	55/16
15/32	.4688	11.906	57/8
31/64	.4844	12.303	513/16
1/2	.5000	12.700	6
33/64	.5156	13.097	65/16
17/32	.5313	13.494	67/8
35/64	.5469	13.891	69/16
9/16	.5625	14.288	67/4
37/64	.5781	14.684	615/16
19/32	.5938	15.081	71/8
39/64	.6094	15.478	75/16
7/8	.6250	15.875	77/8
41/64	.6406	16.272	711/16
21/32	.6563	16.669	79/8
43/64	.6719	17.066	81/16
13/16	.6875	17.463	83/4
45/64	.7031	17.859	87/16
23/32	.7188	18.256	89/8
47/64	.7344	18.653	813/16
3/4	.7500	19.050	9
49/64	.7656	19.447	93/16
25/32	.7813	19.844	95/8
51/64	.7969	20.241	99/16
13/16	.8125	20.638	99/8
53/64	.8281	21.034	915/16
27/32	.8438	21.431	101/8
55/64	.8594	21.828	105/16
7/8	.8750	22.225	107/8
57/64	.8906	22.622	1011/16
29/32	.9063	23.019	107/4
59/64	.9219	23.416	113/16
7/5	.9375	23.813	117/4
61/64	.9531	24.209	117/16
31/32	.9688	24.606	119/8
63/64	.9844	25.003	1113/16
1	1.0000	25.400	12

D-24

(by permission of Buffalo Tank Div., Bethlehem Steel Corp.)

**PROPERTIES OF THE CIRCLE**

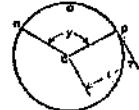


$$\begin{aligned} \text{Circumference} &= 2\pi r = 3.14159 d \\ \text{Diameter} &= 0.31831 \text{ circumference} \\ \text{Area} &= 3.14159 r^2 \end{aligned}$$

$$\begin{aligned} \text{Arc } a &= \frac{\pi r^2}{360^\circ} = 0.017453 r A^\circ \\ \text{Angle } A^\circ &= \frac{180^\circ r}{\pi} = 57.29578 \frac{r}{\pi} \\ \text{Radius } r &= \frac{c^2 + e^2}{2b} \\ \text{Chord } c &= 2\sqrt{2br - b^2} = 2r \sin \frac{A}{2} \\ \text{Rise } b &= r - \sqrt{r^2 - c^2} = \frac{r}{2} \tan \frac{A}{4} \\ y &= b - r + \sqrt{r^2 - x^2} \\ x &= \sqrt{r^2 - (r - y)^2} \end{aligned}$$

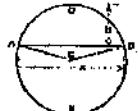
Diameter of circle of equal periphery as square = 1.27324 side of square  
 Side of square of equal periphery as circle = 0.78540 diameter of circle  
 Diameter of circle circumscribed about square = 1.41421 side of square  
 Side of square inscribed in circle = 0.70711 diameter of circle

**CIRCULAR SECTOR**



$$\begin{aligned} r &= \text{radius of circle} \quad y = \text{angle } \theta \text{ in degrees} \\ \text{Area of Sector } \theta^\circ &= \frac{1}{2} (\text{length of arc } \theta^\circ \times r) \\ &= \text{Area of Circle} \times \frac{\theta}{360^\circ} \\ &= 0.0087268 \times r^2 \times \theta \end{aligned}$$

**CIRCULAR SEGMENT**



$$\begin{aligned} r &= \text{radius of circle} \quad x = \text{chord} \quad b = \text{rise} \\ \text{Area of Segment } \theta^\circ &= \text{Area of Sector } \theta^\circ - \text{Area of triangle } \theta^\circ \\ &= \frac{(\text{Length of arc } \theta^\circ \times r) - x(r - b)}{2} \\ \text{Area of Segment } \theta^\circ &= \text{Area of Circle} - \text{Area of Segment } \theta^\circ \end{aligned}$$

**VALUES FOR FUNCTIONS OF  $\pi$**

$$\begin{aligned} \pi &= 3.141592653589, \log = 0.4971499 \\ \pi^2 &= 9.8699044, \log = 0.9942988 \quad \frac{1}{\pi} = 0.3183099, \log = 7.6028301 \quad \sqrt{\frac{1}{\pi}} = 0.5641894, \log = 7.7854881 \\ \pi^4 &= 31.0082767, \log = 1.4916497 \quad \frac{1}{\pi^2} = 0.1013212, \log = 7.0057002 \quad \frac{1}{\pi^4} = 0.0174533, \log = 7.2419774 \\ \sqrt{\pi} &= 1.7724539, \log = 0.2485749 \quad \frac{1}{\pi^3} = 0.0322818, \log = 7.0086503 \quad \frac{1}{\pi^6} = 17.2967791, \log = 7.7861820 \end{aligned}$$

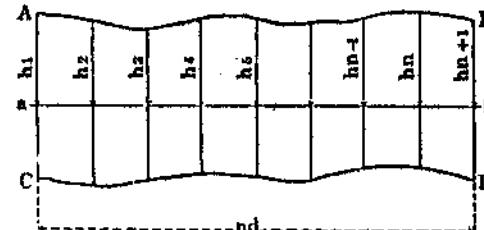
D-24

(Continued).

**AREA OF PLANE FIGURES**

<b>Triangle:</b>	Base $\times \frac{1}{2}$ perpendicular height.
	$\sqrt{s(s-a)(s-b)(s-c)}$ , where $s = \frac{1}{2}$ sum of the three sides $a, b$ and $c$ .
<b>Trapezium:</b>	Sum of area of the two triangles.
<b>Trapezoid:</b>	$\frac{1}{2}$ sum of parallel sides $\times$ perpendicular height.
<b>Parallelogram:</b>	Base $\times$ perpendicular height.
<b>Regular Polygons:</b>	$\frac{1}{2}$ sum of sides $\times$ inside radius.
<b>Circle:</b>	$\pi r^2 = 0.78540 \times \text{dia.}^2 = 0.07958 \times \text{circumference}^2$
<b>Sector of Circle:</b>	$\frac{\pi r^2 A^\circ}{360^\circ} = 0.0087268 r^2 A^\circ = \text{arc } \times \frac{1}{2} \text{ radius.}$
<b>Segment of Circle:</b>	$\frac{1}{2} \left( \frac{\pi \Delta^2}{360^\circ} - \sin \Delta^\circ \right)$
<b>Circle of same area as square:</b>	diameter = side $\times 1.12838$
<b>Square of same area as circle:</b>	side = diameter $\times 0.88623$
<b>Ellipse:</b>	Long diameter $\times$ short diameter $\times 0.78540$
<b>Parabola:</b>	Base $\times \frac{3}{8}$ perpendicular height.

Irregular plane surface

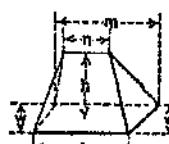


Divide any plane surface A, B, C, D, along a line a-b into an even number,  $n$ , of parallel and sufficiently small strips,  $d$ , whose ordinates are  $h_1, h_2, h_3, h_4, \dots, h_{n-1}, h_n, h_{n+1}$ , and considering contours between three ordinates as parabolic curves, then for section ABCD,

$$\text{Area} = \frac{d}{3} \left[ h_1 + h_{n+1} + 4(h_2 + h_4 + h_6 + \dots + h_{n-2}) + 2(h_3 + h_5 + h_7 + \dots + h_{n-1}) \right]$$

or, approximately, Area = Sum of ordinates  $\times$  width,  $d$ .

**VOLUME OF A WEDGE**



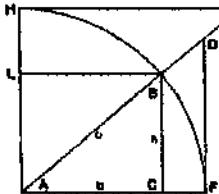
This formula is useful in obtaining the contents of special, wedge-shaped, tank bottoms.

$$\text{Volume} = \frac{wh}{6} (l + m + n)$$

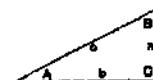
**D-24**  
(Continued).

**TRIGONOMETRIC FORMULAS**

**TRIGONOMETRIC FUNCTIONS**



**RIGHT ANGLED TRIANGLES**

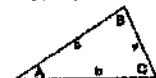


$$\begin{aligned}a^2 &= c^2 - b^2 \\b^2 &= c^2 - a^2 \\c^2 &= a^2 + b^2\end{aligned}$$

Radius AF	= 1
	= $\sin^2 A + \cos^2 A = \sin A \cos A$
	= $\cos A \sec A = \tan A \cot A$
Sine A	= $\frac{\sin A}{\cos A} = \frac{1}{\sec A} = \cos A \tan A = \sqrt{1 - \cos^2 A} = BG$
Cosine A	= $\frac{\cos A}{\sin A} = \frac{1}{\csc A} = \sin A \cot A = \sqrt{1 - \sin^2 A} = AG$
Tangent A	= $\frac{\sin A}{\cos A} = \frac{1}{\cos A} = \sin A \sec A = FD$
Cotangent A	= $\frac{\cos A}{\sin A} = \frac{1}{\sin A} = \cos A \csc A = HG$
Secant A	= $\frac{1}{\cos A} = \frac{1}{\sin A} = AD$
Cosecant A	= $\frac{1}{\sin A} = \frac{1}{\cos A} = AG$

Known	Required					
	A	B	a	b	c	Area
a, b	$\tan A = \frac{a}{b}$	$\tan B = \frac{b}{a}$			$\sqrt{a^2 + b^2}$	$\frac{ab}{2}$
a, c	$\sin A = \frac{a}{c}$	$\cos B = \frac{a}{c}$			$\frac{a\sqrt{c^2 - a^2}}{c}$	
A, a	$90^\circ - A$			$a \cot A$	$\frac{a}{\sin A}$	$\frac{a^2 \cot A}{2}$
A, b	$90^\circ - A$		$b \tan A$		$\frac{b}{\cos A}$	$\frac{b^2 \tan A}{2}$
A, c	$90^\circ - A$		$a \sin A$	$b \cos A$		$\frac{ab \sin 2A}{4}$

**OBLIQUE ANGLED TRIANGLES**



$$\begin{aligned}s &= \frac{a+b+c}{2} \\a^2 &= b^2 + c^2 - 2bc \cos A \\b^2 &= a^2 + c^2 - 2ac \cos B \\c^2 &= a^2 + b^2 - 2ab \cos C\end{aligned}$$

Known	Required					
	A	B	C	a	b	Area
a, b, c	$\cos \frac{1}{2} A = \sqrt{\frac{(s-a)(s-b)}{bc}}$	$\cos \frac{1}{2} B = \sqrt{\frac{(s-a)(s-c)}{ac}}$	$\cos \frac{1}{2} C = \sqrt{\frac{(s-b)(s-c)}{ab}}$			$\sqrt{s(s-a)(s-b)(s-c)}$
a, A, B			$180^\circ - (A+B)$	$\frac{a \sin B}{\sin A}$	$\frac{a \sin C}{\sin A}$	
a, b, A		$\sin B = \frac{b \sin A}{a}$		$\frac{b \sin C}{\sin B}$		
a, b, C	$\tan A = \frac{a \sin C}{b - a \cos C}$			$\sqrt{a^2 + b^2 - 2ab \cos C}$	$\frac{ab \sin C}{2}$	

**D-24**  
(Continued).

**PROPERTIES OF SECTIONS**

**SQUARE**

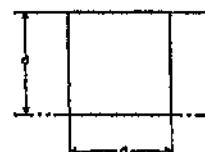
Axis of moments through center



$$\begin{aligned}A &= d^2 \\c &= \frac{d}{2} \\I &= \frac{d^4}{12} \\S &= \frac{d^2}{3} \\r &= \frac{d}{\sqrt{12}} = .288675 d\end{aligned}$$

**SQUARE**

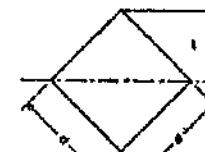
Axis of moments on base



$$\begin{aligned}A &= d^2 \\c &= d \\I &= \frac{d^4}{3} \\S &= \frac{d^2}{3} \\r &= \frac{d}{\sqrt{3}} = .577350 d\end{aligned}$$

**SQUARE**

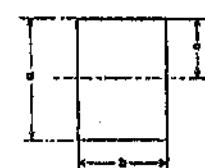
Axis of moments on diagonal



$$\begin{aligned}A &= d^2 \\c &= \frac{d}{\sqrt{2}} = .707107 d \\I &= \frac{d^4}{12} \\S &= \frac{d^2}{8\sqrt{2}} = .117851 d^2 \\r &= \frac{d}{\sqrt{12}} = .288675 d\end{aligned}$$

**RECTANGLE**

Axis of moments through center



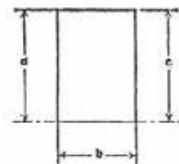
$$\begin{aligned}A &= bd \\c &= \frac{d}{2} \\I &= \frac{bd^3}{12} \\S &= \frac{bd^2}{3} \\r &= \frac{d}{\sqrt{12}} = .288675 d\end{aligned}$$

**D-24**  
(Continued).

### PROPERTIES OF SECTIONS

#### RECTANGLE

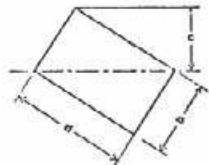
Axis of moments on base



$$\begin{aligned} A &= bd \\ c &= d \\ I &= \frac{bd^3}{3} \\ S &= \frac{bd^2}{3} \\ r &= \sqrt{\frac{d^2}{3}} = .577350 d \end{aligned}$$

#### RECTANGLE

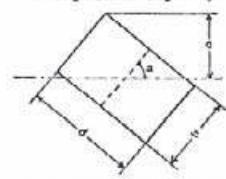
Axis of moments on diagonal



$$\begin{aligned} A &= bd \\ c &= \sqrt{b^2 + d^2} \\ I &= \frac{b^2 d^2}{6(b^2 + d^2)} \\ S &= \frac{b^2 d^2}{6\sqrt{b^2 + d^2}} \\ r &= \frac{bd}{\sqrt{6(b^2 + d^2)}} \end{aligned}$$

#### RECTANGLE

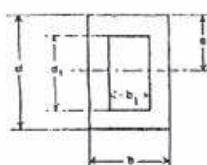
Axis of moments any line through center of gravity



$$\begin{aligned} A &= bd \\ c &= \frac{b \sin \alpha + d \cos \alpha}{2} \\ I &= \frac{bd(b^2 \sin^2 \alpha + d^2 \cos^2 \alpha)}{12} \\ S &= \frac{bd(b^2 \sin^2 \alpha + d^2 \cos^2 \alpha)}{6(b \sin \alpha + d \cos \alpha)} \\ r &= \sqrt{\frac{b^2 \sin^2 \alpha + d^2 \cos^2 \alpha}{12}} \end{aligned}$$

#### HOLLOW RECTANGLE

Axis of moments through center



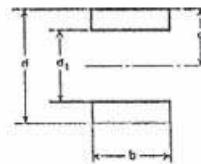
$$\begin{aligned} A &= bd - b_1 d_1 \\ c &= \frac{d}{2} \\ I &= \frac{bd^3 - b_1 d_1^3}{12} \\ S &= \frac{bd^2 - b_1 d_1^2}{8d} \\ r &= \sqrt{\frac{bd^2 - b_1 d_1^2}{12A}} \end{aligned}$$

**D-24**  
(Continued).

### PROPERTIES OF SECTIONS

#### EQUAL RECTANGLES

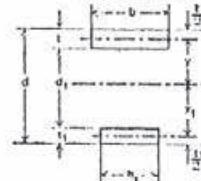
Axis of moments through center of gravity



$$\begin{aligned} A &= b(d - d_1) \\ c &= \frac{d}{2} \\ I &= \frac{b(d^2 - d_1^2)}{12} \\ S &= \frac{b(d^2 - d_1^2)}{8d} \\ r &= \sqrt{\frac{d^2 - d_1^2}{12(d - d_1)}} \end{aligned}$$

#### UNEQUAL RECTANGLES

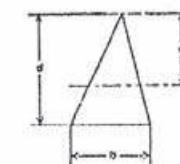
Axis of moments through center of gravity



$$\begin{aligned} A &= bt + b_1 t_1 \\ c &= \frac{t_1 b^2 + b_1 t_1 (d - \frac{1}{2} t_1)}{A} \\ I &= \frac{bt^3}{12} + bty^2 + \frac{b_1 t_1^3}{12} + b_1 t_1 y_1^2 \\ S &= \frac{I}{c} \quad S_1 = \frac{I}{c_1} \\ r &= \sqrt{\frac{I}{A}} \end{aligned}$$

#### TRIANGLE

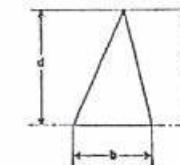
Axis of moments through center of gravity



$$\begin{aligned} A &= \frac{bd}{2} \\ c &= \frac{2d}{3} \\ I &= \frac{bd^3}{36} \\ S &= \frac{bd^2}{24} \\ r &= \frac{d}{\sqrt{18}} = .235702 d \end{aligned}$$

#### TRIANGLE

Axis of moments on base



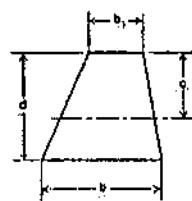
$$\begin{aligned} A &= \frac{bd}{2} \\ c &= d \\ I &= \frac{bd^3}{12} \\ S &= \frac{bd^2}{12} \\ r &= \frac{d}{\sqrt{6}} = .408248 d \end{aligned}$$

**D-24**  
(Continued).

**PROPERTIES OF SECTIONS**

**TRAPEZOID**

Axis of moments through center of gravity



$$A = \frac{d(b + b_1)}{2}$$

$$c = \frac{d(2b + b_1)}{3(b + b_1)}$$

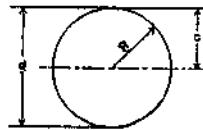
$$I = \frac{d^3(b^2 + 4bb_1 + b_1^2)}{36(b + b_1)}$$

$$S = \frac{d^4(b^2 + 4bb_1 + b_1^2)}{12(2b + b_1)}$$

$$r = \frac{d}{6(b + b_1)} \sqrt{2(b^2 + 4bb_1 + b_1^2)}$$

**CIRCLE**

Axis of moments through center



$$A = \frac{\pi d^2}{4} = \pi R^2 = .785398 R^2 = 3.141593 R^2$$

$$c = \frac{d}{2} = R$$

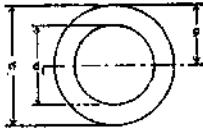
$$I = \frac{\pi d^4}{64} = \frac{\pi R^4}{4} = .049087 R^4 = .785398 R^4$$

$$S = \frac{\pi d^3}{32} = \frac{\pi R^3}{4} = .098176 R^3 = .785398 R^3$$

$$r = \frac{d}{4} = \frac{R}{2}$$

**HOLLOW CIRCLE**

Axis of moments through center



$$A = \frac{\pi(d^2 - d_1^2)}{4} = .785398(d^2 - d_1^2)$$

$$c = \frac{d}{2}$$

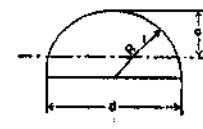
$$I = \frac{\pi(d^4 - d_1^4)}{64} = .049087(d^4 - d_1^4)$$

$$S = \frac{\pi(d^3 - d_1^3)}{32d} = .098176 \frac{d^3 - d_1^3}{d}$$

$$r = \frac{\sqrt{d^2 + d_1^2}}{4}$$

**HALF CIRCLE**

Axis of moments through center of gravity



$$A = \frac{\pi R^2}{2} = 1.570796 R^2$$

$$c = R \left(1 - \frac{4}{3\pi}\right) = .575587 R$$

$$I = R^4 \left(\frac{\pi}{8} - \frac{8}{3\pi}\right) = .108787 R^4$$

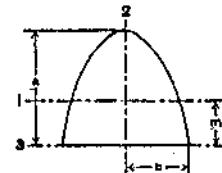
$$S = \frac{R^3}{24} \frac{(8\pi^2 - 64)}{(3\pi - 4)} = .190887 R^3$$

$$r = R \frac{\sqrt{8\pi^2 - 64}}{6\pi} = .264396 R$$

**D-24**  
(Continued).

**PROPERTIES OF SECTIONS**

**PARABOLA**



$$A = \frac{4}{3}ab$$

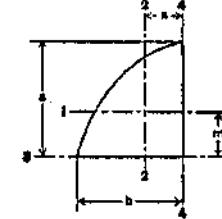
$$c = \frac{2}{3}a$$

$$I_1 = \frac{16}{175}a^2b$$

$$I_2 = \frac{4}{15}ab^2$$

$$I_3 = \frac{32}{105}a^2b$$

**HALF PARABOLA**



$$A = \frac{2}{3}ab$$

$$c = \frac{2}{3}a$$

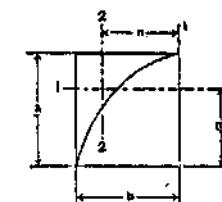
$$I_1 = \frac{2}{175}a^2b$$

$$I_2 = \frac{19}{480}ab^2$$

$$I_3 = \frac{16}{105}a^2b$$

$$I_4 = \frac{2}{15}ab^2$$

**COMPLEMENT OF HALF PARABOLA**



$$A = \frac{1}{3}ab$$

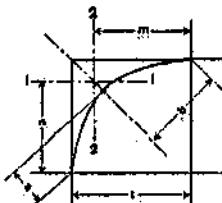
$$c = \frac{7}{10}a$$

$$I_1 = \frac{3}{4}b$$

$$I_2 = \frac{27}{2100}b^3$$

$$I_3 = \frac{1}{90}ab^3$$

**PARABOLIC FILLET IN RIGHT ANGLE**



$$a = \frac{t}{2\sqrt{2}}$$

$$b = \frac{t}{\sqrt{2}}$$

$$A = \frac{1}{8}t^2$$

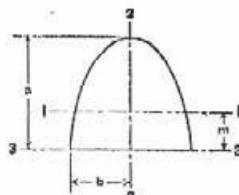
$$m = n = \frac{4}{3}t$$

$$I_1 = I_2 = \frac{11}{315}t^4$$

**D-24**  
(Continued).

**PROPERTIES OF SECTIONS**

\* HALF ELLIPSE



$$A = \frac{1}{2} \pi ab$$

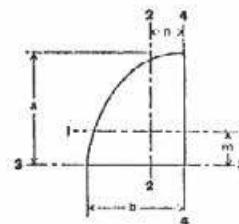
$$m = \frac{4a}{3\pi}$$

$$I_x = a^3 b \left( \frac{\pi}{8} - \frac{1}{9\pi} \right)$$

$$I_y = \frac{1}{8} \pi a^3 b^2$$

$$I_z = \frac{1}{8} \pi a^3 b$$

\* QUARTER ELLIPSE



$$A = \frac{1}{4} \pi ab$$

$$m = \frac{4a}{3\pi}$$

$$n = \frac{4b}{3\pi}$$

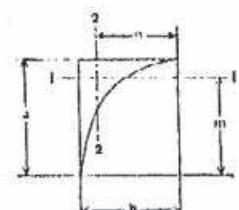
$$I_x = a^3 b \left( \frac{\pi}{16} - \frac{4}{9\pi} \right)$$

$$I_y = ab^3 \left( \frac{\pi}{16} - \frac{4}{9\pi} \right)$$

$$I_z = \frac{1}{16} \pi a^3 b$$

$$I_e = \frac{1}{16} \pi ab^3$$

ELLIPTIC COMPLEMENT



$$A = ab \left( 1 - \frac{\pi}{4} \right)$$

$$m = \frac{\pi}{8} \left( 1 - \frac{\pi}{4} \right)$$

$$n = \frac{b}{8} \left( 1 - \frac{\pi}{4} \right)$$

$$I_x = a^3 b \left( \frac{1}{2} - \frac{\pi}{16} - \frac{1}{36} \left( 1 - \frac{\pi}{4} \right)^2 \right)$$

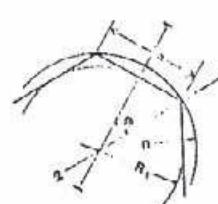
$$I_y = ab^3 \left( \frac{1}{2} - \frac{\pi}{16} - \frac{1}{36} \left( 1 - \frac{\pi}{4} \right)^2 \right)$$

**D-24**  
(Continued).

**PROPERTIES OF SECTIONS**

REGULAR POLYGON

Axis of moments through center



$$n = \text{Number of sides}$$

$$\phi = \frac{180^\circ}{n}$$

$$a = 2\sqrt{R^2 - R_1^2}$$

$$R = \frac{a}{2 \sin \phi}$$

$$R_1 = \frac{a}{2 \tan \phi}$$

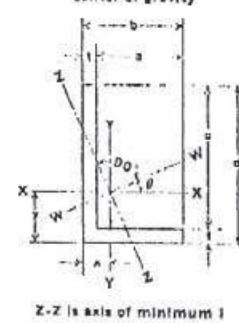
$$A = \frac{1}{4} na^2 \cot \phi = \frac{1}{2} nR^2 \sin 2\phi = nR_1^2 \tan \phi$$

$$I_x - I_y = \frac{A(6R^2 - a^2)}{24} = \frac{A(12R_1^2 + a^2)}{48}$$

$$r_x = r_y = \sqrt{\frac{6R^2 - a^2}{24}} = \sqrt{\frac{12R_1^2 + a^2}{48}}$$

ANGLE

Axis of moments through center of gravity



$$\tan 2\theta = \frac{2K}{I_x - I_y}$$

$$A = t(b+c), \quad x = \frac{b^2 + ct}{2(b+c)}, \quad y = \frac{ct^2 + at}{2(b+c)}$$

$$K = \text{Product of inertia about } X-X \text{ & } Y-Y \\ = \frac{abdt}{4(b+c)}$$

$$I_x = \frac{1}{3} (t(b-x)^2 + dy^2 - a(y-t)^2)$$

$$I_y = \frac{1}{3} (t(b-x)^2 + dx^2 - c(x-t)^2)$$

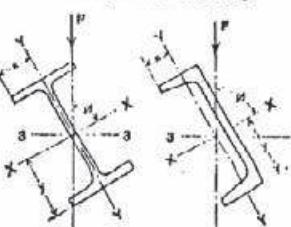
$$I_z = I_x \sin^2 \theta + I_y \cos^2 \theta + K \sin 2\theta$$

$$I_w = I_x \cos^2 \theta + I_y \sin^2 \theta - K \sin 2\theta$$

K is negative when heel of angle, with respect to  $\theta$ , is in 1st or 3rd quadrant, positive when in 2nd or 4th quadrant.

BEAMS AND CHANNELS

Transverse force oblique through center of gravity



$$I_x = I_x \sin^2 \theta + I_y \cos^2 \theta$$

$$I_y = I_x \cos^2 \theta + I_y \sin^2 \theta$$

$$= M \left( \frac{y}{I_x} \sin \theta + \frac{x}{I_y} \cos \theta \right)$$

Where M is bending moment due to force F.  
Extreme fiber assumed same as for case  $\theta=0$ . If not, locate extreme fiber and find  $f$  by usual method.

**D-25**  
**Wind Chill Equivalent Temperatures on  
Exposed Flesh at Varying Velocity**

Temperature, F	WIND VELOCITY (MILES PER HOUR)										
	45	35	25	20	15	10	5	3	2	1	0
90	89.5	89	88.5	88	88.75	87.5	87	86	84.5	83	
82	81	80.5	80	79.5	78	76	74	72.5	70	60	
72	71	69.5	68	67	65	60	57	53.5	47.5	23	
63	61	59	57	55	52	44.5	39	34.5	20	-11	
51	49	47	45	42.5	38	28	18.5	11	0	-27	
41	39	36	34	30.5	25	11	0	-9	-23.5	-38	
30	28	25	23	18	11	-5	-16.5	-40	Below -40	Below -40	
20	18	14	11	6	-2	-19	-40	Below -40	do	do	
10	7.5	3	0	-6	-15	-35	Below -40	do	do	do	
0	-2.5	-8	-12	-18	-29	Below -40	do	do	do	do	
-11	-14	-18	-23	-30	Below -40	do	do	do	do	do	
-21	-24	-30	-35	Below -40	do	do	do	do	do	do	
-32	-35	-40	-40					do	do	do	

InSTRUCTIONS FOR USE OF THE TABLE:

- (1) First obtain the temperature and wind velocity forecast data.
- (2) Locate the number at the top corresponding to the expected wind speed (or the number closest to this).
- (3) Read down this column until the number corresponding to the expected temperature (or the number closest to this) is reached.
- (4) From this point follow across to the right on the same line until the last number is reached under the column marked zero (0) wind speed.
- (5) This is the equivalent temperature reading. Example: weather information gives the expected temperature (at a given time, such as midnight) to be 35°F, and the expected wind speed (at the same time, midnight) to be 20 miles per hour (mph). Locate the 20 mph column at the top, follow down this column to the number nearest to 35°F. The nearest number is 34°F. From this point, move all the way to the right on the same line and find the last number, which is -38°F. This means that with a temperature of 35°F, and a windspeed of 20 mph the rate of cooling of all exposed flesh is the same as -38°F, with no wind.

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do means ditto.

**D-26**  
**Impurities in Water**

*U. S. Systems of Expressing Impurities*

- 1 grain per gallon ..... = 1 grain calcium carbonate ( $\text{CaCO}_3$ ) per U. S. gallon of water
- 1 part per million ..... = 1 part calcium carbonate ( $\text{CaCO}_3$ ) per 1,000,000 parts of water
- 1 part per hundred thousand ..... = 1 part calcium carbonate ( $\text{CaCO}_3$ ) per 100,000 parts of water

*Foreign Systems of Expressing Impurities*

- 1 English degree (or "Clark") ..... = 1 grain calcium carbonate ( $\text{CaCO}_3$ ) per British Imperial gal. of water
- 1 French degree ..... = 1 part calcium carbonate ( $\text{CaCO}_3$ ) per 100,000 parts of water
- 1 German degree ..... = 1 part calcium oxide ( $\text{CaO}$ ) ..... per 100,000 parts of water

*Conversions*

CONVERSION TABLE (Expressed to 3 Significant Figures)		Parts Calcium per Million (ppm)	Parts $\text{CaCO}_3$ per Hundred Thousand (ppm/100,000)	Grains $\text{CaCO}_3$ per U.S. Gallon (gral)	English Degree or "Clark"	French Degree or "French"	German Degree or "German"	MILL-equivalents per Liter or Equivalents per Million
1 Part per Million	1.	.1	.002	.07	.1	.000	.000	.000
1 Part per Hundred Thousand	10.0	1.	.002	.7	1.	.000	.000	.000
1 Part per U. S. Gallon	17.1	1.71	1.	1.2	1.71	.000	.000	.000
1 English or Clark Degree	14.0	1.42	.002	1.	1.42	.000	.000	.000
1 French Degree	10.	1.	.002	.7	1.	.000	.000	.000
1 German Degree	17.0	1.70	1.04	1.24	1.70	1.	.000	.000
1 MIL-equivalent per Liter	50.	5.	5.00	3.50	5.00	5.	5.	5.
1 Equivalent per Million	50.	5.	5.00	3.50	5.00	5.	5.	5.

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## D-27

### Water Analysis Conversions for Units Employed : Equivalents

WATER ANALYSIS UNITS CONVERSION TABLE (Expressed to 3 Significant Figures)	Parts per Million (ppm)	Milligrams per Liter (mgm/l.)	Grams per Liter (grams/l.)	Parts per Hundred Thousand (Pts./100,000)	Grains U.S. Gallon (grs./U.S. gal.)	Grains per British Imp. Gallon	Kilograins per Cubic Foot (Kgr./cu. ft.)
1 Part per Million	1.	1.	.001	.1	.0583	.07	.0004
1 Milligram per Liter	1.	1.	.001	.1	.0583	.07	.0004
1 Gram per Liter	1000.	1000.	1.	100.	58.3	70.	.436
1 Part per Hundred Thousand	10.	10.	.01	1.	5.83	.7	.00436
1 Grain per U.S. Gallon	17.1	17.1	.017	1.71	1.	1.2	.0075
1 Grain per British Imp. Gallon	14.3	14.3	.014	1.43	.833	.8	.0062
1 Kilograins per Cubic Foot	2294.	2294.	2.294	229.4	134.	161.	1.

Note: In practice, water analysis samples are measured by volume, not by weight and corrections for variations in specific gravity are practically never made. Therefore, parts per million are assumed to be the same as milligrams per liter and hence the above relationships are, for practical purposes, true.

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## D-28

### Parts Per Million to Grains Per U.S. Gallon

A. To convert parts per million of hardness to grains per U. S. gallon, divide by the factor 17.1.

Example:

$$\frac{242 \text{ parts/million}}{17.1} = 14.1 \text{ grains/U. S. gallon}$$

B. To convert grains per U. S. gallon to parts per million of hardness, multiply by the factor 17.1.

$$2. 24.3 \text{ grains/U. S. gallon} \times 17.1 = 416 \text{ parts/million}$$

#### Equivalents

Water analyses may also be expressed as:

- (1) Equivalents per million (epm) .....  $= \frac{\text{No. of ppm of substance present}}{\text{Equivalent weight of substance}}$
- (2) Milli equivalents per liter (meq/l.) .....  $= \text{Equivalents per million}$
- (3) Parts per million expressed as  $\text{CaCO}_3$  .....  $= \text{No. of ppm CaCO}_3 \text{ equivalent to No. of ppm of substance present}$
- (4) Fiftieths of equivalents per million (epm/50) .....  $= \frac{\text{No. of ppm of substance present} \times 50}{\text{Equivalent weight of substance}}$

Notes: Numerically (1) and (2) are equal.

Numerically (3) and (4) are equal.

Section xxiii contains equivalent weights of a number of substances.

Section xxiv contains factors for converting various substances to  $\text{CaCO}_3$ .

Section xxv contains factors for various chemical conversions.

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## D-29

**Formulas, Molecular and Equivalent Weights, and Conversion  
Factors to  $\text{CaCO}_3$  of Substances Frequently Appearing  
in the Chemistry of Water Softening**

Substance	Formula	Molecular weight	Equivalent weight	Multiplying Factor Considering molecular wt. of $\text{CaCO}_3$ as 100.	
				Substance to $\text{CaCO}_3$ equivalent	$\text{CaCO}_3$ equivalent ( $\times$ ) substance
Aluminum	$\text{Al}$	27.0	9.0	6.56	0.18
Aluminum Chloride	$\text{AlCl}_3$	133	44.4	1.13	0.39
Aluminum Chloride	$\text{AlCl}_3 \cdot 6\text{H}_2\text{O}$	241	80.5	0.62	1.61
Aluminum Sulfate	$\text{Al}_2(\text{SO}_4)_3 \cdot 18\text{H}_2\text{O}$	666.4	111.1	0.45	2.22
Aluminum Sulfate	$\text{Al}_2(\text{SO}_4)_3$ (anhydrous)	342.1	57.0	0.98	1.14
Aluminum Hydrate	$\text{Al}(\text{OH})_3$	78.0	26.0	1.92	0.52
Alumina	$\text{Al}_2\text{O}_3$	101.9	17.0	2.94	0.34
Sodium Aluminate	$\text{NaAlO}_2 \cdot 2\text{H}_2\text{O}$	163.9	27.3	1.83	0.65
Ammomium Alum	$\text{Al}_2(\text{SO}_4)_3 \cdot (\text{NH}_4)_2\text{SO}_4 \cdot 24\text{H}_2\text{O}$	906.6	161.1	0.33	3.02
Potassium Alum	$\text{Al}_2(\text{SO}_4)_3 \cdot \text{K}_2\text{SO}_4 \cdot 24\text{H}_2\text{O}$	948.8	156.1	0.32	3.12
Anarosine	$\text{NH}_3$	17.0	17.0	2.94	0.34
Ammonium (Low)	$\text{NH}_4^+$	18.0	18.0	2.78	0.36
Ammonium Chloride	$\text{NH}_4\text{Cl}$	53.5	53.5	0.94	1.07
Ammonium Hydroxide	$\text{NH}_4\text{OH}$	25.1	36.1	1.43	0.70
Ammonium Sulfate	$(\text{NH}_4)_2\text{SO}_4$	132	66.1	0.76	1.32
Barium	$\text{Ba}$	137.4	68.7	0.73	1.37
Barium Carbonate	$\text{BaCO}_3$	197.4	98.7	0.51	1.97
Barium Chloride	$\text{BaCl}_2 \cdot 2\text{H}_2\text{O}$	244.3	122.2	0.44	2.44
Barium Hydroxide	$\text{Ba}(\text{OH})_2$	171	85.5	0.59	1.71
Barium Oxide	$\text{BaO}$	153	76.7	0.66	1.63
Barium Sulfate	$\text{BaSO}_4$	233.4	116.7	0.43	2.33
Calcium	$\text{Ca}$	40.1	20.0	2.50	0.40
Calcium Bicarbonate	$\text{Ca}(\text{HCO}_3)_2$	162.1	81.1	0.62	1.62
Calcium Carbonate	$\text{CaCO}_3$	100.08	50.04	1.00	1.00
Calcium Chloride	$\text{CaCl}_2$	111.0	55.5	0.38	1.11
Calcium Hydrate	$\text{Ca}(\text{OH})_2$	74.1	37.1	1.35	0.74
Calcium Hypochlorite	$\text{Ca}(\text{ClO})_2$	143.1	35.8	0.70	1.43
Calcium Oxide	$\text{CaO}$	56.1	28.0	1.79	0.56
Calcium Sulfate	$\text{CaSO}_4$ (anhydrous)	136.1	68.1	0.74	1.36
Calcium Sulfate	$\text{CaSO}_4 \cdot 2\text{H}_2\text{O}$ (crysprum)	177.2	88.6	0.58	1.72
Calcium Nitrate	$\text{Ca}(\text{NO}_3)_2$	164.1	82.1	0.61	1.64
Calcium Phosphate	$\text{Ca}_3(\text{PO}_4)_2$	319.3	51.7	0.97	1.03
Carbon	$\text{C}$	12.0	3.00	16.67	0.06

Chlorine (Ion)	$\text{Cl}^-$	35.5	35.5	1.41	0.71
Copper (Cupric)	$\text{Cu}$	63.6	31.8	1.57	0.64
Copper Sulfate (Cupric)	$\text{CuSO}_4$	160	80.0	0.68	1.60
Copper Sulfate (Cupric)	$\text{CuSO}_4 \cdot 5\text{H}_2\text{O}$	236	125	0.49	2.50
Iron (Ferric)	$\text{Fe}^{3+}$	55.8	27.9	1.79	0.56
Iron (Ferric)	$\text{Fe}^{3+}$	55.8	28.5	2.69	0.37
Ferrous Carbonate	$\text{FeCO}_3$	116	57.5	0.96	1.16
Ferrous Hydroxide	$\text{Fe}(\text{OH})_2$	89.9	44.9	1.11	0.96
Ferrous Oxide	$\text{FeO}$	71.8	35.9	1.39	0.72
Ferrous Sulfate	$\text{FeSO}_4$ (anhydrous)	151.9	76.0	0.66	1.52
Ferrous Sulfate	$\text{FeSO}_4 \cdot 7\text{H}_2\text{O}$	273.0	139.0	0.36	2.76
Ferrous Sulfate	$\text{FeSO}_4$ (anhydrous)	151.9	151.9		oxidation
Ferric Chloride	$\text{FeCl}_3$	162	54.1	0.93	1.06
Ferric Chloride	$\text{FeCl}_3 \cdot 6\text{H}_2\text{O}$	270	90.1	0.56	1.00
Ferric Hydrosulfide	$\text{Fe}(\text{OH})_3$	107	35.5	1.44	0.71
Ferric Oxide	$\text{Fe}_2\text{O}_3$	160	26.6	1.88	0.53
Ferric Sulfate (Ferric)	$\text{Fe}_2(\text{SO}_4)_3$	339.9	68.7	0.76	1.39
Ferrous or Ferric	$\text{Fe}^{2+}$	55.8	55.8		oxidation
Ferrous Sulfate	$\text{FeSO}_4$	151.9	151.9		oxidation
Fluorine	$\text{F}$	19.0	19.0	2.66	0.38
Hydrogen (Ion)	$\text{H}^+$	1.01	1.01	50.0	0.82
Iodine	$\text{I}$	127	127	0.40	2.64
Lead	$\text{Pb}$	207	104	0.48	2.06
Magnesium	$\text{Mg}$	24.3	12.2	4.16	0.24
Magnesium Oxide	$\text{MgO}$	40.3	20.2	2.48	0.40
Magnesium Bicarbonate	$\text{Mg}(\text{HCO}_3)_2$	146.3	73.2	0.68	1.48
Magnesium Carbonate	$\text{MgCO}_3$	84.3	42.2	1.19	0.84
Magnesium Chloride	$\text{MgCl}_2$	95.2	47.6	1.06	0.95
Magnesium Hydrate	$\text{Mg}(\text{OH})_2$	58.3	29.2	1.71	0.58
Magnesium Nitrate	$\text{Mg}(\text{NO}_3)_2$	146.3	74.2	0.67	1.48
Magnesium Phosphate	$\text{Mg}_3(\text{PO}_4)_2$	262.9	42.8	1.14	0.88
Magnesium Sulfate	$\text{MgSO}_4$	120.4	60.2	0.63	1.20
Manganese (Manganese)	$\text{Mn}^{2+}$	54.9	27.5	1.32	0.55
Manganese (Manganic)	$\text{Mn}^{3+}$	54.9	18.0	2.73	0.37
Manganese Chloride	$\text{MnCl}_2$	125.8	62.9	0.58	1.26
Manganese Dioxide	$\text{MnO}_2$	67.9	21.7	2.36	0.43
Manganese Hydrosulfide	$\text{Mn}(\text{OH})_2$	89.0	44.4	1.13	0.89
Manganese Oxide	$\text{MnO}_2$	158	26.3	1.90	0.53
Manganese Oxide	$\text{MnO}_2$	70.9	35.5	1.41	0.71

## D-29

(Continued). Formulas, Molecular and Equivalent Weights, and Conversion Factors to  $\text{CaCO}_3$  of Substances Frequently Appearing in the Chemistry of Water Softening

Substance	Formula	Molecular weight	Equivalent weight	Multiplying Factor Considering molecular wt. of $\text{CaCO}_3$ as 100.	
				Substance to $\text{CaCO}_3$ equivalent	$\text{CaCO}_3$ equivalent to substance
Nitrate (Ion)	$\text{NO}_3^-$	62.0	62.0	0.81	1.24
Nitric Acid	$\text{HNO}_3$	63.0	63.0	0.79	1.26
Nitrogen (Valence 3)	$\text{N}^{III}$	14.0	4.67	18.8	0.093
Nitrogen (Valence 5)	$\text{N}^{V}$	14.0	2.80	17.9	0.056
Oxygen	$\text{O}$	16.0	8.00	6.25	0.16
Phosphorus (Valence 3)	$\text{P}^{III}$	31.0	10.3	4.76	0.21
Phosphorus (Valence 5)	$\text{P}^{V}$	31.0	6.28	8.33	0.12
Potassium	$\text{K}$	39.1	39.1	1.28	0.78
Potassium Carbonate	$\text{K}_2\text{CO}_3$	138.	69.0	0.72	1.38
Potassium Chloride	$\text{KCl}$	74.6	74.6	0.67	1.49
Potassium Hydroxide	$\text{KOH}$	56.1	56.1	0.88	1.12
Silver Chloride	$\text{AgCl}$	143.3	142.3	0.35	2.87
Silver Nitrate	$\text{AgNO}_3$	169.9	165.9	0.29	3.40
Silicon	$\text{Si}$	60.1	30.0	1.67	0.60
Silicon	$\text{Si}$	28.1	7.03	7.14	0.14
Sodium	$\text{Na}$	23.0	23.0	2.12	0.46
Sodium Bicarbonate	$\text{NaHCO}_3$	84.0	84.0	0.60	1.63
Sodium Bisulfate	$\text{NaHSO}_4$	126.			
Sodium Bisulfite	$\text{NaHSO}_3$	104.			
Sodium Carbonate	$\text{Na}_2\text{CO}_3 \cdot 10\text{H}_2\text{O}$	206.	53.0	0.94	1.06
Sodium Carbonate	$\text{Na}_2\text{CO}_3$	106.	143.	0.35	2.86
Sodium Chloride	$\text{NaCl}$	58.5	58.5	0.85	1.17
Sodium Hypochlorite	$\text{NaOCl}$	74.5	37.3	0.67	1.43
Sodium Hydrate	$\text{NaOH}$	48.0	40.0	1.25	0.80
Sodium Nitrate	$\text{NaNO}_3$	85.0	85.0	0.59	1.70
Sodium Nitrite	$\text{NaNO}_2$	69.0	34.5	0.73	1.38
Sodium Oxide	$\text{Na}_2\text{O}$	62.0	31.0	1.61	0.62
Tri-sodium Phosphate	$\text{Na}_3\text{PO}_4 \cdot 12\text{H}_2\text{O}$ (10.7% $\text{P}_2\text{O}_5$ )	380.2	126.7	0.40	2.53
Tri-sodium Phos. (anhydrous)	$\text{Na}_3\text{PO}_4 \cdot 43.2\%$ $\text{P}_2\text{O}_5$ )	164.0	54.7	0.91	1.09
Di-sodium Phosphate	$\text{Na}_2\text{HPO}_4 \cdot 12\text{H}_2\text{O}$ (19.8% $\text{P}_2\text{O}_5$ )	358.2	119.4	0.42	2.39
Di-sodium Phos. (anhydrous)	$\text{Na}_2\text{HPO}_4$ (50% $\text{P}_2\text{O}_5$ )	142.0	47.3	1.06	0.96
Mono-sodium I-phosphate	$\text{NaH}_2\text{PO}_4 \cdot \text{H}_2\text{O}$ (61.4% $\text{P}_2\text{O}_5$ )	138.1	46.0	1.09	0.92
Mono-sod. phos. (anhydrous)	$\text{NaH}_2\text{PO}_4$ (59.1% $\text{P}_2\text{O}_5$ )	120.0	40.0	1.25	0.80
Meta-Phosphate (Liquor)	$\text{NaPO}_3$ (69% $\text{P}_2\text{O}_5$ )	102.0	34.0	1.47	0.68
Sodium Sulfate	$\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$	323.1	161.1	0.31	3.22
Sodium Sulfate	$\text{Na}_2\text{SO}_4$	142.1	71.0	0.70	1.42
Sodium Thiosulfate	$\text{Na}_2\text{S}_2\text{O}_3$	158.1	158.1	0.63	1.59
Sodium Tetraacetate	$\text{NaC}_4\text{H}_7\text{O}_2$	270.2	135.1	0.37	2.71
Sodium Sulfite	$\text{Na}_2\text{SO}_3$	126.1	63.0	0.79	1.27
Sulfur (Valence 2)	$\text{S}^{\prime\prime}$	32.1	16.0	3.13	0.32
Sulfur (Valence 4)	$\text{S}^{\prime\prime\prime}$	32.1	8.00	3.25	0.16
Sulfur (Valence 6)	$\text{S}^{\prime\prime\prime\prime}$	32.1	5.34	9.16	0.11
Sulfur Dioxide	$\text{SO}_2$	64.1	32.0		
Tin	$\text{Sn}$	119.			
Water	$\text{H}_2\text{O}$	18.0	9.00	5.56	0.16
Zinc	$\text{Zn}$	65.4	32.7	1.54	0.66
ACID RADICALS					
Bicarbonate	$\text{HCO}_3^-$	61.0	61.0	0.82	1.22
Carbonate	$\text{CO}_3^-$	60.0	30.0	1.67	.66
Carbo. Dioxide	$\text{CO}_2$	44.0	22.0	2.27	.44
Chloride	$\text{Cl}^-$	35.5	35.5	1.41	.71
Iodide	$\text{I}^-$	126.9	126.9	0.46	2.54
Nitrate	$\text{NO}_3^-$	62.0	62.0	0.81	1.24
Hydrate	$\text{OH}^-$	17.8	17.8	2.94	0.24
Phosphate	$\text{PO}_4^{3-}$	95.0	31.7	1.58	0.63
Phosphorous Oxide	$\text{P}_2\text{O}_5$	142.0	23.7	2.11	6.47
Sulfide	$\text{S}^-$	32.1	16.0	3.11	0.32
Sulfate	$\text{SO}_4^-$	96.1	48.0	1.04	0.56
Sulfur Trioxide	$\text{SO}_3$	96.1	48.0	1.25	0.38
ACIDS					
Hydrogen	$\text{H}$	1.0	1.0	50.00	0.02
Acetic Acid	$\text{CH}_3\text{CO}_2\text{H}$	60.1	30.1	0.83	1.29
Carboxic Acid	$\text{HC}_2\text{O}_4\text{H}$	62.0	31.0	1.61	0.52
Hydrochloric Acid	$\text{HCl}$	36.5	18.25	1.37	0.73
Phosphoric Acid	$\text{H}_3\text{PO}_4$	98.0	32.7	1.53	0.65
Sulfurous Acid	$\text{H}_2\text{SO}_3$	82.1	41.1	1.22	0.82
Sulfuric Acid	$\text{H}_2\text{SO}_4$	98.1	49.0	1.02	0.98
Hydrogen Sulfide	$\text{H}_2\text{S}$	—	—	—	—
Manganese Acid	$\text{H}_2\text{MoO}_4$	104.9	52.5	0.95	1.05

### D-30

**Grains Per U.S. Gallons --  
Pounds Per 1000 Gallons**

- A. To convert grains per U. S. gallons to pounds per 1000 gallons multiply by the factor 0.143.
- B. To convert pounds per 1000 gallons to grains per U. S. gallons multiply by the factor 7.0.

*Example:*

- 1. 4.5 grains/U. S. gallon  $\times$  0.143 = 0.644 lbs./1000 gals.
- 2. 0.5 lbs./1000 gallons  $\times$  7.0 = 3.5 grains/U. S. gal.

### D-31

**Parts Per Million --  
Pounds Per 1000 Gallons**

- A. To convert parts per million to pounds per 1000 gallons divide by the factor 120.
- B. To convert pounds per 1000 gallons to parts per million multiply by the factor 120.

*Example:*

- 1. 39 parts/million  $\div$  120 = 0.325 lbs./1000 gals.
- 2. 0.167 lbs./1000 gals.  $\times$  120 = 20 parts/million

### D-32

**Coagulant, Acid, and Sulfate—1 ppm Equivalents**

1 Ppm Name of Chemical	1 Ppm. Formula of Chemical	ppm Alkalinity Reduction	ppm SO <sub>4</sub> as CaCO <sub>3</sub> Increase	ppm Na <sub>2</sub> SO <sub>4</sub> Increase	ppm CO <sub>2</sub> Increase	ppm Total Solids Increase
Filter Alum	Al <sub>2</sub> (SO <sub>4</sub> ) <sub>3</sub> · 18H <sub>2</sub> O	0.45	0.45	0.61	0.46	0.16
Ammonia Alum	Al <sub>2</sub> (SO <sub>4</sub> ) <sub>3</sub> · (NH <sub>4</sub> ) <sub>2</sub> SO <sub>4</sub> · 24H <sub>2</sub> O	0.33	0.44	0.63	0.39	0.27
Potash Alum	Al <sub>2</sub> (SO <sub>4</sub> ) <sub>3</sub> · K <sub>2</sub> SO <sub>4</sub> · 24H <sub>2</sub> O	0.22	0.43	0.66	0.28	0.20
Copperas (Ferrous sulfate)	FeSO <sub>4</sub> · 7H <sub>2</sub> O	0.36	0.36	0.61	0.31	0.13
Chlorinated Copperas	FeSO <sub>4</sub> · 7H <sub>2</sub> O + (½ Cl <sub>2</sub> )	0.54	0.36	0.51	0.46	0.18
Ferric Sulfate (100% Fe <sub>2</sub> (SO <sub>4</sub> ) <sub>3</sub> )	Fe <sub>2</sub> (SO <sub>4</sub> ) <sub>3</sub>	0.75	0.75	1.07	0.66	0.27
Sulfuric Acid—98%	H <sub>2</sub> SO <sub>4</sub>	1.08	1.08	1.42	0.85	0.36
Sulfuric Acid—93.2% (66° Be)	H <sub>2</sub> SO <sub>4</sub>	0.96	0.95	1.35	0.84	0.34
Sulfuric Acid—71.7% (66° Be)	H <sub>2</sub> SO <sub>4</sub>	0.79	0.79	1.13	0.76	0.28
Salt Caustic—96%	Na <sub>2</sub> SO <sub>4</sub>	—	0.66	0.95	—	1.06

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**D-33**  
**Alkali and Lime - 1 ppm Equivalents**

Name 1 ppm	Formula 1 ppm	Alkalinity A Increase ppm	Free CO <sub>2</sub> Reduction ppm	T.E. as CaCO <sub>3</sub> Increase ppm
Sodium Bicarbonate	NaHCO <sub>3</sub>	8.60	—	—
Soda Ash (50% NaOH = 39.18% Na <sub>2</sub> CO <sub>3</sub> )	Na <sub>2</sub> CO <sub>3</sub>	0.94	4.41	—
Cassite Soda (30% NaOH = 50.00% Na <sub>2</sub> CO <sub>3</sub> )	NaOH	1.22	1.05	—
Chemical Lime (Quicklime—Usually 99% CaO)	CaO	1.41	5.41	1.61
Hydrated Lime (Usually 55% Ca(OH) <sub>2</sub> )	Ca(OH) <sub>2</sub>	1.28	1.11	1.28

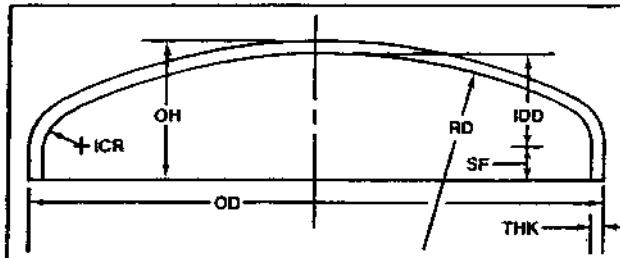
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**D-34**  
**Sulfuric, Hydrochloric Acid Equivalent**

Name		Formula	Specific Gravity 60°/60° F.	Concentration	Grosses/Liter	CaCO <sub>3</sub> Equivalent to one lb. Acid	
Sulfuric Acid	60° Br	H <sub>2</sub> SO <sub>4</sub>	1.7623	77.67%	1325	.7921	5240
Boric Acid	60° Br	H <sub>3</sub> BO <sub>3</sub>	1.7024	52.15%	1710	.3505	8657
Sulfate Acid	30%	H <sub>2</sub> SO <sub>4</sub>	1.6467	36.96%	1804	1.0000	7800
Hydrochloric Acid	10% Br	HCl	1.1617	27.98%	213	.3821	2672

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For "Overall Height" add length of straight flange to IDD given, plus thickness of material.

### ASME FLANGED AND DISHED HEADS IDD CHART

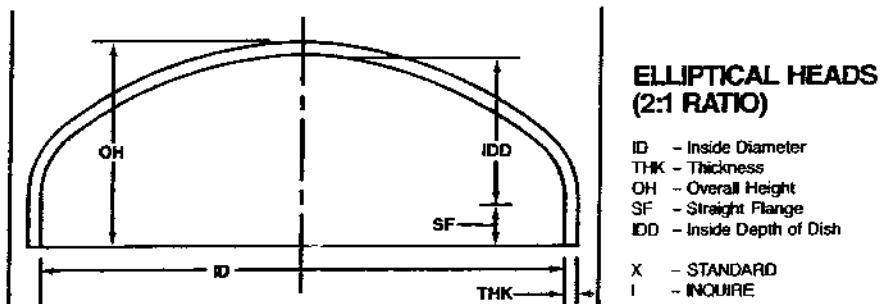
OD - Outside Diameter  
THK - Thickness  
OH - Overall Height  
SF - Straight Flange  
RD - Radius of Dish  
ICR - Inside Corner Radius  
IDD - Inside Depth of Dish

Use when RD EQUALS DIAMETER

OD	THK	1/8"	1/4"	3/8"	1/2"	5/8"	3/4"	7/8"	1"	5/8"	11/16"	13/16"	7/8"	15/16"	17/16"	19/16"	21/16"	23/16"
12	1/8	1.59	1.92	2.00	2.09													
14	1/8	2.79	2.25	2.27	2.39													
16	1	2.63	2.60	2.57	2.62													
18	1/8	2.97	2.94	2.91	2.98													
20	1/8	3.31	3.28	3.25	3.22													
22	1/8	3.73	3.75	3.67	3.64													
24	1/8	4.00	3.97	3.90	3.96													
26	1/8	4.72	4.69	4.66	4.63													
28	1/8	4.99	4.96	4.92	4.89													
30	1/8	5.25	5.22	5.19	5.16													
32	1/8	5.52	5.46	5.45	5.42													
34	1/8	5.78	5.75	5.72	5.69													
36	1/8	6.05	6.02	5.98	5.95													
38	3	6.78	6.74	5.77	6.68	6.85	6.82	6.59	6.58	6.53	6.50	6.44	6.38					
40	3	7.04	7.01	6.98	6.95	6.91	6.88	6.81	6.82	6.79	6.76	6.70	6.64					
42	3	7.31	7.27	7.24	7.21	7.19	7.15	7.12	7.09	7.06	7.03	6.96	6.90					
44	3	7.57	7.54	7.51	7.47	7.44	7.41	7.38	7.35	7.32	7.29	7.22	7.16					
46	3	7.84	7.80	7.77	7.74	7.71	7.68	7.64	7.61	7.58	7.55	7.48	7.43					
48	3	8.10	8.07	8.04	8.00	7.97	7.94	7.91	7.88	7.85	7.81	7.75	7.69					
50	3	8.37	8.34	8.30	8.27	8.24	8.21	8.17	8.14	8.11	8.06	8.02	7.95					
52	3	9.08	9.08	9.03	8.99	8.98	8.93	8.9	8.87	8.84	8.8	8.74	8.68					
54	3	9.35	9.32	9.29	9.26	9.23	9.19	9.16	9.13	9.10	9.07	9.01	8.94					
56	3	9.62	9.59	9.58	9.52	9.49	9.46	9.43	9.40	9.36	9.33	9.27	9.21					
58	3	9.89	9.85	9.82	9.79	9.76	9.72	9.69	9.66	9.63	9.60	9.53	9.47					
60	3	10.15	10.12	10.09	10.05	10.02	9.98	9.95	9.93	9.89	9.85	9.80	9.74					
62	3	10.42	10.39	10.35	10.32	10.29	10.26	10.22	10.19	10.16	10.13	10.06	10.00					
64	4	10.99	10.95	10.92	10.89	10.88	10.83	10.79	10.76	10.73	10.70	10.64	10.57					
66	4	11.25	11.22	11.19	11.16	11.12	11.09	11.06	11.03	10.99	10.96	10.90	10.84					
68	4	11.52	11.49	11.45	11.42	11.39	11.35	11.32	11.29	11.26	11.23	11.17	11.10					
70	4	11.78	11.73	11.70	11.66	11.65	11.62	11.58	11.55	11.53	11.48	11.43	11.37					
72	4	12.26	12.22	12.20	12.16	12.22	12.19	12.16	12.13	12.10	12.06	12.00	11.94					
74	4	12.62	12.59	12.56	12.53	12.49	12.46	12.43	12.39	12.36	12.33	12.27	12.20					
76	4	12.89	12.85	12.82	12.79	12.76	12.72	12.69	12.66	12.63	12.59	12.53	12.47					
78	4	13.16	13.12	13.09	13.05	13.02	12.99	12.96	12.92	12.89	12.86	12.80	12.73					
80	5	13.57	13.54	13.50	13.47	13.44	13.41	13.37	13.34	13.31	13.25	13.21	13.15					
82	5	14.84	14.80	14.77	14.74	14.70	14.67	14.64	14.61	14.58	14.55	14.48	14.41					
84	5	14.56	14.52	14.49	14.46	14.43	14.40	14.38	14.36	14.30	14.27	14.20	14.14					
86	5	14.82	14.79	14.76	14.72	14.69	14.66	14.63	14.60	14.56	14.53	14.47	14.40					
88	5	15.09	15.08	15.02	14.99	14.96	14.92	14.88	14.86	14.83	14.80	14.73	14.67					
90	5	15.35	15.32	15.29	15.26	15.22	15.19	15.16	15.13	15.09	15.06	15.00	14.93					
92	5	15.62	15.69	15.55	15.52	15.49	15.46	15.42	15.39	15.36	15.33	15.26	15.20					
94	5	15.89	15.85	15.82	15.79	15.75	15.72	15.69	15.65	15.62	15.59	15.53	15.46					
96	5	16.61	16.57	16.54	16.51	16.48	16.44	16.41	16.38	16.35	16.32	16.25	16.19					
98	5	16.87	16.84	16.81	16.77	16.74	16.71	16.68	16.64	16.61	16.59	16.52	16.45					
100	5	17.14	17.10	17.07	17.04	17.01	16.97	16.94	16.91	16.88	16.85	16.78	16.72					
102	5	17.40	17.37	17.34	17.31	17.27	17.24	17.21	17.18	17.14	17.11	17.05	16.98					
104	5	17.87	17.84	17.80	17.79	17.54	17.51	17.47	17.44	17.41	17.38	17.31	17.25					
106	5	17.94	17.96	17.97	17.94	17.90	17.77	17.74	17.71	17.67	17.64	17.58	17.51					
108	5	18.20	18.17	18.14	18.10	18.07	18.04	18.00	17.97	17.94	17.91	17.84	17.78					
110	5	18.92	18.20	18.06	18.02	18.09	18.16	18.73	18.63	18.56	18.49	18.37	18.30					
112	5	19.19	19.15	19.12	19.09	19.03	18.99	18.98	18.93	18.89	18.86	18.82	18.77					
114	5	19.45	19.42	19.39	19.36	19.32	19.29	19.28	19.23	19.19	19.16	19.10	18.03					
116	5	19.72	19.69	19.65	19.62	19.59	19.58	19.52	19.49	19.48	19.43	19.36	19.30					
118	5	19.99	19.95	19.92	19.89	19.85	19.82	19.79	19.78	19.75	19.69	19.63	19.58					
120	5	20.25	20.22	20.19	20.15	20.12	20.09	20.05	20.02	19.96	19.96	19.93	19.93					

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**SIZES AND THICKNESSES OF HEADS**

ID	THK																THK
	4 <sup>1</sup> / <sub>2</sub>	5	5 <sup>1</sup> / <sub>2</sub>	6	6 <sup>1</sup> / <sub>2</sub>	7	7 <sup>1</sup> / <sub>2</sub>	8	8 <sup>1</sup> / <sub>2</sub>	9	9 <sup>1</sup> / <sub>2</sub>	10	10 <sup>1</sup> / <sub>2</sub>	11	11 <sup>1</sup> / <sub>2</sub>	12	
6	X	X	X	X	X	X	X	X	X	X	X						6
8																	8
10	X	X	X	X	X	X	X	X	X	X	X						10
12	X	X	X	X	X	X	X	X	X	X	X	I					12
14	X	X	X	X	X	X	X	X	X	X	X	I					14
16	X	X	X	X	X	X	X	X	X	X	X	I					16
18	X	X	X	X	X	X	X	X	X	X	X	I					18
20	X	X	X	X	X	X	X	X	X	X	X	I					20
22	X	X	X	X	X	X	X	X	X	X	X						22
24	X	X	X	X	X	X	X	X	X	X	X						24
26	X	X	X	X	X	X	X	X	X	X	X						26
28	X	X	X	X	X	X	X	X	X	X	X						28
30	X	X	X	X	X	X	X	X	X	X	X						30
32	X	X	X	X	X	X	X	X	X	X	X						32
34	X	X	X	X	X	X	X	X	X	X	X						34
36	X	X	X	X	X	X	X	X	X	X	X						36
38	X	X	X	X	X	X	X	X	X	X	X						38
40	X	X	X	X	X	X	X	X	X	X	X	I	I	I	I	I	40
42	X	X	X	X	X	X	X	X	X	X	X						42
44	X	X	X	X	X	X	X	X	X	X	X						44
46	X	X	X	X	X	X	X	X	X	X	X						46
48	X	X	X	X	X	X	X	X	X	X	X						48
50	X	X	X	X	X	X	X	X	X	X	X	I	I	I	I	I	50
52	X	X	X	X	X	X	X	X	X	X	X						52
54	X	X	X	X	X	X	X	X	X	X	X						54
56	X	X	X	X	X	X	X	X	X	X	X						56
58	X	X	X	X	X	X	X	X	X	X	X						58
60	X	X	X	X	X	X	X	X	X	X	X						60
62	X	X	X	X	X	X	X	X	X	X	X						62
64	X	X	X	X	X	X	X	X	X	X	X						64
66	X	X	X	X	X	X	X	X	X	X	X						66
68	X	X	X	X	X	X	X	X	X	X	X						68
70	X	X	X	X	X	X	X	X	X	X	X						70
72	X	X	X	X	X	X	X	X	X	X	X						72
74	X	X	X	X	X	X	X	X	X	X	X						74
76	X	X	X	X	X	X	X	X	X	X	X						76
78	X	X	X	X	X	X	X	X	X	X	X						78
80	X	X	X	X	X	X	X	X	X	X	X						80
82	X	X	X	X	X	X	X	X	X	X	X						82
84	X	X	X	X	X	X	X	X	X	X	X						84
86	X	X	X	X	X	X	X	X	X	X	X						86
88	X	X	X	X	X	X	X	X	X	X	X						88
90	X	X	X	X	X	X	X	X	X	X	X						90
92	X	X	X	X	X	X	X	X	X	X	X						92
94	X	X	X	X	X	X	X	X	X	X	X						94
96	X	X	X	X	X	X	X	X	X	X	X						96
98	X	X	X	X	X	X	X	X	X	X	X						98
100	X	X	X	X	X	X	X	X	X	X	X						100
102	X	X	X	X	X	X	X	X	X	X	X						102
104	X	X	X	X	X	X	X	X	X	X	X						104
106	X	X	X	X	X	X	X	X	X	X	X						106
108	X	X	X	X	X	X	X	X	X	X	X						108
110	X	X	X	X	X	X	X	X	X	X	X						110
112	X	X	X	X	X	X	X	X	X	X	X						112
114	X	X	X	X	X	X	X	X	X	X	X						114
116	X	X	X	X	X	X	X	X	X	X	X						116
118	X	X	X	X	X	X	X	X	X	X	X						118
120	X	X	X	X	X	X	X	X	X	X	X						120

Elliptical  
(2:1 Ratio)

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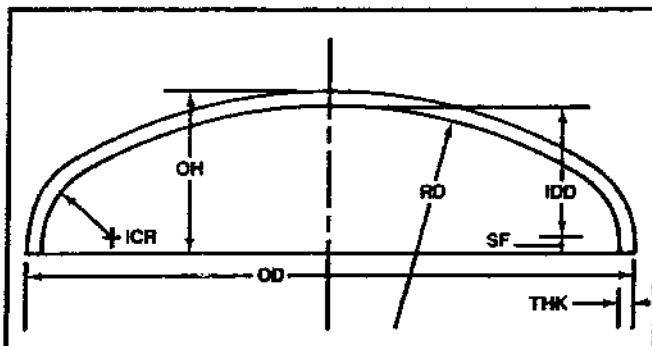


Figure 1.  
**INTERNAL PRESSURE COMPARISON**

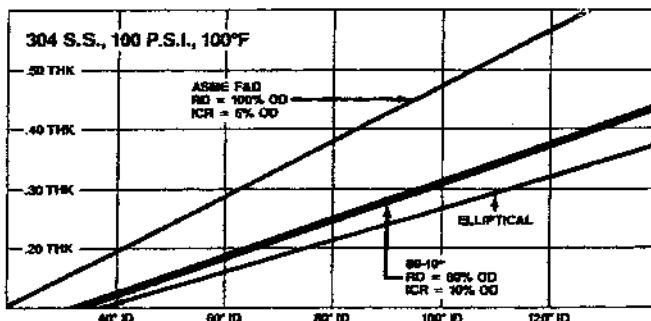
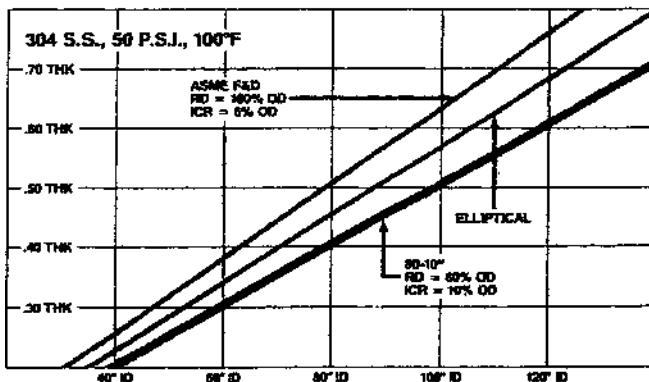


Figure 2.  
**EXTERNAL PRESSURE COMPARISON**



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### 80-10° HEADS

OD — Outside Diameter  
THK — Thickness  
OH — Overall Height  
SF — Straight Flange  
RD — Radius of Dish  
ICR — Inside Corner Radius  
IDD — Inside Depth of Dish

Meeting all A.S.M.E. Unfired Pressure Vessel Code requirements, the 80-10° Head permits significantly higher pressures than other configurations selected for the same service. The 80-10° Head is named for its unique dimensions—the dish radius equals 80% of the head diameter and the inside corner radius equals 10% of the head diameter. These dimensions compare to 100% and 6% respectively for A.S.M.E. F&D Heads.

# Appendix E

---

## E.1 FITTING EQUATIONS TO DATA

Engineers, technologists, and scientists frequently collect paired data in order to understand the behavior of the system. Numerical methods of analysis have been employed to solve a wide range of steady and transient state problems. The fundamentals are essential in the basic operations of curve fitting, approximation, interpolation, numerical solutions of simultaneous linear and non-linear equations, numerical differentiation, and integration. These requirements are greater when new processes are designed.

Various types of software packages are now readily available for scientists and engineers. They must fit a function or functions to measure data that fluctuate, which result from random error of measurement. A Fortran computer program (PROG1) was developed which determines the coefficients that provide the best fit for the following equations:

$$Y = a + bX \quad (\text{E-1})$$

$$Y = a + bX^2 \quad (\text{E-2})$$

$$Y = a + b/X \quad (\text{E-3})$$

$$Y = a + bX^{0.5} \quad (\text{E-4})$$

$$Y = aX^b \quad (\text{E-5})$$

$$Y = ae^{bX} \quad (\text{E-6})$$

$$Y = a + b/X \quad (\text{E-7})$$

$$Y = a + be^X \quad (\text{E-8})$$

The non-linear Eqs. (E-5), (E-6), and (E-8) can be transformed by linearizing as follows:

$$Y = aX^b \quad \ln Y = \ln a + b \ln X \quad (\text{E-9})$$

$$Y = ae^{bX} \quad \ln Y = \ln a + bX \quad (\text{E-10})$$

$$Y = a + be^X \quad \ln Y = \ln a + (\ln b) X \quad (\text{E-11})$$

## E.2 LINEAR REGRESSION ANALYSIS

Regression analysis uses statistical and mathematical methods to analyze experimental data and to fit mathematical models to these data. The least squares provide the best method for objectively determining the best straight line through a series of points. The method assumes that all deviations from the line are the result of error in the measurement of the dependent variable. The method of least squares yields the parameters which minimize the sum of squares of the residuals (e.g., the deviation of each measurement of the dependent variable from its calculated value). If  $\hat{Y}$  is the calculated value and  $Y$  is the original value of the dependent variable, then the residual sum of the squares can be minimized by the following expression.

$$\text{SRS} = \sum_{i=1}^n r_i^2 = \sum_{i=1}^n s_i^2 = \text{minimum} \quad (\text{E-12})$$

where  $n$  = number of observations of  $X-Y$  data.

$$\hat{Y}_i = a + bX_i \quad (\text{E-13})$$

$$r_i = a + bX_i - Y_i \quad (\text{E-14})$$

and

$$\text{SRS} = \sum_{i=1}^n r_i^2 = \sum_{i=1}^n (a + bX_i - Y_i)^2 \quad (\text{E-15})$$

The values of  $a$  and  $b$  are found by minimizing Eq. (E-15). This involves taking the partial derivative of the equation with respect to each variable  $a$  and  $b$ , and setting the result to zero.

$$\frac{\partial \sum r_i^2}{\partial a} = 0 \text{ and } \frac{\partial \sum r_i^2}{\partial b} = 0 \quad (\text{E-16})$$

Substituting Eq. (E-15) into Eq. (E-16) gives

$$\frac{\partial \sum (a + bX_i - Y_i)^2}{\partial a} = 0 \quad (\text{E-17})$$

and

$$\frac{\partial \sum (a + bX_i - Y_i)^2}{\partial b} = 0 \quad (\text{E-18})$$

This is equivalent to

$$\frac{2 \sum (a + bX_i - Y_i) \partial \sum (a + bX_i - Y_i)}{\partial a} = 0 \quad (\text{E-19})$$

and

$$\frac{2X_i \sum (a + bX_i - Y_i) \partial \sum (a + bX_i - Y_i)}{\partial b} = 0 \quad (\text{E-20})$$

Since  $b$ ,  $X$ , and  $Y$  are not functions of  $a$ , and the partial derivative of  $a$  with respect to itself is unity, Eq. (E-19) reduces to

$$\sum a + \sum bX_i = \sum Y_i \quad (\text{E-21})$$

Similarly,  $a$ ,  $X$ , and  $Y$  are not functions of  $b$ . Therefore, Eq. (E-20) becomes

$$\sum aX_i + \sum bX_i^2 = \sum X_i Y_i \quad (\text{E-22})$$

where  $a$  and  $b$  are constants. Equations (E-17) and (E-18) are expressed as:

$$an + b \sum_{i=1}^n X_i = \sum_{i=1}^n Y_i \quad (\text{E-23})$$

and

$$a \sum_{i=1}^n X_i + b \sum_{i=1}^n X_i^2 = \sum_{i=1}^n X_i Y_i \quad (\text{E-24})$$

Equations (E-23) and (E-24) apply only when fitting a straight line to a set of  $X-Y$  data points. Both equations are linear in  $X$ ,  $Y$ , and  $n$  and the unknowns  $a$  and  $b$ . Using Cramer's rule for the simultaneous equations, we have

$$a = \frac{\begin{vmatrix} \sum Y_i & \sum X_i \\ \sum X_i Y_i & \sum X_i^2 \end{vmatrix}}{\begin{vmatrix} n & \sum X_i \\ \sum X_i & \sum X_i^2 \end{vmatrix}} \quad (\text{E-25})$$

and

$$b = \frac{\begin{vmatrix} n & \sum Y_i \\ \sum X_i & \sum X_i Y_i \end{vmatrix}}{\begin{vmatrix} n & \sum X_i \\ \sum X_i & \sum X_i^2 \end{vmatrix}} \quad (\text{E-26})$$

Solving Eqs. (E-25) and (E-26) gives

$$a = \frac{\sum X_i^2 \sum Y_i - \sum X_i \sum X_i Y_i}{n \sum X_i^2 - \sum X_i \sum X_i} \quad (\text{E-27})$$

and

$$b = \frac{n \sum X_i Y_i - \sum X_i \sum Y_i}{n \sum X_i^2 - \sum X_i \sum X_i} \quad (\text{E-28})$$

respectively, where all the sums are taken over all experimental observations. The developed computer program PROG1 computes the constants  $a$  and  $b$  for Eqs. (E-1)–(E-8). The program evaluates the correlation coefficient, which gives a measure of the accuracy of fit. Alternatively, a table with column heading as  $X_i$ ,  $Y_i$ ,  $X_i^2$ , and  $X_i Y_i$  can be constructed using the Microsoft Excel spreadsheet; the constants  $a$  and  $b$  can be evaluated from Eqs. (E-27) and (E-28) respectively. The Excel spreadsheet also employs a *trendline* through the data after being plotted into a worksheet. The following are the steps being employed to fit a straight line to a set of data in the Excel spreadsheet.

1. Open a new worksheet and enter the  $X$ -data (the independent variable) in the leftmost column.
2. Enter the  $Y$ -data (the dependent variable) in the next column.

### EXAMPLE E-1

The first-order rate constant  $k$  for the rotation about the C–N bond in *n*, *n*-dimethylnicotinamide measured at different temperatures by nuclear magnetic resonance are

$T(^{\circ}\text{C})$	10.0	15.7	21.5	27.5	33.2	38.5	45.7
$k(\text{s}^{-1})$	2.08	4.57	8.24	15.8	28.4	46.1	93.5

(Source: B.G. Cox, Modern Liquid Phase Kinetics, Oxford Chemistry Primers, ZENECA, Oxford University Press, 1994.)

Determine the activation energy  $E$  and the pre-exponential factor  $k_o$  for the rotation.

*Solution*

The Arrhenius equation is expressed as:

$$k = k_o e^{-E/RT}$$

which can be linearized in the form of model Eq. (E-1) as

$$\ln k = \ln k_o - \frac{E}{RT}$$

or used in the form of Eq. (E-6) as:

$$Y = ae^{bx} \quad (\text{E-6})$$

3. Plot the data as a line graph (i.e., an  $X$ – $Y$  chart) with arithmetic coordinates. *Do not interconnect the individual data points.*
4. Activate the graph for editing. Then click on one of the plotted data points, thus selecting the data set as the active editing object (the data points will appear highlighted when this step is carried out correctly).
5. Choose *Trendline* from the Insert menu. They specify the type of curve, and request any pertinent options when the *Trendline* dialog box appears. These are, displaying the intercept, equation of the curve and its associated correlation coefficient ( $r^2$ ) squared. Extrapolating the curve fit forward, that is beyond the rightmost data point or backward beyond the leftmost data point, is another option to explore. (Generally the equation of the curve and its associated  $r^2$  value are requested for display on the graph.)
6. Press the OK button. The curve fitting will then be carried out and the results are displayed automatically.

Commercial curve fitting packages are now available in the market and details can be retrieved from the world wide web (www) using the Google search engine by typing *linear regression*. Typical websites are:

- [www.ebicom.net/~dhyams](http://www.ebicom.net/~dhyams)
- [www.analyse-it.com](http://www.analyse-it.com)
- [www.wessa.net](http://www.wessa.net).

Table E-1 shows the computer results of Example E-1.

The Excel spreadsheet Prog1a.xls is used to determine the activation energy  $E$  and the pre-exponential factor  $k_o$  as:

$$E = 78781.8 \text{ J/mol}, \quad k_o = 7.7 \times 10^{14} \text{ s}^{-1}$$

Figure E-1 shows the snap shot of the spreadsheet.

The Fortran computer program PROG1 also determines the activation energy  $E$  and the pre-exponential factor  $k_o$  from the linearized data or alternatively from the non-linear equation  $Y = ae^{bx}$ . The best regression is determined from the various model equations in program PROG1. The regression coefficients from the linearized equation  $Y = A + BX$  are as follows.

$$A = 0.3425E + 02$$

$$B = -0.9468E + 04$$

The activation energy  $E$  is obtained from the slope of  $1/T$  vs  $\ln k$

$$\text{Slope: } \frac{-E}{R} = -0.9468 \times 10^4$$

where

$$R = 8.314 \text{ J/mol K}$$

$$E = 78716.9 \text{ J/mol.}$$

The pre-exponential factor,  $k_o = \exp(A) = \exp(34.25) = 7.49 \times 10^4 \text{ s}^{-1}$ .

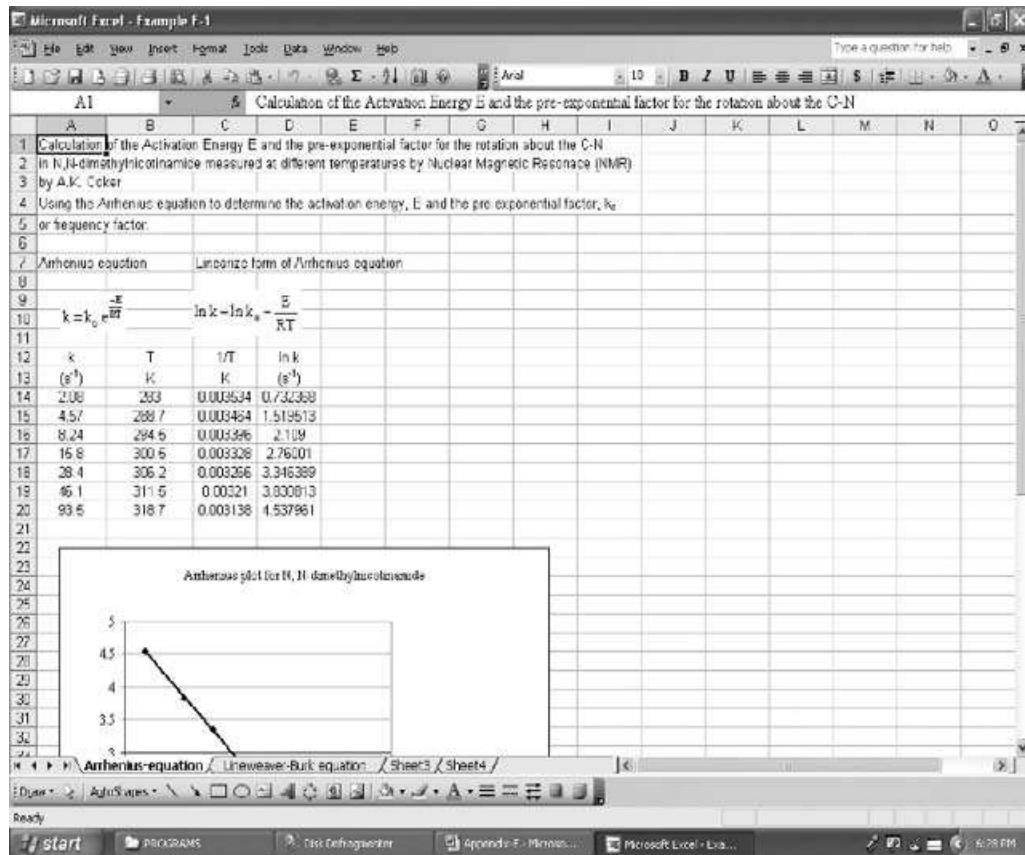


Figure E-1 The Excel spreadsheet snapshot of Example E-1.

Table E-1 Curve Fitting for Two Variables

X	-1/X	Y
283.0	-0.00353	2.08
288.7	-0.00346	4.57
294.5	-0.003396	8.24
300.5	-0.003328	15.8
306.2	-0.003266	26.4
311.6	-0.00321	46.1
318.7	-0.003138	93.5

The computer results give the constants  $A$  and  $B$  from the model Eq. (E-6) as:

$$A = 0.7206E + 15$$

$$B = -0.9457E + 04$$

$$\text{Slope : } \frac{-E}{R} = -0.9457 \times 10^4$$

where

$$R = 8.314 \text{ J/mol K}$$

$$E = 78625 \text{ J/mol.}$$

The pre-exponential factor,  $k_0 = A = 7.206 \times 10^4 \text{ s}^{-1}$ .

The Excel spreadsheet Example E-1.xls shows another example using the Lineweaver-Burk equation to determine the constants  $k_m$  and  $V_{max}$ .

### E.3 POLYNOMIAL REGRESSION

Some engineering data are often poorly represented by a linear regression. Some form of non-linear regression can be developed, if the dependence of  $Y$  on  $X$  is known, although the total convergence of this iterative regression procedure cannot be guaranteed. However, if the form of dependence is unknown, then  $Y$  can be treated as a general function of  $X$  by trigonometric terms (Fourier analysis) or polynomial function. The least squares procedure can be readily extended to fit the data to an  $n$ th-degree polynomial:

$$Y = C_0 + C_1 X + C_2 X^2 + \cdots + C_n X^n \quad (\text{E-29})$$

where  $C_0, C_1, C_2, \dots, C_n$  are constants.

For this case, the sum of the squares of the residuals is minimized.

$$S = \sum_{j=1}^N [Y_j - C_0 - C_1 X_j - C_2 X_j^2 - \cdots - C_n X_j^n]^2 \quad (\text{E-30})$$

At the minimum, all the partial derivatives with respect to the chosen constants are zero, that is

$$\frac{\partial S}{\partial C_0}, \frac{\partial S}{\partial C_1}, \dots, \frac{\partial S}{\partial C_n} = 0 \quad (\text{E-31})$$

This gives a system of  $(n+1)$  linear equations in  $(n+1)$  unknowns,  $C_0, C_1, C_2, \dots, C_n$

$$\begin{aligned}
\frac{\partial S}{\partial C_0} = 0 &= \sum_{j=1}^N 2(Y_j - C_0 - C_1 X_j \\
&\quad - C_2 X_j^2 - \cdots - C_n X_j^n)(-1) \\
\frac{\partial S}{\partial C_1} = 0 &= \sum_{j=1}^N 2(Y_j - C_0 - C_1 X_j \\
&\quad - C_2 X_j^2 - \cdots - C_n X_j^n)(-X_j) \\
\frac{\partial S}{\partial C_2} = 0 &= \sum_{j=1}^N 2(Y_j - C_0 - C_1 X_j \\
&\quad - C_2 X_j^2 - \cdots - C_n X_j^n)(-X_j^2) \\
&\vdots \\
\frac{\partial S}{\partial C_n} = 0 &= \sum_{j=1}^N 2(Y_j - C_0 - C_1 X_j \\
&\quad - C_2 X_j^2 - \cdots - C_n X_j^n)(-X_j^n)
\end{aligned} \tag{E-32}$$

The above equations are set to equal zero, and can be rearranged in the following set of normal equations.

$$\begin{aligned}
C_0 N + C_1 \sum X_j + C_2 \sum X_j^2 + \cdots + C_n \sum X_j^n &= \sum Y_j \\
C_0 \sum X_j + C_1 \sum X_j^2 + C_2 \sum X_j^3 + \cdots + C_n \sum X_j^{n+1} \\
&= \sum X_j Y_j \\
C_0 \sum X_j^2 + C_1 \sum X_j^3 + C_2 \sum X_j^4 + \cdots + C_n \sum X_j^{n+2} \\
&= \sum X_j^2 Y_j \\
&\vdots \\
C_0 \sum X_j^n + C_1 \sum X_j^{n+1} + C_2 \sum X_j^{n+2} + \cdots + C_n \sum X_j^{2n} \\
&= \sum X_j^n Y_j
\end{aligned} \tag{E-33}$$

Equation (E-33) in matrix form becomes

$$UC = V$$

### EXAMPLE E-2

The following data are obtained from  $y(x) = x^4 + 3x^3 + 2x^2 + x + 5$ . Show that a fourth degree polynomial provides the best least squares approximation to the given data. Determine this polynomial.

Fortran program PROG2 calculates (i) the coefficients for each degree of the polynomial, (ii) the variance, (iii) error sum of squares, (iv) total sum of squares, (v) coefficient of determination, and (vi) the correlation coefficient. Alternatively, the Microsoft Excel spreadsheet can be employed using the *Trendline* from the Insert menu. The program shows that the fourth degree gives the lowest value of variance and therefore shows the best fit. The

$$U = \begin{bmatrix} N & \sum X_j & \sum X_j^2 & \cdots & \sum X_j^n \\ \sum X_j & \sum X_j^2 & \sum X_j^3 & \cdots & \sum X_j^{n+1} \\ \sum X_j^2 & \sum X_j^3 & \sum X_j^4 & \cdots & \sum X_j^{n+2} \\ \vdots & \vdots & \vdots & \ddots & \vdots \\ \sum X_j^n & \sum X_j^{n+1} & \sum X_j^{n+2} & \cdots & \sum X_j^{2n} \end{bmatrix}$$

$$C = \begin{bmatrix} C_0 \\ C_1 \\ C_2 \\ \vdots \\ C_n \end{bmatrix} \quad V = \begin{bmatrix} \sum Y_j \\ \sum X_j Y_j \\ \sum X_j^2 Y_j \\ \vdots \\ \sum X_j^n Y_j \end{bmatrix}$$

Linear equations generated by polynomial regression can be ill-conditioned when the coefficients have very small and very large numbers. This results in smooth curves that fit poorly. Ill-conditioning often happens if the degree of polynomial is large and if the  $Y$  values cover a wide range. The error of the polynomial regression by a standard error of the estimate is shown below:

$$\sigma^2 = \frac{\sum SSE}{N - n - 1} \tag{E-34}$$

where

$n$  = the degree of polynomial  
 $N$  = the number of data points.

The coefficient of determination can be expressed as:

$$\begin{aligned}
r^2 &= 1 - \frac{SSE}{SST} \\
&= 1 - \frac{\sum_{j=1}^N (Y_j - \hat{Y}_j)^2}{\sum_{j=1}^N (Y_j - \bar{Y})^2}
\end{aligned} \tag{E-35}$$

The correlation coefficient is given by

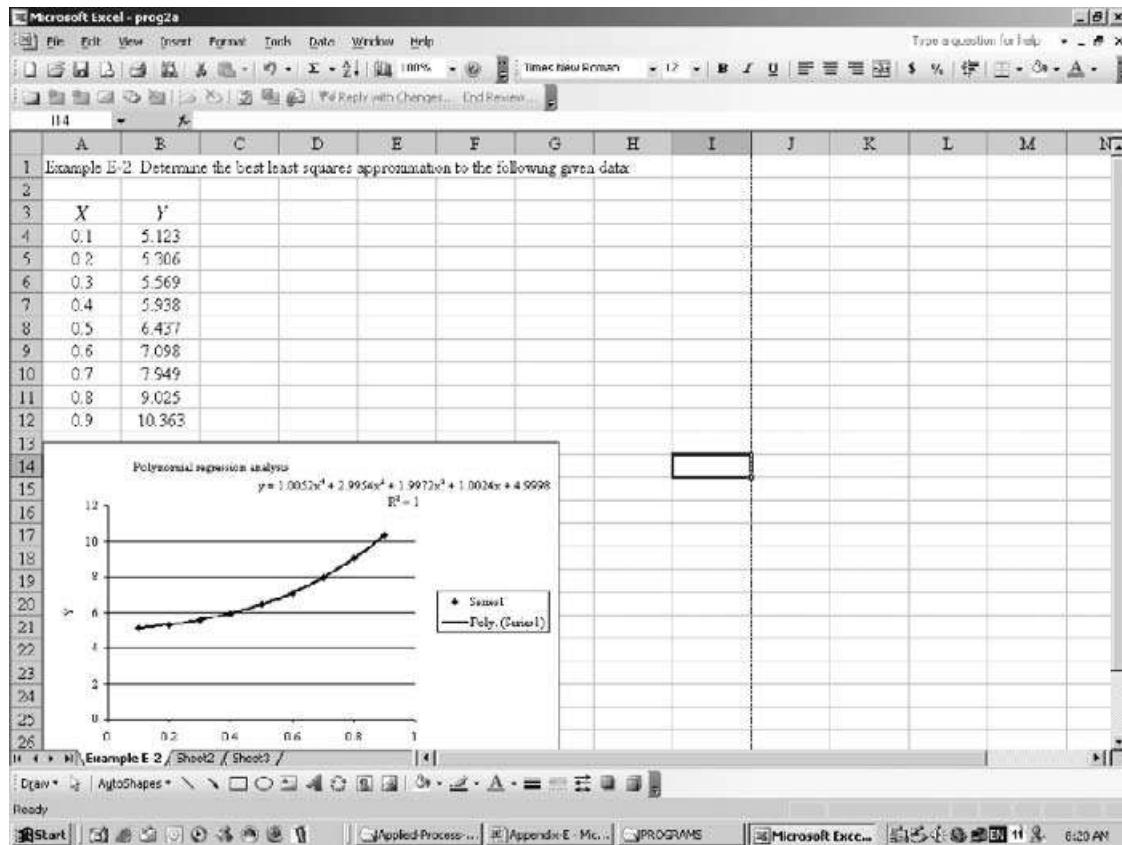
$$r = \left(1 - \frac{SSE}{SST}\right)^{0.5} \tag{E-36}$$

The numerator of Eq. (E-34) should continually decrease as the degree of the polynomial is raised. Alternatively, the denominator of Eq. (E-34) causes  $\sigma^2$  to increase once there is departure from the optimum degree.

computer results are shown in Table E-2. The Excel spreadsheet Prog2a.xls was used to determine the polynomial regression of Example E-2. Figure E-2 shows the Excel spreadsheet snap shot of Example E-2.

**Table E-2 Polynomial Regression Analysis for an Equation to an Nth Degree**

X	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
Y	5.123	5.306	5.569	5.938	6.437	7.098	7.949	9.025	10.363



**Figure E-2** The Excel spreadsheet snapshot of Example E-2.

#### E.4 MULTIPLE REGRESSION ANALYSIS

Inadequate results are sometimes obtained with a single independent variable. This shows that one independent variable does not provide enough information to predict the corresponding value of the dependent variable. We can approach this problem, if we use additional independent variables, and develop a multiple regression analysis to achieve a meaningful relationship. Here, we can employ a linear regression model in cases where the dependent variable is affected by two or more controlled variables.

The linear multiple regression equation is expressed as:

$$Y = C_0 + C_1 X_1 + C_2 X_2 + \cdots + C_K X_K \quad (\text{E-37})$$

where

- $Y$  = the dependent variable
- $X_1, X_2, \dots, X_K$  = the independent variable
- $C_0, C_1, C_2, \dots, C_K$  = the unknown regression coefficients
- $K$  = the number of independent variables.

The unknown coefficients are estimated based on  $n$  observation for the dependent variable  $Y$ , and for each of the independent variables  $X_i$ 's where  $i = 1, 2, 3, \dots, K$ .

These observations are of the form:

$$Y_j = C_0 + C_1 X_{1j} + C_2 X_{2j} + \cdots + C_K C_{Kj} + \xi_j \quad (\text{E-38})$$

For  $j = 1, 2, \dots, N$

where

- $Y_j$  = the  $j$ th observation of the dependent variable
- $X_{1j}, \dots, X_{Kj}$  = the  $j$ th observation of the  $X_1, X_2, \dots, X_K$  independent variables.

We can use a least squares technique to calculate estimates of  $\hat{C}_0, \hat{C}_1, \dots, \hat{C}_K$  by minimizing the following equation:

$$S = \sum_{j=1}^N \left[ Y_j - \left( \hat{C}_0 + \hat{C}_1 X_{1j} + \cdots + \hat{C}_K X_{Kj} \right) \right]^2 = \sum_{j=1}^N \xi_j^2 \quad (\text{E-39})$$

Taking the partial derivatives of  $S$  with respect to  $\hat{C}_0, \hat{C}_1, \dots, \hat{C}_K$ , that is  $\frac{\partial S}{\partial \hat{C}_0}, \frac{\partial S}{\partial \hat{C}_1}, \dots, \frac{\partial S}{\partial \hat{C}_K}$  and setting them equal to zero, we obtain the following set of equations:

$$\begin{aligned} N \hat{C}_0 + (\sum X_{1j}) \hat{C}_1 + \cdots + (\sum X_{Kj}) \hat{C}_K &= \sum Y_j \\ (\sum X_{1j}) \hat{C}_0 + (\sum X_{1j}^2) \hat{C}_1 + \cdots + (\sum X_{1j} X_{Kj}) \hat{C}_K &= \sum X_{1j} Y_j \\ (\sum X_{2j}) \hat{C}_0 + (\sum X_{1j} X_{2j}) \hat{C}_1 + \cdots + (\sum X_{2j} X_{Kj}) \hat{C}_K &= \sum X_{2j} Y_j \\ &\vdots \\ (\sum X_{Kj}) \hat{C}_0 + (\sum X_{1j} X_{Kj}) \hat{C}_1 + \cdots + (\sum X_{Kj}^2) \hat{C}_K &= \sum X_{Kj} Y_j \end{aligned} \quad (\text{E-40})$$

Equation (E-40) can be expressed in matrix form as:

$$U\hat{C} = V$$

where

$$U = \begin{bmatrix} N & \sum X_{1j} & \cdots & \sum X_{Kj} \\ \sum X_{1j} & \sum X_{1j}^2 & \cdots & \sum X_{1j}X_{Kj} \\ \vdots & & & \\ \sum X_{Kj} & \sum X_{1j}X_{Kj} & \cdots & \sum X_{Kj}^2 \end{bmatrix} \quad (\text{E-41})$$

$$\hat{C} = \begin{bmatrix} \hat{C}_0 \\ \hat{C}_1 \\ \vdots \\ \hat{C}_K \end{bmatrix} \quad V = \begin{bmatrix} \sum Y_j \\ \sum X_{1j}Y_j \\ \vdots \\ \sum X_{Kj}Y_j \end{bmatrix} \quad (\text{E-42})$$

$U$  is a symmetric matrix.

We can obtain estimates for the coefficients  $\hat{C}_0, \hat{C}_1, \dots, \hat{C}_K$  by successive elimination or by solving for the inverse of  $U$ . That is,

$$\hat{C} = U^{-1}V \quad (\text{E-43})$$

where

$U^{-1}$  = the inverse of  $U$ .

After solving for  $\hat{C}_0, \hat{C}_1, \dots, \hat{C}_K$ , the estimates of the dependent variable observations  $\hat{Y}_j$  can be obtained as follows:

$$\hat{Y}_j = \hat{C}_0 + \hat{C}_1 X_{1j} + \cdots + \hat{C}_K X_{Kj} \quad (\text{E-44})$$

The power equations have often been derived to calculate the parameters of experimental data. Such an equation can be expressed in the form:

$$Y = C_0 \cdot X_1^{C_1} \cdot X_2^{C_2} \cdots \cdot X_K^{C_K} \quad (\text{E-45})$$

We can calculate the coefficients of the independent variables, if Eq. (E-45) is linearized by taking its natural logarithm to give

$$\ln Y = \ln C_0 + C_1 \ln X_1 + C_2 \ln X_2 + \cdots + C_K \ln X_K \quad (\text{E-46})$$

### EXAMPLE E-3

In a fluid flow experiment, the volumetric rate of fluid through a pipe is dependent on the pipe diameter and slope by the equation

$$Q = C_0 D^{C_1} S^{C_2} \quad (\text{E-50})$$

where

$Q$  = flow rate,  $\text{ft}^3/\text{s}$

$D$  = pipe diameter, ft

$S$  = slope,  $\text{ft}/\text{ft}$ .

Determine the flow rate of fluid for a pipe with a diameter of 3.25 ft and slope of 0.025 ft/ft with the following data:

**TABLE E-3 Analysis of Variance Table for Linear Multiple Regression**

Source of Variance	Degree of Freedom	Sum of Squares	Mean Squares
Total	$N - 1$	$SST = \sum (Y_j - \bar{Y})^2$	$MST = SST/(N - 1)$
Regression	$K$	$SSR = \sum (\hat{Y}_j - \bar{Y})^2$	$MSR = SSR/K$
Error	$N - K - 1$	$SSE = \sum (Y_j - \hat{Y}_j)^2$	$MSE = SSE/(N - K - 1)$

The coefficients  $C_0, C_1, C_2, \dots, C_K$  can then be obtained by Gaussian elimination. Table E-3 shows the variance table for linear multiple regression. The coefficient of determination is

$$r^2 = 1 - \frac{SSE}{SST} \quad (\text{E-47})$$

and the correlation coefficient is

$$r = \left(1 - \frac{SSE}{SST}\right)^{0.5} \quad (\text{E-48})$$

The test statistic is the  $F$ -ratio, which is defined by

$$F = \frac{MSR}{MSE} \quad (\text{E-49})$$

A computer program PROG3 has been developed to determine the coefficients and correlation coefficient of a multiple regression equation.

### E.5 SIMULTANEOUS EQUATIONS USING THE MATRIX METHODS

Matrix solution of sets of linear equations are solved by Excel as follows.

First, write a set of simultaneous equations in matrix format

$$Ax = b \quad (\text{E-51})$$

where  $A$  is the coefficient matrix,  $x$  is the vector of unknowns, and  $b$  is the result vector. To solve this matrix equation, multiply both sides from the left by the inverse of  $A$ :

$$A^{-1}Ax = A^{-1}b \quad (\text{E-52})$$

**Table E-4 Linear Multiple Regression Analysis for an Equation**

Diameter, ft <i>D</i>	Slope, ft/ft <i>S</i>	Flow rate, $\text{ft}^3/\text{s}$ <i>Q</i>
1.0	0.001	1.5
2.5	0.005	9.0
3.0	0.010	25.0
4.0	0.010	5.0
1.5	0.050	30.0
3.5	0.050	100.0

Computer program PROG3 is developed to calculate the values of the coefficients  $C_0, C_1$ , and  $C_2$  and the correlation coefficient. Table E-4 shows the results of the program.

By definition, a matrix times its inverse gives the identity matrix, and the identity matrix times any vector gives the vector back. The equation above reduces to the following solution:

$$x = A^{-1} b \quad (\text{E-53})$$

The following example uses the Fortran program PROG4 and Excel spreadsheet Prog4a.xls.

#### EXAMPLE E-4

The final product from a chemical factory is made by blending four liquids ( $\alpha, \beta, \gamma, \delta$ ) together. Each of these liquids contains four components  $A, B, C$ , and  $D$ . The product leaving the factory has to have a closely specified composition. Determine the relative quantities of  $\alpha, \beta, \gamma$ , and  $\delta$  required to meet the blend specifications in the following data:

Computer program PROG4 uses the Gaussian elimination method to determine the quantities of each  $\alpha, \beta, \gamma$ , and  $\delta$  required to meet the blending specifications. Table E-5 shows the computer results.

**TABLE E-5**

Component	$\alpha$	$\beta$	$\gamma$	$\delta$	composition of specification (W/W)
A	51.30	43.20	56.40	47.40	48.80
B	11.30	11.50	15.50	8.50	11.56
C	29.40	31.50	22.50	30.40	29.43
D	8.00	10.30	5.60	13.70	10.21

(Source: B.Sc. Final year 1978, Aston University, Birmingham, UK.)

Answers:

$$\begin{aligned}\alpha &= 0.1172 \\ \beta &= 0.3789 \\ \gamma &= 0.2117 \\ \delta &= 0.3054\end{aligned}$$

Alternatively, the Excel spreadsheet Prog4a.xls is used to solve the above example problem. The procedure is as follows.

1. Start with a new worksheet, and name it Matrix Method.
  2. In cell A1, enter solving sets of equations; Matrix inversion.
  3. In cell B3, enter  $Ax = b$ .
- Enter the coefficient matrix  $A$  and the results vector  $b$ .
4. In cell A5, enter input matrix  $A$ .
  5. In cells A6:D9, enter the constants of matrix  $A$ :

Cell	Constant	Cell	Constant	Cell	Constant	Cell	Constant
A6	51.3	B6	43.2	C6	56.4	D6	47.4
A7	11.3	B7	11.5	C7	15.5	D7	8.5
A8	29.4	B8	31.5	C8	22.5	D8	30.4
A9	8	B9	10.3	C9	5.6	D9	13.7

6. In cell F5, enter result vector  $b$
7. In cells F6:F9, enter the constants of the result vector.

8. In cell A12, enter Inverse matrix (1/A)

9. Select cells A13:D16, then type

= MINVERSE (A6:D9)

and press Ctrl + Shift + Enter to insert the formula into the whole section.

10. In cell F12, enter solution vector  $x = (1/A)b$ .

11. Select cells F13:F16, then type

+ MMULT (A13:D16, F6:F9)

and press Ctrl + Shift + Enter to insert the formula into the whole selection.

12. Turn off the gridlines, and outline the worksheet cells as shown in Matrix Method.

The worksheet should now look like Matrix Method, with the solution values 0.11722, 0.378939, 0.211747, and 0.305356 for  $\alpha, \beta, \gamma$ , and  $\delta$  in cells F13:F16. This is shown in Figure E-3.

#### E.6 SOLVING SIMULTANEOUS NON-LINEAR EQUATIONS IN EXCEL USING SOLVER

Solver can be used to solve a system of simultaneous equations, which can be either linear or non-linear. Suppose the equations are represented as

$$f_1(x_1, x_2, \dots, x_n) = 0 \quad (\text{E-54})$$

$$f_2(x_1, x_2, \dots, x_n) = 0 \quad (\text{E-55})$$

$$f_3(x_1, x_2, \dots, x_n) = 0 \quad (\text{E-56})$$

⋮

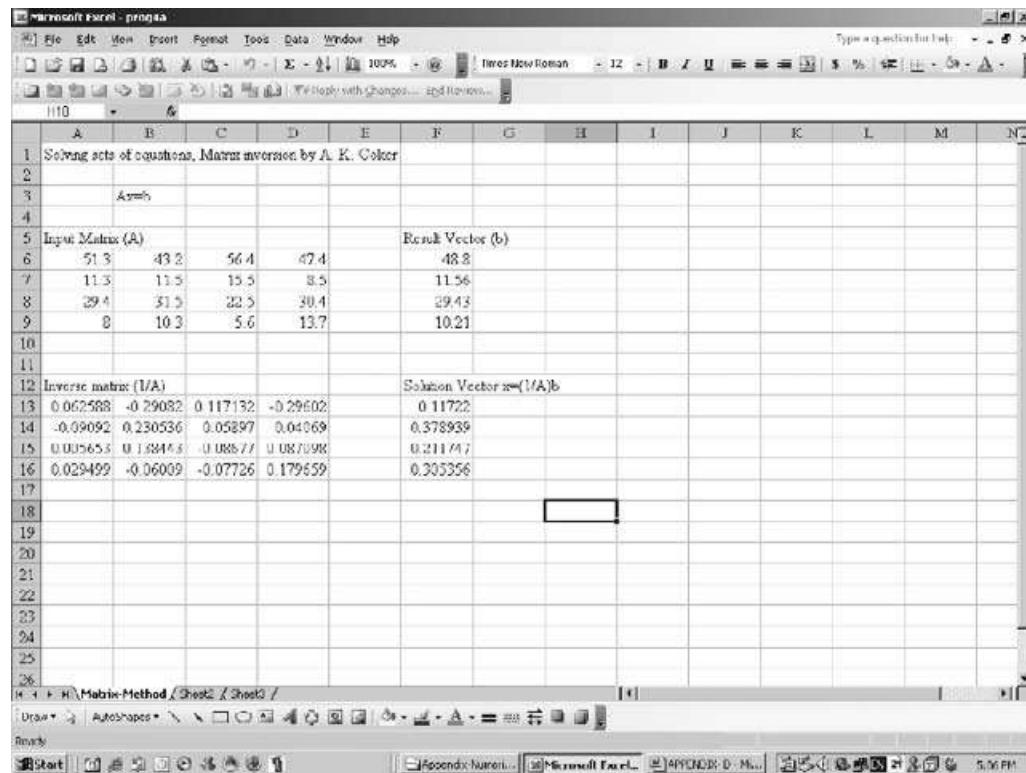
$$f_n(x_1, x_2, \dots, x_n) = 0 \quad (\text{E-57})$$

Thus, we have a system of  $n$  equations in  $n$  unknowns. We can find the values of  $x_1, x_2, x_3, \dots, x_n$  that cause each of the equations to equal zero, if the function

$$y = f_1^2 + f_2^2 + f_3^2 + \dots + f_n^2 \quad (\text{E-58})$$

is equal to zero; that is, to find the values of  $x_1, x_2, x_3, \dots, x_n$  that cause Eq. (E-58) to equal zero. Since all the terms on the right side of Eq. (E-58) are squares, they will all be greater than or equal to zero. Hence, the only way that  $y$  can be equal to zero is that each of the individual  $f$ 's should be equal to zero. Therefore, the values of  $x_1, x_2, x_3, \dots, x_n$  that cause  $y$  to equal zero will be the solutions to the given system of equations. The general approach used with Solver is to define a target function consisting of the squares of the individual equations, as indicated by Eq. (E-58), and to then determine the values of  $x_1, x_2, x_3, \dots, x_n$  that cause the target function to equal zero. To solve a system of simultaneous equations with Solver, proceed as follows.

1. Enter an initial guess for each independent variable ( $x_1, x_2, x_3, \dots, x_n$ ) in a separate cell on the worksheet.
2. Enter the equations for  $f_1, f_2, f_3, \dots, f_n$  and  $y$  in separate cells, expressed as Excel formulas. Within these formulas, express the unknown quantities  $x_1, x_2, x_3, \dots, x_n$  as the addresses of the cells containing the initial guess.
3. Select Solver from the Tools menu.
4. When the Solver Parameters dialog box appears, enter the following information.



**Figure E-3** The Excel spreadsheet snapshot of Example E-4.

- (a) The address of the cell containing the formula for  $y$  in the Set Target Cell location.
- (b) Select value of  $y$  in the Equal to line. Then enter 0 within the associated data area (i.e., determine the values of  $x_1, x_2, x_3, \dots, x_n$  that will drive the target function to zero).
- (c) Enter the range of cell addresses containing the initial values of  $x_1, x_2, x_3, \dots, x_n$  in the area labeled By Changing Cells.
- (d) If you wish to restrict the range of the independent variables, click on the Add button under the heading Subject to the Constraints. Then provide the following information within the Add Constraint dialog box for each of the independent variables.
  - (i) The cell address containing the initial value of the independent variable in the Cell Reference location.
  - (ii) The type of constraint (i.e.,  $\leq$  or  $\geq$ ) from the pull-down menu.
  - (iii) The limiting value in the constraint data area.
  - (iv) Select OK to return to the Solver Parameters dialog box or select Add to add another constraint.
- Note that you can always change a constraint or delete a constraint after it has been added.
- (e) When all the required information has been entered correctly, select Solve. This will initiate the actual solution procedure.

A new dialog box labeled *Solver Results* will then appear, telling you whether or not *Solver* has been able to solve the problem. If a solution has been obtained, the desired values of the independent variables will appear in the cells that originally contained the

initial values. The cell containing the target function will show a value that is zero or nearly zero.

## E.7 GAUSS-SEIDEL ITERATIVE METHOD

Alternatively, Fortran computer program PROG5 employs the Gauss-Seidel method for solving the four material balance equations. The four material balance Eqs. (E-65)–(E-68) are rearranged to solve for the unknown on the diagonal position of each Eq. (E-74).

$$(1000)(1) = 1000C_{A1} + (0.1)(C_{A1})(1000) \quad (E-65)$$

$$1000C_{A1} + 100C_{A3} = 1100C_{A2} + (0.2)(C_{A2})(1500) \quad (E-66)$$

$$1100C_{A2} + 100C_{A4} = 1200C_{A3} + (0.4)(C_{A3})(100) \quad (E-67)$$

$$1000C_{A3} = 1100C_{A4} + (0.3)(C_{A4})(500) \quad (E-68)$$

$$\begin{aligned} 1100C_{A1} &= 1000 \\ 1000C_{A1} - 1400C_{A2} + 100C_{A3} &= 0 \\ 1100C_{A2} - 1240C_{A3} + 100C_{A4} &= 0 \\ 1100C_{A3} - 1250C_{A4} &= 0 \end{aligned} \quad (E-74)$$

Table E-7 shows the computer results of PROG5.

**EXAMPLE E-5**

A chemical reaction takes place in a series of four continuous flow stirred tank reactors arranged as shown. The chemical reaction is a first order irreversible reaction of the type



The conditions of temperature in each reactor are such that the values of  $k_i$  and  $V_i$  are given in Table E-6. Figure E-4 shows four continuous stirred tanks with recycle streams.

**Table E-6**

Reactor	Volume, $V_i L$	Rate constant, $k_i h^{-1}$
1	1000	0.1
2	1500	0.2
3	100	0.4
4	500	0.3

(Source: A. Constantinides, *Applied Numerical Methods With Personal Computers*, McGraw-Hill Book Co., 1987)

The following assumptions are:

1. The system is at steady state.
2. The reactions are in the liquid phase.
3. There is no change in volume or density of the liquid.
4. The rate of disappearance of component A in each reactor is given by

$$(-r_A) = kC_A \quad (\text{E-59})$$

Set up the material balance equations for each of the four reactors, and use the Gauss-Seidel or *Solver* to determine the exit concentration from each reactor.

*Solution*

Set up the material balance equations for each of the four reactions, and use the Gaus-Seidel method to determine the exit concentrations from each reactor.

The general unsteady state material balance for each reactor is

$$\begin{aligned} \text{Input by flow} &= \text{Output by flow} + \text{disappearance} \\ &\quad + \text{accumulation by reaction} \end{aligned}$$

Mass balance for reactor 1:

$$u_{A0} C_{A0} = u_{A0} C_{A1} + (-r_A) V_1 + V_1 \frac{dC_{A1}}{dt} \quad (\text{E-60})$$

Because the system is at steady state, the accumulation is zero, the above equation becomes

$$u_{A0} C_{A0} = u_{A0} C_{A1} + k_1 C_{A1} V_1 \quad (\text{E-61})$$

Mass balance for reactor 2:

$$u_{A0} C_{A1} + u_{A3} C_{A3} = (u_{A0} + u_{A3}) C_{A2} + k_2 C_{A2} V_2 \quad (\text{E-62})$$

Mass balance for reactor 3:

$$(u_{A0} + u_{A3}) C_{A2} + u_{A4} C_{A4} = (u_{A0} + u_{A3} + u_{A4}) C_{A3} + k_3 V_3 C_{A3} \quad (\text{E-63})$$

Mass balance for reactor 4:

$$(u_{A0} + u_{A4}) C_{A3} = (u_{A0} + u_{A4}) C_{A4} + k_4 V_4 C_{A4} \quad (\text{E-64})$$

where

$$u_{A0} = 1000 \text{ L/h}, C_{A0} = 1 \text{ mol/L}$$

$$u_{A3} = 100 \text{ L/h}$$

$$u_{A4} = 100 \text{ L/h}.$$

$$(1000)(1) = 1000 C_{A1} + (0.1)(C_{A1})(1000) \quad (\text{E-65})$$

$$1000 C_{A1} + 100 C_{A3} = 1100 C_{A2} + (0.2)(C_{A2})(1500) \quad (\text{E-66})$$

$$1100 C_{A2} + 100 C_{A4} = 1200 C_{A3} + (0.4)(C_{A3})(100) \quad (\text{E-67})$$

$$1000 C_{A3} = 1100 C_{A4} + (0.3)(C_{A4})(500) \quad (\text{E-68})$$

Rearranging the above equations:

$$f(C_{A1}, C_{A2}, C_{A3}, C_{A4}) = 1100 C_{A1} - 1000 \quad (\text{E-69})$$

$$g(C_{A1}, C_{A2}, C_{A3}, C_{A4}) = 1000 C_{A1} - 1400 C_{A2} + 100 C_{A3} \quad (\text{E-70})$$

$$h(C_{A1}, C_{A2}, C_{A3}, C_{A4}) = 1100 C_{A2} - 1240 C_{A3} + 100 C_{A4} \quad (\text{E-71})$$

$$j(C_{A1}, C_{A2}, C_{A3}, C_{A4}) = 1100 C_{A3} - 1250 C_{A4} \quad (\text{E-72})$$

Thus, we have a system of  $n$  equations in four unknowns. We can find the values of  $C_{A1}$ ,  $C_{A2}$ ,  $C_{A3}$ ,  $C_{A4}$  that cause each of the equations to equal zero, if the function

$$y = f^2 + g^2 + h^2 + j^2 \quad (\text{E-73})$$

is equal to zero; that is, to find the values of  $C_{A1}$ ,  $C_{A2}$ ,  $C_{A3}$ ,  $C_{A4}$  that cause Eq. (E-73) to equal zero. Since all the terms on the right side of Eq.(E-73) are squares, they will all be greater than or equal to zero. Hence, the only way that  $y$  can be equal to zero is that each of the individual  $f$ ,  $g$ ,  $h$ ,  $j$  should be equal to zero. Therefore, the values of  $C_{A1}$ ,  $C_{A2}$ ,  $C_{A3}$ ,  $C_{A4}$  that cause  $y$  to equal zero will be the solutions to the given system of equations. The general approach used with *Solver* is to define a target function consisting of the squares of the individual equations, as indicated by Eq. (E-73), and to then determine the values of  $C_{A1}$ ,  $C_{A2}$ ,  $C_{A3}$ ,  $C_{A4}$  that cause the target function to equal zero. Figures E-5–E-8 of Excel snapshots show the procedures of determining the final concentrations, after initial guesses.

$$C_{A1} = C_{A2} = C_{A3} = C_{A4} = 0.5$$

**TABLE E-7 SOLUTION OF SIMULTANEOUS LINEAR EQUATIONS BY GAUSS-SEIDEL METHOD**


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NUMBER OF LINEAR EQUATIONS: 4

MAXIMUM ITERATIONS 15

CONVERGING TOLERANCE: 0.100000E-03

THE COEFFICIENT MATRIX A(1,1)...A(N+1,N+1) IS:

---

1100.0000	0.0000	0.0000	0.0000	1000.0000
1100.0000	-1400.0000	100.0000	0.0000	0.0000
0.0000	1100.0000	-1240.0000	100.0000	0.0000
0.0000	0.0000	1100.0000	-1250.0000	0.0000

---

THE STARTING VECTOR X(1)....X(N) IS:

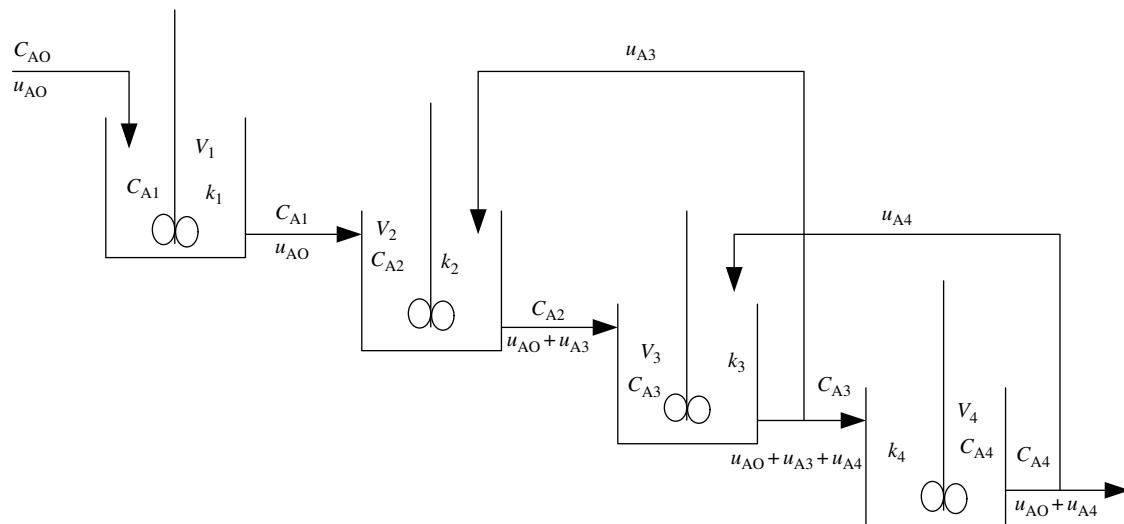
5.000000E-01  
 5.000000E-01  
 5.000000E-01  
 5.000000E-01

PROCEDURE CONVERGED AFTER 5 ITERATIONS

SOLUTION VECTOR X(1).....X(N) IS:

9.090909E-01  
 6.968780E-01  
 6.654189E-01  
 5.855687E-01

---

**Figure E-4** Chemical reaction with recycles in four continuous stirred tanks.

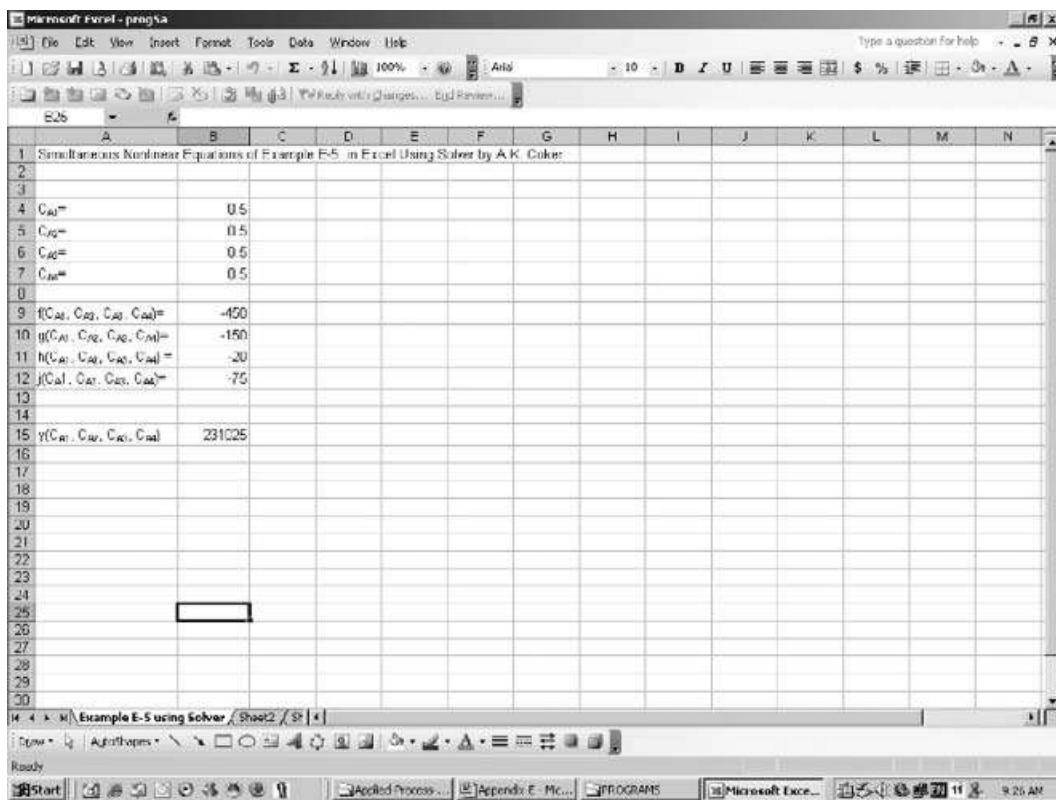


Figure E-5 The Excel spreadsheet snapshot of Example E-5.

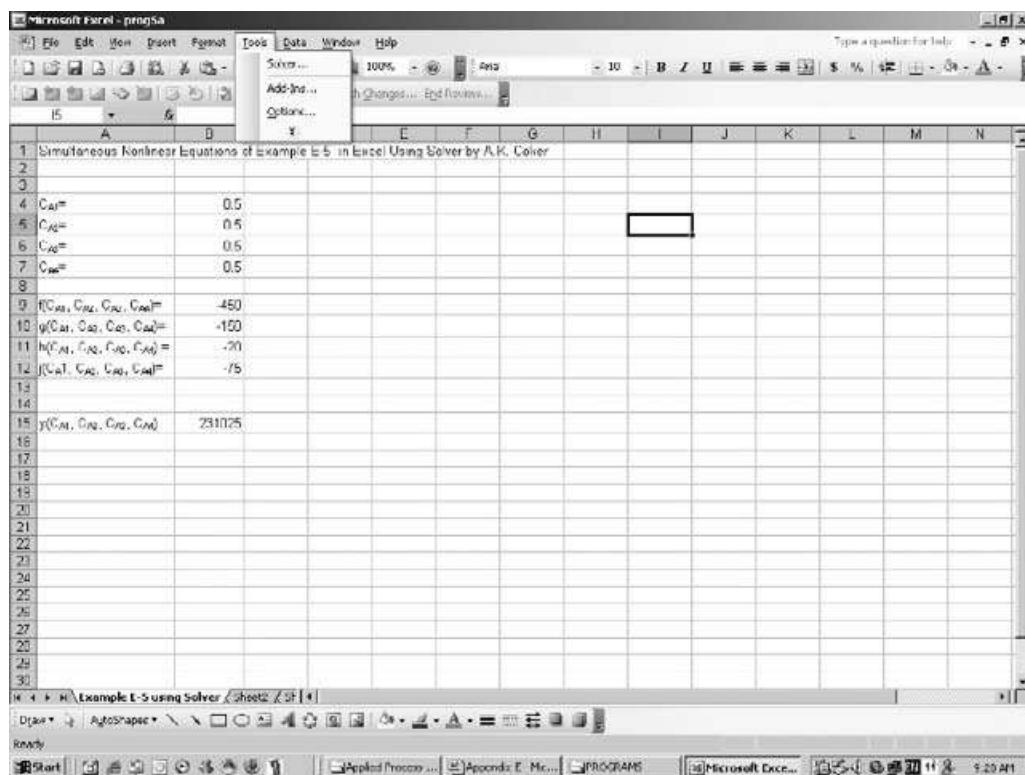


Figure E-6 The Excel spreadsheet snapshot of Example E-5 (continued).

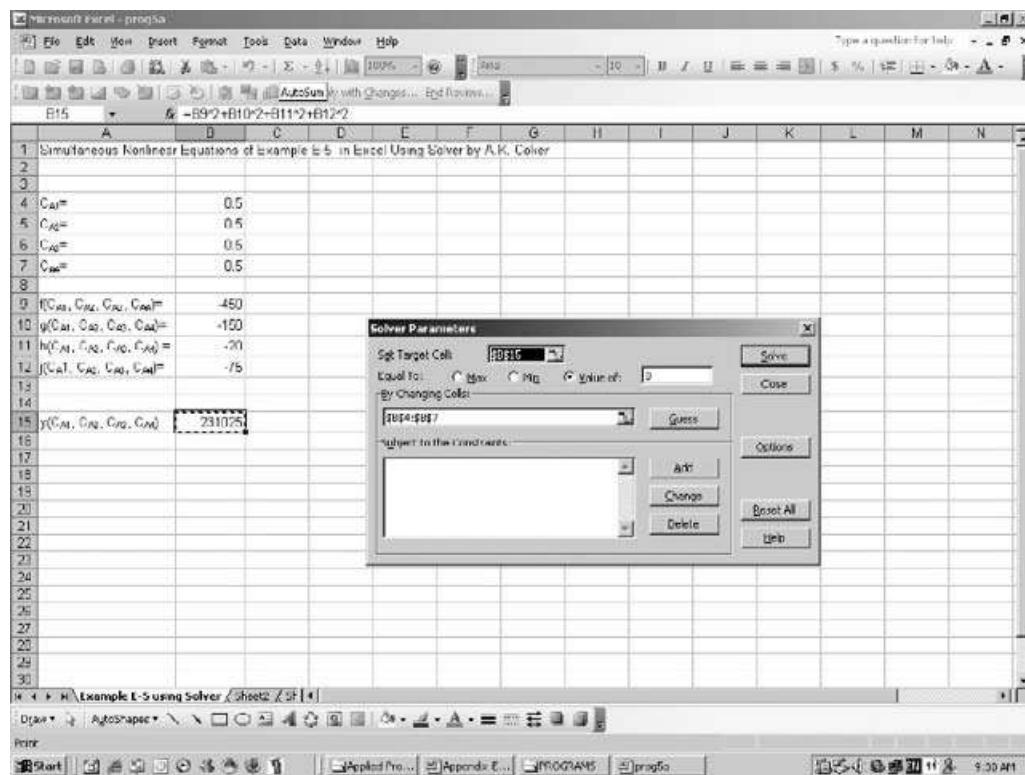


Figure E-7 The Excel spreadsheet snapshot of Example E-5 (continued).

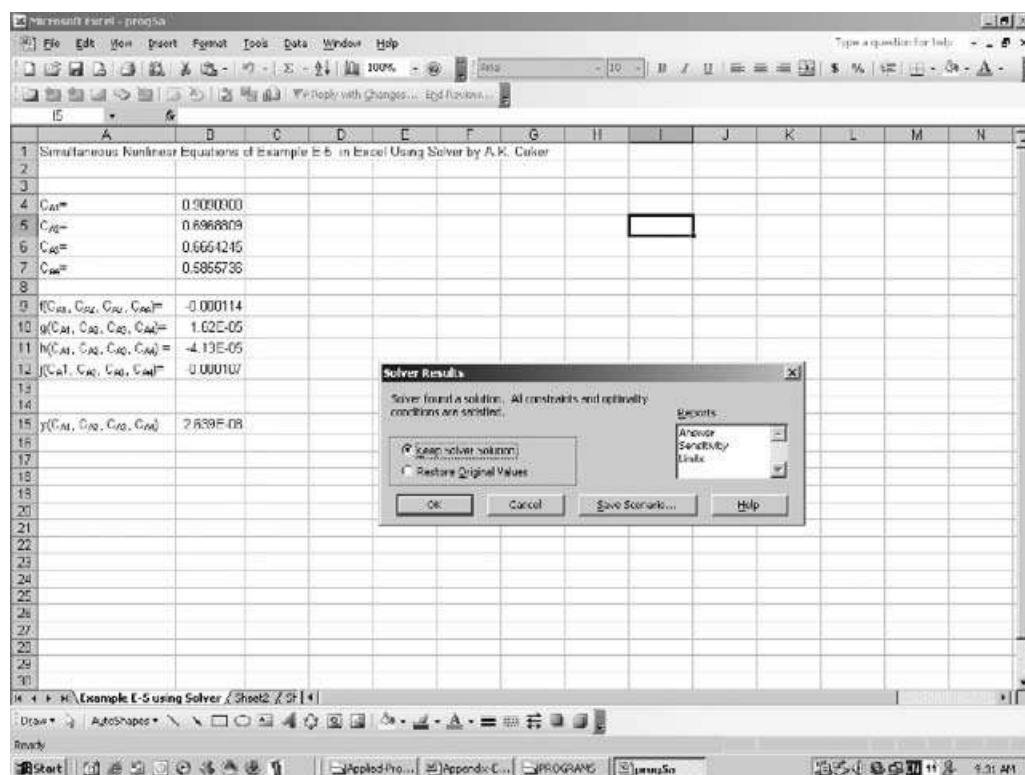


Figure E-8 The Excel spreadsheet snapshot of Example E-5 (continued).

**NOMENCLATURE**

F = F-ratio  
MSE = Error mean squares  
MSR = Regression mean squares  
MST = Total mean squares  
 $r^2$  = Coefficient of determination  
SSE = Error sum of squares  
SSR = Regression sum of squares  
SST = Total sum of squares

**FURTHER READING**

1. Coker, A.K., *Fortran Programs For Chemical Process Design, Analysis and Simulation*, Gulf Publishing Co., Houston, Tx, 1995.
2. Gottfield, B.S., *Spreadsheet Tools for Engineers: Excel 5.0 Version*, The McGraw-Hill Co. Inc., 1996.
3. Billo, E.J., *Excel for Chemists – A Comprehensive Guide*, Wiley-VCH, Inc., 1997.
4. Gottfried, B.S., *Spreadsheet Tools for Engineers: Excel 2000 Version*, McGraw-Hill's Best – Basic Engineering Series and Tools, 2000.

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# Appendix F

## F.1 MICROSOFT EXCEL SOLVER FOR NON-LINEAR EQUATIONS

The *Solver* is an optimization package that finds a maximum, minimum, or specified value of a target cell by varying the values in one or several changing cells. It uses an iterative process, beginning with trial values of the coefficients. The value of each coefficient is changed by a suitable increment, the new value of the function is then calculated, and the change in the value of the function is used to calculate improved values for each of the coefficients. The process is repeated until the desired results are obtained. The *Solver* employs the gradient methods or the simplex method to find the optimum set of coefficients.

Constraints can be applied to the *Solver* using the Add button under the heading “Subject to the Constraints”. The *Solver* is an Excel Add-in, a separate software package. To save memory, it may not automatically be opened whenever Excel is started. To install the *Solver*, choose Add-Ins from the Tools menu. Then select *Solver Add-Ins* from the resulting Add-Ins dialog box. Once the *Solver* feature has been installed, it will remain installed unless it is removed by reversing the above procedure.

To use the *Solver* to perform multiple non-linear least squares curve fitting, the procedure is as follows:

1. Start with a worksheet containing the data (independent variable  $X$  and the dependent variable  $Y_{\text{obsd}}$ ) to be fitted.
2. Add a column containing  $Y_{\text{calc}}$  values, calculated by means of an appropriate formula, and involving the  $X$ -values and one or more coefficients to be varied.
3. Add a column to calculate the square of the residual ( $(Y_{\text{obsd}} - Y_{\text{calc}})^2$ ) for each data point.
4. Calculate the sum of squares of the residuals.
5. Use the *Solver* to minimize the sum of the squares of residuals (the target cell) by changing the coefficients of the function (the changing cells).

Since the *Solver* operates by a search routine, it will find a solution most rapidly and efficiently if the initial estimates that are provided are close to the final values. Conversely, it may not be able to find a solution if the initial estimates are far from the final values. To ensure that the *Solver* has found a global minimum rather than a local minimum, a solution is obtained using different sets of initial estimates.

The least-squares regression coefficients that are returned may be slightly different depending on the starting values being provided. The Excel spreadsheet prog6a.xls (Figures F-1a-f) uses the *Solver* to determine the outlet pressure  $P_2$  of isothermal compressible fluid flow of Example 4-9 in the Chapter 4 of the book.

## F.2 SOLVING EQUATIONS USING GOAL SEEK IN EXCEL

Microsoft Excel provides a built-in way to perform non-linear equations using the Goal Seek command in the Tools menu. Goal-Seek varies the value of a selected cell (the changing cell) to make the value of another cell (the target cell) reach a desired value. This feature permits rapid solutions of algebraic equations using iterative (i.e. trial and error) root-finding techniques based

on a series of successive refinements derived from an initial guess. For an adiabatic compressible fluid using the Excel spreadsheet (Example 4-11.xls) in Chapter 4 of the book, the implicit adiabatic equation in cell C53 is set to zero (Figures F-2a-f). The equation is expressed by

$$\frac{2}{Ma_1^2(\gamma+1)} \left[ \left( \frac{\{\gamma+1\} Ma_1^2}{2 + \{\gamma-1\} Ma_1^2} \right)^{\frac{(\gamma+1)}{2}} - 1 \right] + 2 \ln \sqrt{\frac{2 + (\gamma-1) Ma_1^2}{(\gamma+1) Ma_1^2}} + K = 0 \quad (4-173)$$

The objective is to use Goal Seek to find the value of cell B44 that makes the function (in C53) equal to zero. The accuracy of the result depends on the magnitude of the Maximum Change parameter, which can be adjusted by choosing the Calculation tab in the Options command of the Tools menu. The default value is 0.001, and adjusting the Maximum Change parameter is critical when using Goal Seek. This allows Excel to stop iterating when the change in the result is less than the Maximum Change parameter. Therefore, the Maximum Change parameter needs to be adjusted to match the value of the function. For most calculations, the Maximum Change is set to 1E-12 or 1E-15.

The steps in using Goal Seek are as follows:

1. Select Goal Seek from the Tools menu as shown in Figure F-2a.
2. Enter \$C\$53 in the *Set cell* box.
3. Put the cursor in the *To value* box and enter the desired value, zero.
4. Put the cursor in the *By Changing cell* box and enter \$B\$44 by selecting the cell or by typing.
5. Then click on *OK* box.

After a few iteration cycles the *Status* dialog box Figure (F-2e) is displayed.

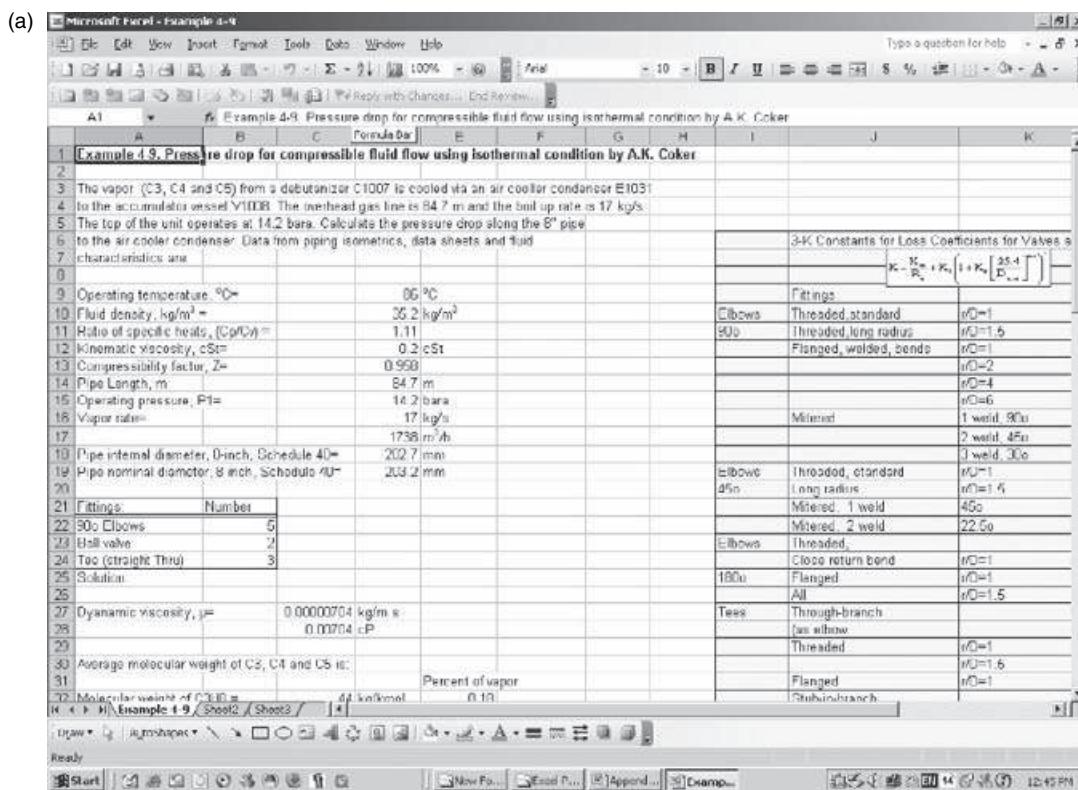
If a solution is obtained, the value of the root will appear in the cell originally containing the initial guess. The value in the cell containing the formula will show a value that is close to zero (or zero). This last value will appear within the Goal Seek status dialog box.

Cell \$B\$44 gives the root of the function cell (B44), which causes the target value to equal zero. The final value of the function depends to some extent on the starting value of  $Ma_1$ . The likelihood of obtaining a converged solution will be enhanced, if the initial guess is as close as possible to the desired root.

For problems requiring the variation of two or more parameters, that is, varying the values of several cells to make the value of another cell reach a desired value involves the use of the *Solver*.

The Excel spreadsheet Prog6a.xls uses the Goal Seek routine to determine the friction factor of Colebrook–White implicit equation as defined by

$$F(f_D) = \frac{1}{f_D^{1/2}} + 0.8686 \ln \left\{ \frac{\varepsilon}{3.7D} + \frac{2.51}{Re \sqrt{f_D}} \right\} \quad (F-1)$$



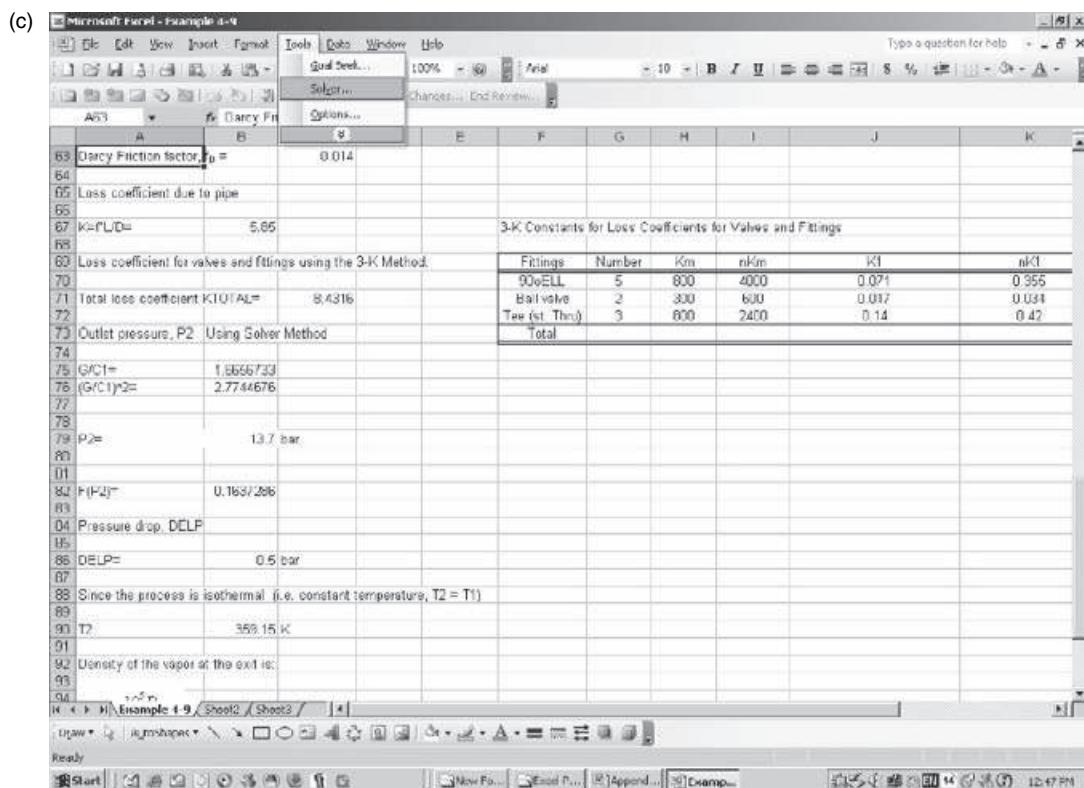


Figure F-1c—(continued).

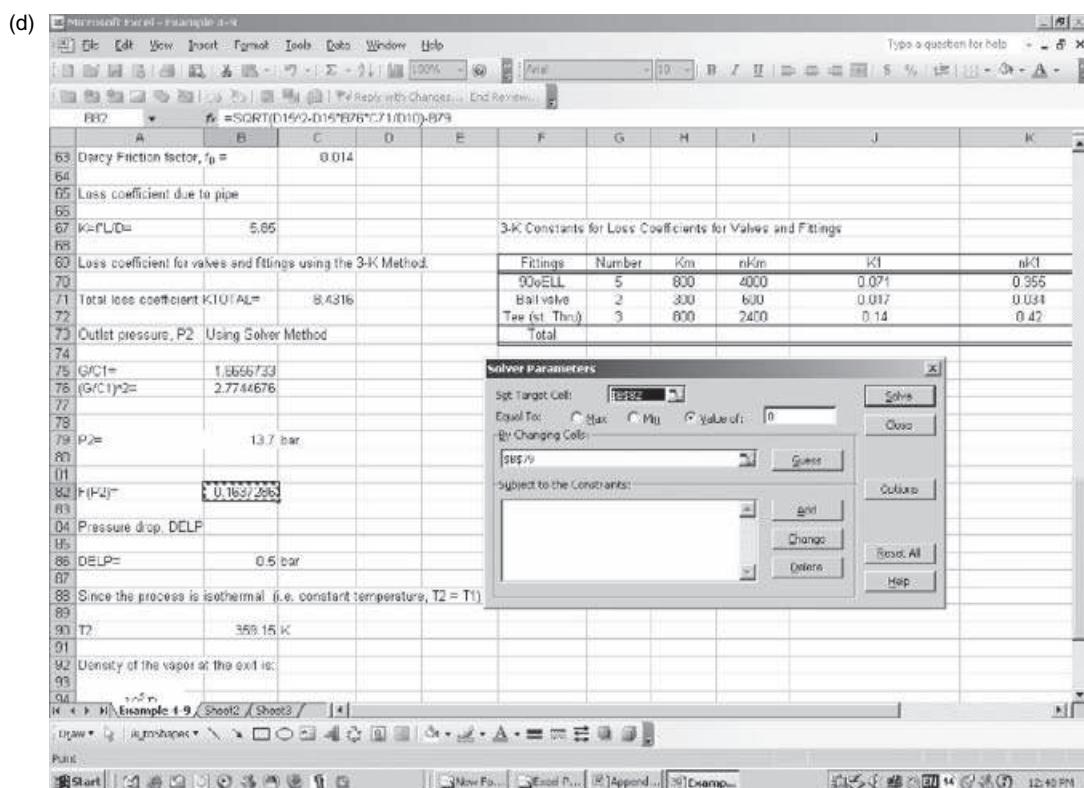


Figure F-1d—(continued).

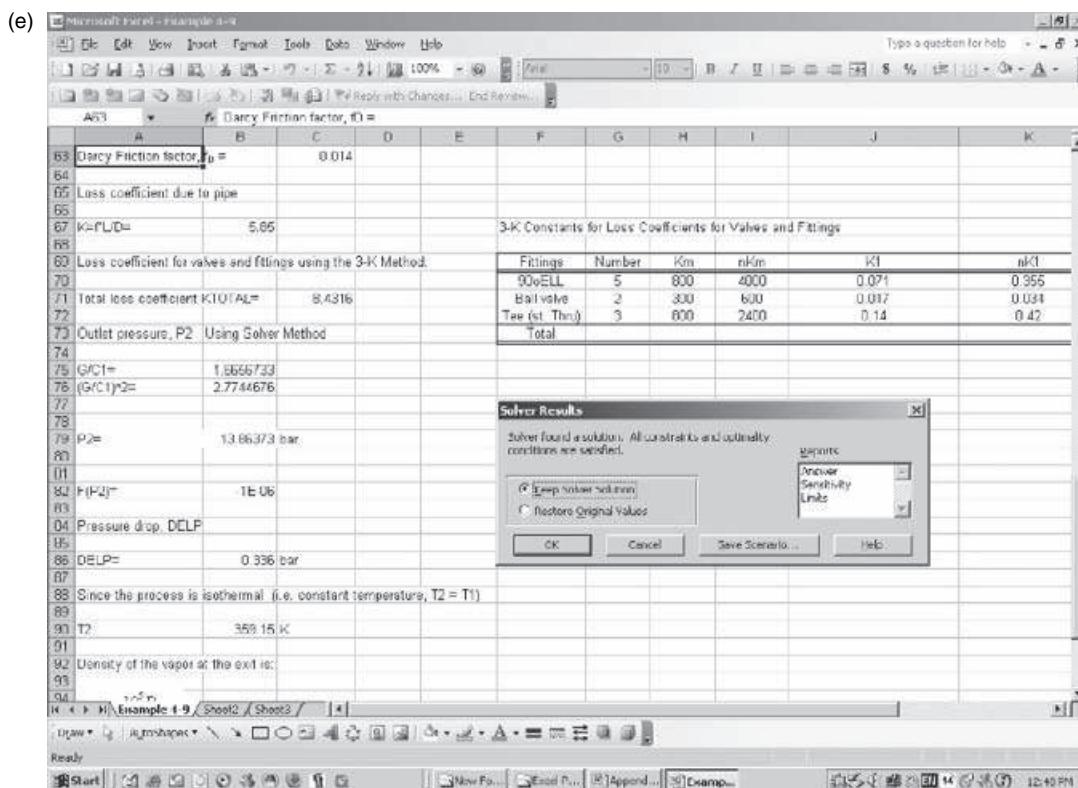


Figure F-1e—(continued).

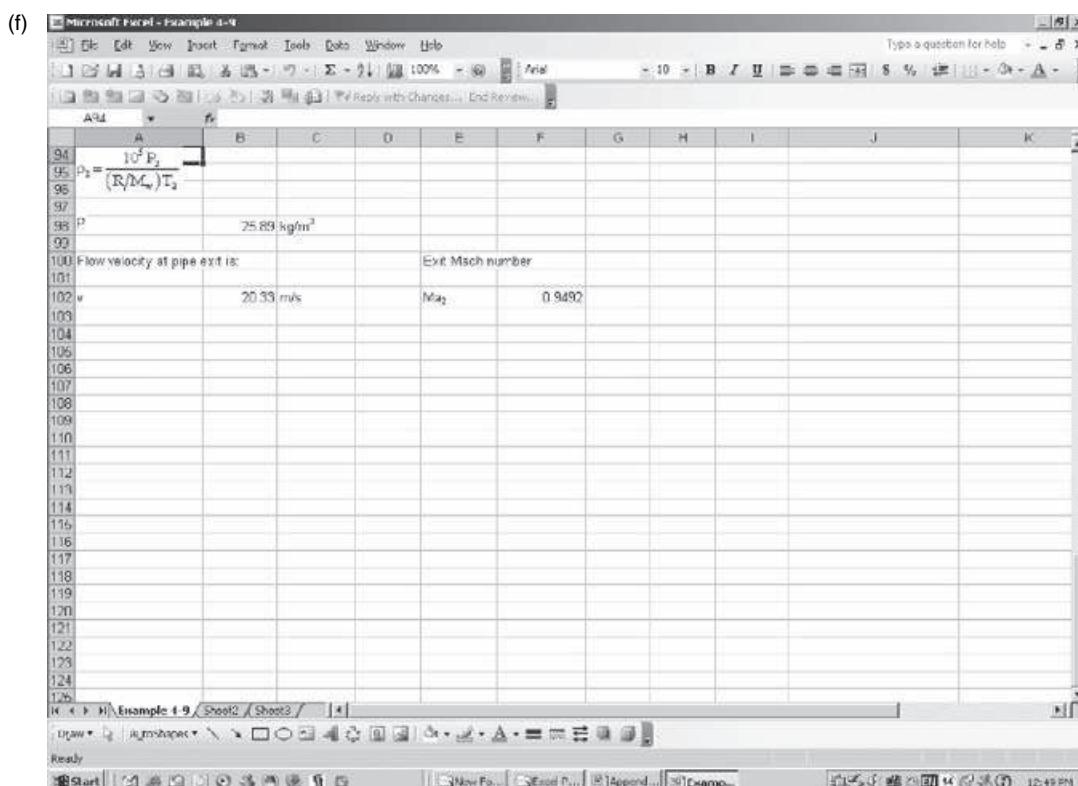


Figure F-1f—(continued).

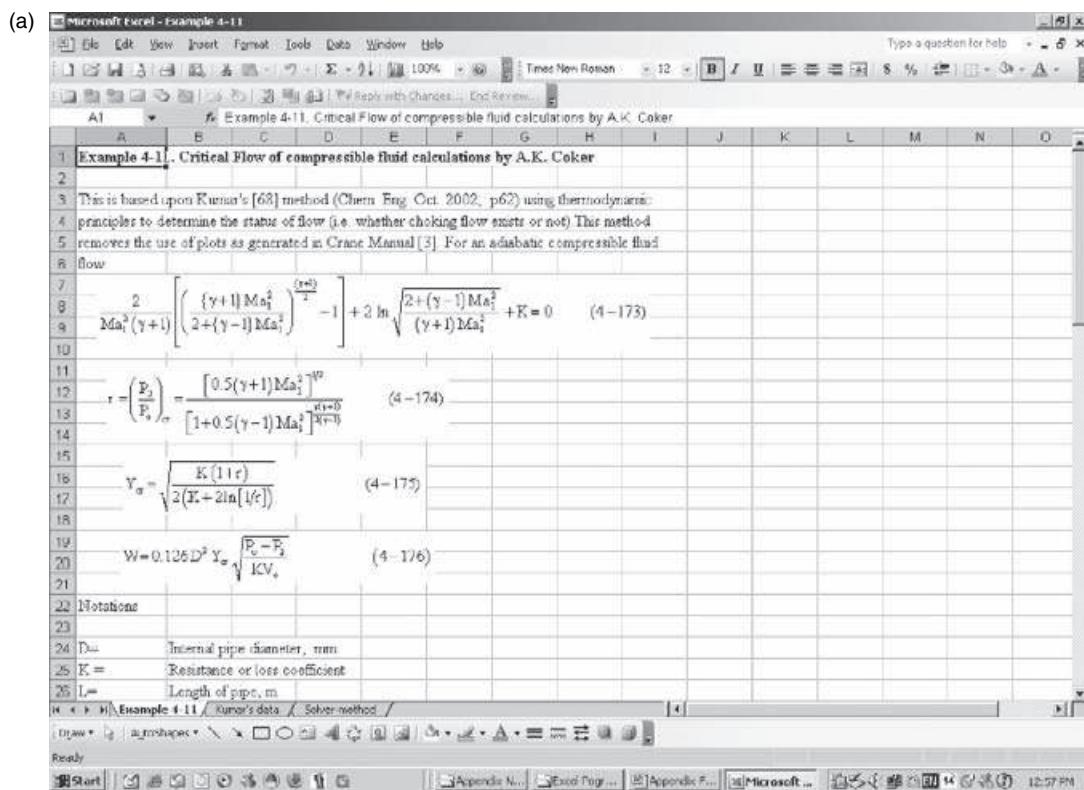


Figure F-2a Solving implicit adiabatic compressible fluid flow equation of Example 4-11 using Goal Seek.

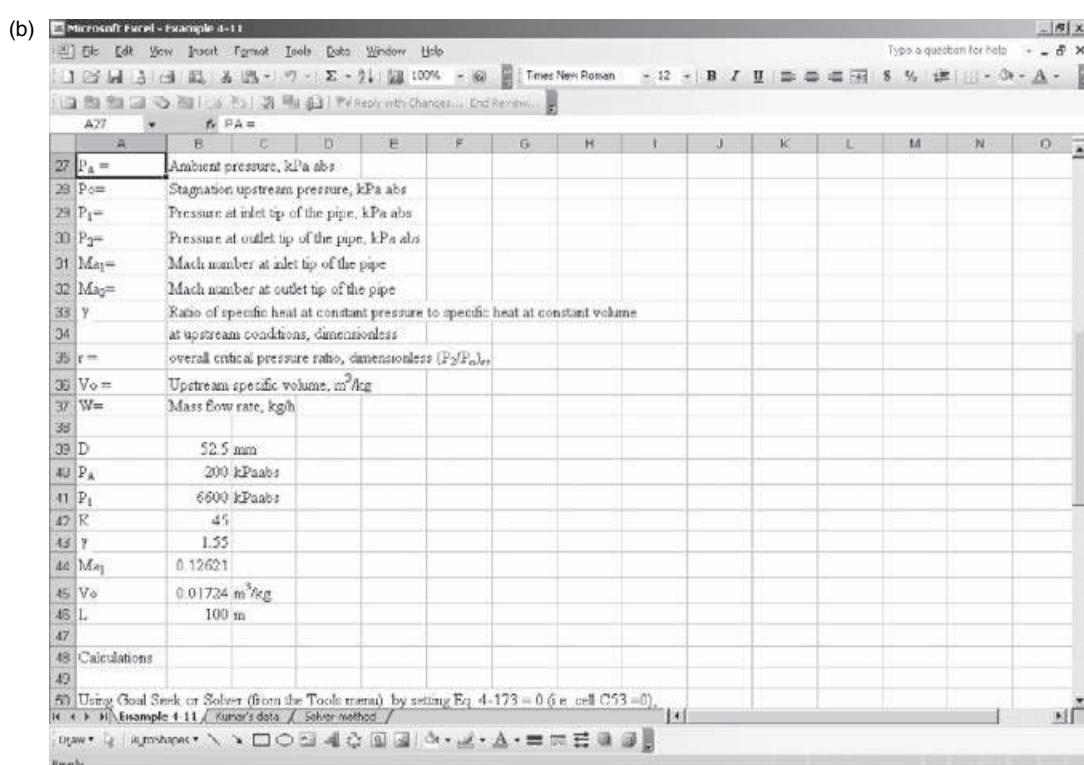
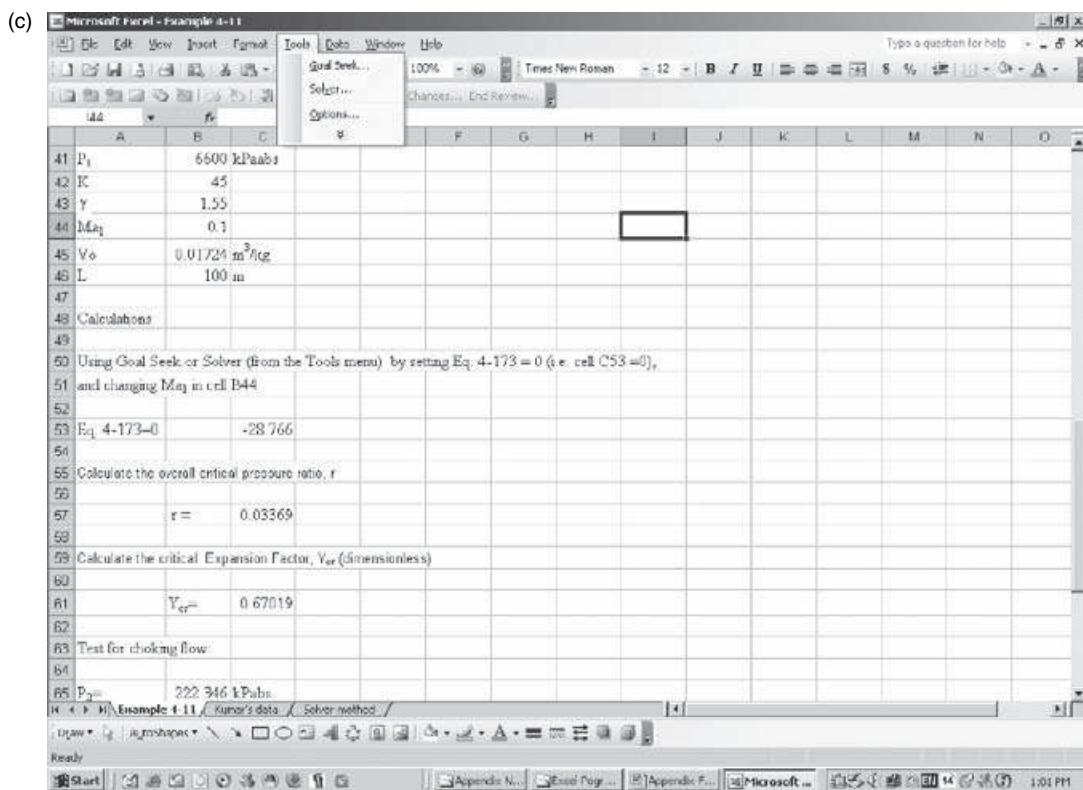
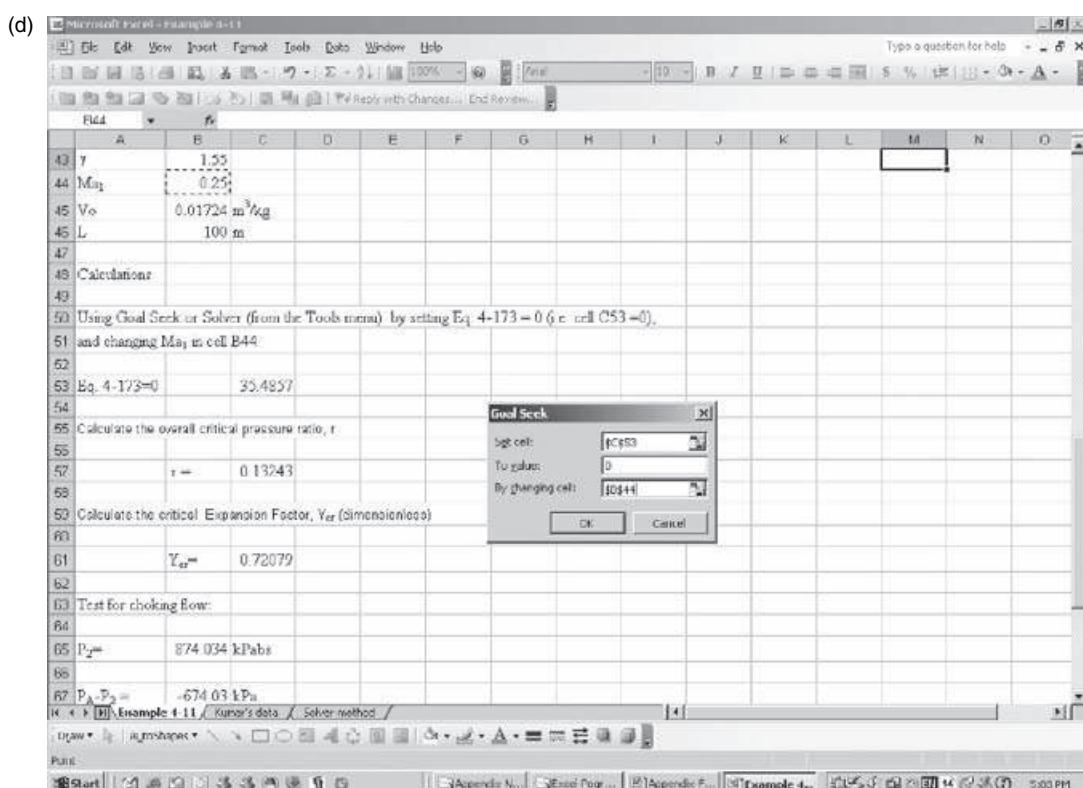


Figure F-2b—(continued).



**Figure F-2c—**(continued).



**Figure F-2d**—(continued).

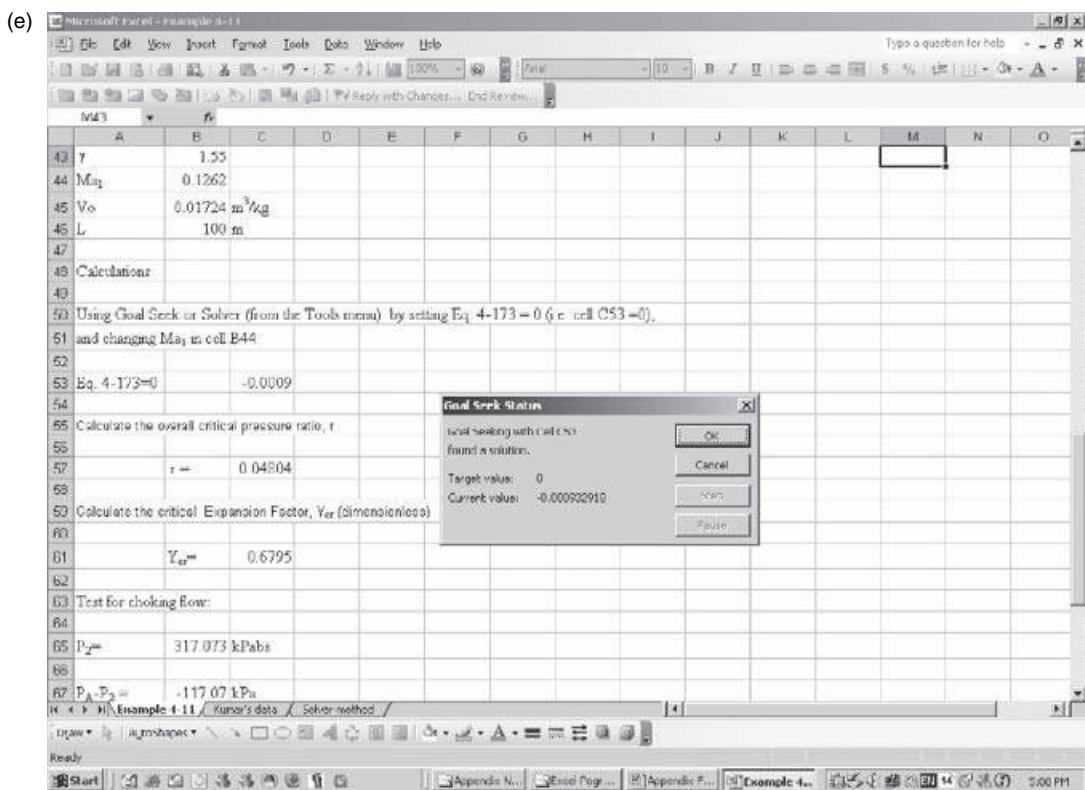


Figure F-2e—(continued).

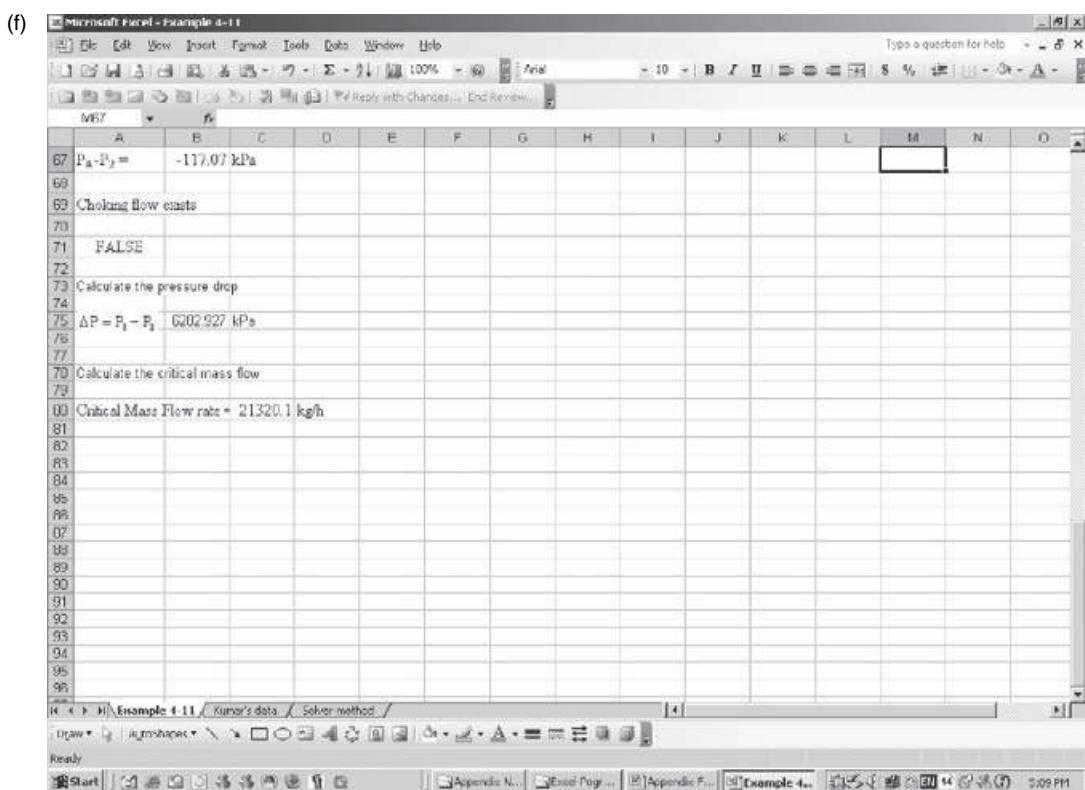


Figure F-2f—(continued).

where

$\varepsilon$  = Relative pipe roughness, in.  
 $D$  = Pipe size, in.  
 $f_D$  = Darcy friction factor  
 $Re$  = number.

Re-arranging the friction factor equation in the form

$$F(f_D) = \frac{1}{f_D^{1/2}} + 0.8686 \ln \left\{ \frac{\varepsilon}{3.7D} + \frac{2.51}{Re \cdot f_D^{1/2}} \right\} \quad (\text{F-2})$$

The worksheet (Solution-1) in Excel spreadsheet program (Prog6a.xls) calculates the friction factor for given relative pipe roughness, pipe size, and Reynolds number. The Excel spreadsheet program Prog6a.xls and the Fortran program PROG6 show how the friction factor is determined. Table F-1 shows the computer results of a 2 in. (internal diameter = 2.067 in.) stainless steel pipe size Sch 40, with pipe roughness of 0.0018 in. and Reynolds number of 184,000 (highly turbulent flow). The calculated friction factor  $f = 0.02063$  with an initial guess of 0.015.

The worksheet (Solution-2) in Excel spreadsheet Prog6a.xls calculates the friction factor for flow of a suspension of fibrous particles to the Reynolds number given by an empirical equation of the form [5]:

$$\frac{1}{\sqrt{f}} = \left( \frac{1}{k} \right) \ln \left( Re \sqrt{f} \right) + \left( 14.0 - \frac{5.6}{k} \right) \quad (\text{F-3})$$

where

$f$  = friction factor  
 $Re$  = Reynolds number  
 $k$  = constant determined by the concentration of the suspension.

Re-arranging Eq. (F-3) gives

$$F(f) = \frac{1}{f^{1/2}} - \frac{1}{k} \ln (Re \cdot f^{1/2}) - 14.0 + \frac{5.6}{k} \quad (\text{F-4})$$

**TABLE F-1 Newton-Raphson Method for a Non-Linear Equation**

Initial Guess of the root $x = 0.0150$		
Iteration	$x$	$F(x)$
1	0.19423E-01	0.22078E+00
2	0.20577E-01	0.93463E-02
3	0.20630E-01	0.17983E-04

*Tolerance met in three iterations*  
Final root  $x = 0.20630E-01$   
 $F(x) = 0.17983E-04$

## FURTHER READING

1. Coker, A.K., *Fortran Programs For Chemical Process Design, Analysis and Simulation*, Gulf Publishing Co., Houston, TX, 1995.
2. Gottfried, B.S., *Spreadsheet Tools for Engineers Excel 5.0 Version*, The McGraw-Hill Co. Inc., 1996.
3. Billio, E.J., *Excel for Chemists – A Comprehensive Guide*, Wiley-VCH, Inc., 1997.
4. Gottfried, B.S., *Spreadsheet Tools for Engineers Excel 2000 Version*, McGraw-Hill's Best – Basic Engineering Series and Tools, 2000.
5. Lee and Duffy, "Relationships Between Velocity Profiles and Drag Reduction in Turbulent Fiber Suspension Flow", *AICHE J.*, 1976, pp. 750–753.

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# Appendix G

## ANALYTICAL TECHNIQUES

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### G.1 USEFUL INTEGRALS

$$\int_0^x \frac{dx}{1-x} = \ln \frac{1}{1-x} \quad (G-1)$$

$$\int_0^x \frac{dx}{(1-x)^2} = \frac{x}{1-x} \quad (G-2)$$

$$\int_0^x \frac{dx}{1+\varepsilon x} = \frac{1}{\varepsilon} \ln(1+\varepsilon x) \quad (G-3)$$

$$\int_0^x \frac{1+\varepsilon x}{1-x} dx = (1+\varepsilon) \ln \frac{1}{1-x} - \varepsilon x \quad (G-4)$$

$$\int_0^x \frac{1+\varepsilon x}{(1-x)^2} dx = \frac{(1-x)x}{1-x} - \varepsilon \ln \frac{1}{1-x} \quad (G-5)$$

$$\int_0^x \frac{(1+\varepsilon x)^2}{(1-x)^2} dx = 2\varepsilon(1+\varepsilon) \ln(1-x) + \varepsilon^2 x + \frac{(1+\varepsilon)^2 x}{1-x} \quad (G-6)$$

$$\int_0^x \frac{dx}{(1-x)(\Theta_B - x)} = \frac{1}{\Theta_B - 1} \ln \frac{\Theta_B - x}{\Theta_B(1-x)} \quad \Theta_B \neq 1 \quad (G-7)$$

$$\int_0^x \frac{dx}{ax^2 + bx + c} = \frac{-2}{2ax+b} + \frac{2}{b} \quad \text{for } b^2 = 4ac \quad (G-8)$$

$$\int_0^x \frac{dx}{ax^2 + bx + c} = \frac{1}{a(p-q)} \ln \left[ \frac{q}{p} \cdot \frac{x-p}{x-q} \right] \quad \text{for } b^2 > 4ac \quad (G-9)$$

$$\int x^n dx = \frac{1}{n+1} x^{n+1} + C \quad (G-10)$$

$$\int \frac{1}{x} dx = \ln x + C \quad (G-11)$$

$$\int \frac{dx}{ax+b} = \frac{1}{a} \ln(ax+b) + C \quad (G-12)$$

$$\int a^x dx = \frac{1}{\ln a} a^x + C \quad (G-13)$$

$$\int e^x dx = e^x + C \quad (G-14)$$

$$\int \sin x dx = -\cos x + C \quad (G-15)$$

$$\int \cos x dx = \sin x + C \quad (G-16)$$

$$\int \tan x dx = -\ln(\cos x) + C = \ln(\sec x) + C \quad (G-17)$$

$$\int \sin ax dx = -\frac{1}{a} \cos ax + C \quad (G-18)$$

$$\int \sin(ax+b) dx = -\frac{1}{a} \cos(ax+b) + C \quad (G-19)$$

$$\int \cos ax dx = \frac{1}{a} \sin ax + C \quad (G-20)$$

$$\int \cos(ax+b) dx = \frac{1}{a} \sin(ax+b) + C \quad (G-21)$$

$$\int \tan ax dx = \frac{1}{a} \ln(\sec ax) + C \quad (G-22)$$

$$\int \frac{dx}{\sqrt{a^2 - x^2}} = \sin^{-1} \frac{x}{a} + C \quad (G-23)$$

$$\int \frac{dx}{a^2 + x^2} = \frac{1}{a} \tan^{-1} \frac{x}{a} + C \quad (G-24)$$

$$\int u dv = uv - \int v du \quad (G-25)$$

$$\int \ln x \, dx = x \ln(x) - x + C \quad (G-26)$$

$$\int \frac{x \, dx}{a^2 + x^2} = \frac{1}{2} \ln(a^2 + x^2) + C \quad (G-27)$$

## USEFUL WEBSITES

- [www.integrals.com](http://www.integrals.com)
- <http://integrals.wolfram.com>

**G.2 LIEBNITZ'S RULE – HIGHER DERIVATIVES OF PRODUCTS**

$$\begin{aligned} D^n(uv) &= uD^n v + \binom{n}{1} (Du)(D^{n-1}v) \\ &\quad + \binom{n}{2} (D^2u)(D^{n-2}v) + \cdots + vD^n u \end{aligned} \quad (G-28)$$

where

$$D^n = \frac{d^n}{dx^n} = \text{nth differential operator}$$

$\binom{n}{1}, \binom{n}{2}, \dots, \binom{n}{n}$  are binomial coefficients.

**EXAMPLE**

$$D^3(uv) = u \frac{d^3v}{dx^3} + 3 \frac{du}{dx} \cdot \frac{d^2v}{dx^2} + 3 \frac{d^2u}{dx^2} \cdot \frac{dv}{dx} + v \frac{d^3u}{dx^3} \quad (G-29)$$

**G.3 DEFINITION OF A DERIVATIVE**

$$\begin{aligned} \frac{dy}{dx} &= \frac{d}{dx} f(x) = f'(x) = y' \\ &= \lim_{\Delta x \rightarrow 0} \frac{f(x + \Delta x) - f(x)}{\Delta x}, \quad y = f(x) \end{aligned} \quad (G-30)$$

**G.4 PRODUCT RULE**

$$\frac{d}{dx}(uv) = V \frac{du}{dx} + U \frac{dv}{dx} \quad (G-31)$$

**G.5 QUOTIENT RULE**

$$\frac{d}{dx}\left(\frac{u}{v}\right) = \frac{v \frac{du}{dx} - u \frac{dv}{dx}}{v^2} \quad (G-32)$$

$$\frac{d}{dx}f(u) = \frac{df(u)}{du} \cdot \frac{du}{dx} \quad (G-33)$$

**G.6 EXPONENTIAL/LOGARITHMIC FUNCTIONS**

$$\frac{d}{dx} \log_a u = \frac{\log_a e}{u} \frac{du}{dx} \quad (G-34)$$

$$\frac{d}{dx} a^u = \frac{1}{u} \frac{du}{dx} \quad (G-35)$$

$$\frac{d}{dx} e^u = e^u \frac{du}{dx} \quad (G-36)$$

**G.7 TAYLOR'S AND MACLAURIN'S SERIES**

## TAYLOR'S SERIES

$$\begin{aligned} f(x+h) &= f(x) + \frac{h}{1!} f'(x) + \frac{h^2}{2!} f''(x) \\ &\quad + \frac{h^3}{3!} f'''(x) + \cdots + \frac{h^n}{n!} f^n(x) + \cdots \end{aligned} \quad (G-37)$$

$$\begin{aligned} f(x+h) &= (x+h)^n = f(x) + \frac{h}{1!} f'(x) + \frac{h^2}{2!} f''(x) \\ &\quad + \frac{h^3}{3!} f'''(x) + \cdots \end{aligned} \quad (G-38)$$

(G-28) when  $h = 0$

$$\begin{aligned} f(x) &= x^n \\ f'(x) &= nx^{n-1} \\ f''(x) &= n(n-1)x^{n-2} \\ f'''(x) &= n(n-1)(n-2)x^{n-3} \end{aligned}$$

Substituting into the expansion for Taylor's expression

$$\begin{aligned} (x+h)^n &= x^n + h nx^{n-1} + \frac{h^2}{2!} n(n-1)x^{n-2} \\ &\quad + \frac{h^3}{3!} n(n-1)(n-2)x^{n-3} + \cdots \end{aligned} \quad (G-39)$$

and re-arranging gives the following.

$$\begin{aligned} (x+h)^n &= x^n + nx^{n-1}h + \frac{n(n-1)}{2!} x^{n-2}h^2 \\ &\quad + \frac{n(n-1)(n-2)}{3!} x^{n-3}h^3 + \cdots \end{aligned} \quad (G-40)$$

## MACLAURIN'S SERIES

Maclaurin's series is a special case of Taylor's series. If  $x = 0$  and  $h$  is replaced by  $x$ , then Taylor's expansion becomes

$$f(x) = f(0) + \frac{x}{1!} f'(0) + \frac{x^2}{2!} f''(0) + \frac{x^3}{3!} f'''(0) + \frac{x^n}{n!} f^n(0) + \cdots \quad (G-41)$$

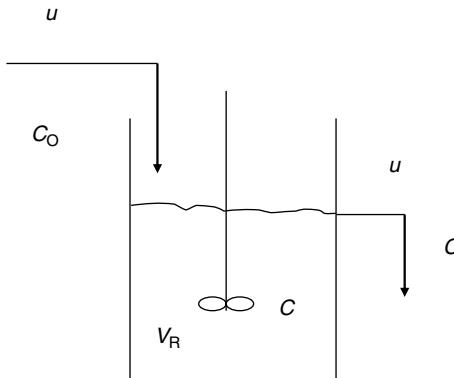
**G.8 DIFFERENTIAL EQUATIONS**

## SEPARATION OF VARIABLES

$$f(x) \, dy + g(x) \, dx = 0 \quad (G-42)$$

Solve by direct integration

$$\int f(y) \, dy + \int g(x) \, dx = C \quad (G-43)$$

**EXAMPLE CONTINUOUS FLOW STIRRED TANK REACTOR**


$u$  = flow rate, L/s

$V_R$  = volume of tank, L

$C_0$  = inlet concentration, g/L

$C$  = tank and outlet concentration, g/L

$T$  = time, s.

Determine  $C$  as a function of time.

*Step 1* A mass balance on tank

Mass in – Mass out = Accumulation in tank

$$u(C_0 - C) dt = V_R dC \quad (G-44)$$

*Step 2* Separate  $C$  and  $t$  and integrate

$$\int \frac{dC}{C_0 - C} = \frac{u}{V_R} \int dt \quad (G-45)$$

$$-\ln(C_0 - C) = \frac{u t}{V_R} + \ln A \quad (G-46)$$

or

$$\ln(C - C_0) = \frac{-ut}{V_R} + \ln A \quad (G-47)$$

$$C - C_0 = Ae^{\frac{-ut}{V_R}} \quad (G-48)$$

*Step 3* Determine  $A$  using the boundary condition

$$t = 0, \quad C = C_i, \text{ therefore } A = C_i - C_0$$

*Step 4.* The final solution is

$$C - C_0 = (C_i - C_0)e^{\frac{-ut}{V_R}} \quad (G-49)$$

or

$$C = C_0 + (C_i - C_0)e^{\frac{-ut}{V_R}} \quad (G-50)$$

**VARIABLE SEPARABLE**

$$f(y) \frac{dy}{dx} + F(x) = 0 \quad (G-51)$$

or

$$f(y) dy + F(x) dx = 0 \quad (G-52)$$

in which  $F(x)$  is a function of  $x$  only, and  $f(y)$  a function of  $y$  only.  
The general solution is

$$\int f(y) dy + \int F(x) dx = C \quad (G-53)$$

**EXAMPLE**

Solve the differential

$$xdy + ydx = 0 \quad (G-54)$$

dividing by  $xy$ ,

$$\frac{dy}{y} + \frac{dx}{x} = 0 \quad (G-55)$$

$$\int \frac{dy}{y} + \int \frac{dx}{x} = 0 \quad (G-56)$$

$$\ln y + \ln x = C_1 \quad (G-57)$$

where the constant  $C_1 = \ln C$

$$\ln y + \ln x = \ln C \quad (G-58)$$

giving

$$xy = c \quad (G-59)$$

The factor  $\frac{1}{xy}$  used to multiply throughout to separate the variables is called an integrating factor.

**G.9 LINEAR EQUATIONS**

An equation of the form

$$\frac{dy}{dx} + Py = Q \quad (G-60)$$

(G-47) is called a linear differential equation where  $P$  and  $Q$  are constants or functions of  $x$  only.

Multiplying such an equation throughout by the integrating factor  $e^{\int P dx}$  gives an equation that can be solved.

$$e^{\int P dx} \left( \frac{dy}{dx} + Py \right) = Q e^{\int P dx} \quad (G-61)$$

Differentiating  $ye^{\int P dx}$  gives

$$e^{\int P dx} \frac{dy}{dx} + e^{\int P dx} Py \quad (G-62)$$

which is the left-hand side of the equation.

Therefore the solution is

$$ye^{\int P dx} = \int \left( Q e^{\int P dx} \right) dx \quad (G-63)$$

**EXAMPLE**

Solve the equation

$$\frac{dy}{dx} + 2y = 0 \quad (G-64)$$

$$\text{Integrating Factor IF} = e^{\int P dx} = e^{\int 2dx} = e^{2x} \quad (G-65)$$

Multiplying the equation by this integrating factor  $e^{2x}$  gives

$$e^{2x} \frac{dy}{dx} + 2e^{2x}y = 0 \quad (G-66)$$

Integrating this equation gives

$$ye^{2x} = A \quad (G-67)$$

where  $A$  is a constant.

Therefore, the solution is

$$y = Ae^{-2x} \quad (G-68)$$

**G.10 EXACT DIFFERENTIAL EQUATION**

If  $M$  and  $N$  are functions of  $(x, y)$  and  $\frac{\partial M}{\partial y} = \frac{\partial N}{\partial x}$  then  $Mdx + Ndy = 0$  is an exact differential equation, the solution of which is

$$\int Mdx + \int \left( N - \int \frac{\partial M}{\partial y} dx \right) dy + C = 0 \quad (G-69)$$

or

$$\int Ndy + \int \left( M - \int \frac{\partial N}{\partial x} dy \right) dx + C = 0 \quad (G-70)$$

If the condition  $\frac{\partial M}{\partial y} = \frac{\partial N}{\partial x}$  is not satisfied, there exists a function  $\psi(x, y) = \psi$  such that  $\frac{\partial(\psi M)}{\partial y} = \frac{\partial(\psi N)}{\partial x}$

**G.11 HOMOGENEOUS SECOND ORDER LINEAR DIFFERENTIAL EQUATION WITH CONSTANT COEFFICIENTS**

$$\frac{d^2y}{dx^2} + a \frac{dy}{dx} + by = 0 \quad (G-71)$$

First construct and solve the auxiliary equation

$$m^2 + am + b = 0 \quad (G-72)$$

If the auxiliary equation has distinct real roots  $m_1$  and  $m_2$ , then the solution is

$$y = Ae^{m_1 x} + Be^{m_2 x} \quad (G-73)$$

If the auxiliary equation has complex roots,  $m_1 + im_2$  and  $m_1 - im_2$ , then the solution is:

$$y = e^{m_1 x} (A \cos m_2 x + B \sin m_2 x) \quad (G-74)$$

If the auxiliary equation has two roots equal to  $m$ , then the solution is

$$y = (Ax + B)e^{mx} \quad (G-75)$$

**G.12 TABLE OF LAPLACE TRANSFORM**

$\bar{f}(s)$	$f(t)$
$\frac{1}{s}$	1
$\frac{1}{s^2}$	$t$
$\frac{1}{s^n} (n = 1, 2, \dots)$	$\frac{t^{n-1}}{(n-1)!}$
$\frac{1}{s-a}$	$e^{at}$
$\frac{1}{(s-a)^2}$	$te^{at}$
$\frac{1}{(s-a)^n} (n = 1, 2, \dots)$	$\frac{1}{(n-1)!} t^{n-1} e^{at}$
$\frac{1}{s^2+a^2}$	$\frac{1}{a} \sin at$
$\frac{s}{s^2+a^2}$	$\cos at$
$\frac{1}{(s-a)^2+b^2}$	$\frac{1}{b} e^{at} \sin bt$
$\frac{(s-a)}{(s-a)^2+b^2}$	$e^{at} \cos bt$
$\frac{e^{-ks}}{s^2}$	$\begin{cases} 0 & \text{when } 0 < t < k \\ t-k & \text{when } t > k \end{cases}$
$s\bar{f}(s) - f(0)$	$\frac{df(t)}{dt}$
$s^2\bar{f}(s) - sf(0) - f'(0)$	$\frac{d^2f(t)}{dt^2}$
$\frac{n!}{s^{n+1}}$ where $n$ is an integer	$t^n$
$\frac{\Gamma(n)}{s^{n+1}} n$ is not an integer	$t^n$
$\frac{1}{(s-b)^2+a^2}$	$\frac{e^{bt} \sin at}{a}$
$\frac{s-b}{(s-b)^2+a^2}$	$e^{bt} \cos at$
$s^n\bar{f}(s) - [s^{n-1}f(0) + s^{n-2}f'(0) + \dots + sf^{n-2}(0) + f^{n-1}(0)]$	$\frac{d^n f(t)}{dt^n}$

where  $f'(0), f''(0), \dots, f^{(n-1)}(0)$  are the values of the first, second, up to the  $(n-1)$ th derivative of the function when the independent variable is zero.

**G.13 CUBIC EQUATIONS**

$$ax^3 + bx^2 + cx + d = 0 \quad (G-76)$$

This reduces by the substitution

$$x = y - \frac{b}{3a} \text{ to } y^3 + py + q = 0 \quad (G-77)$$

where

$$p = \frac{1}{3} \left[ 3 \left( \frac{c}{a} \right) - \left( \frac{b}{a} \right)^2 \right] \quad (G-78)$$

$$q = \frac{1}{27} \left[ 2 \left( \frac{b}{a} \right)^3 - 9 \left( \frac{b}{a} \right) \left( \frac{c}{a} \right) + 27 \left( \frac{d}{a} \right) \right] \quad (\text{G-79})$$

$$\begin{aligned} y_1 &= u + v \\ y_2 &= \frac{(-u+v)}{2} + \frac{(u-v)}{2} i\sqrt{3} \\ y_3 &= \frac{(-u+v)}{2} - \frac{(u-v)}{2} i\sqrt{3}. \end{aligned} \quad (\text{G-80})$$

$$D = \left( \frac{p}{3} \right)^3 + \left( \frac{q}{2} \right)^2 \quad (\text{G-81})$$

$$u = \sqrt[3]{\frac{-q}{2} + \sqrt{D}} \quad v = \sqrt[3]{\frac{-q}{2} - \sqrt{D}} \quad (\text{G-82})$$

If  $a, b, c$ , and  $d$  are real and if

- $D > 0$  there are one real and two conjugate imaginary roots

- $D = 0$  there are three real roots of which at least two are equal
- $D < 0$ , trigonometric formulas of the roots are

$$y_1 = \sqrt[3]{\frac{|p|}{3}} \cos \frac{\phi}{3} \quad (\text{G-83})$$

$$y_2 = -2 \sqrt[3]{\frac{|p|}{3}} \cos \left( \frac{\phi + \pi}{3} \right) \quad (\text{G-84})$$

$$y_2 = -2 \sqrt[3]{\frac{|p|}{3}} \cos \left( \frac{\phi - \pi}{3} \right) \quad (\text{G-85})$$

The value of  $\phi$  is calculated from the expression

$$\cos \phi = \frac{-q/2}{\sqrt[3]{\frac{|p^3|}{27}}} \quad (\text{G-86})$$

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# Appendix H

## NUMERICAL TECHNIQUES

**T**his appendix shows the various numerical techniques that can be employed in solving design problems, which could pose difficulty

if an analytical method is used. The numerical methods can readily be incorporated into computer programs to obtain results of design problems.

### H.1 SIMPSON'S RULE FOR AREA UNDER THE CURVE

Simpson's rule is a numerical integration technique that is widely used in calculating the area under the curve. It is simple and has a greater degree of accuracy than the trapezoidal rule. The Simpson's 1/3 rule is based on quadratic polynomial interpolation.

Figure H-1 shows a section of a curve and three coordinates erected to it at equally spaced intervals along the  $x$ -axis. Simpson's rule states that the area  $P'PQQ'$  is given approximately by the formula

$$\text{Area} = \frac{h}{3} (y_1 + 4y_2 + y_3) \quad (\text{H-1})$$

If we reduce the step size  $h$ , the result becomes more accurate. The interval over which the integral is to be taken is divided into larger number of equal sub intervals as shown in Figure H-2.

We will divide the total area into four sections, namely  $P'PRR'$ ,  $R'RSS'$ ,  $S'STT'$ , and  $T'TQQ'$ .

We shall write down the expression for each area and sum them up to obtain the total area  $P'PQQ'$ .

$$\text{Area of } P'PRR' = \frac{h}{3} (y_1 + 4y_2 + y_3) \quad (\text{H-2})$$

$$\text{Area of } R'RSS' = \frac{h}{3} (y_3 + 4y_4 + y_5) \quad (\text{H-3})$$

$$\text{Area of } S'STT' = \frac{h}{3} (y_5 + 4y_6 + y_7) \quad (\text{H-4})$$

$$\text{Area of } T'TQQ' = \frac{h}{3} (y_7 + 4y_8 + y_9) \quad (\text{H-5})$$

The total area is the sum of the areas  $P'PRR'$ ,  $R'RSS'$ ,  $S'STT'$ , and  $T'TQQ'$ .

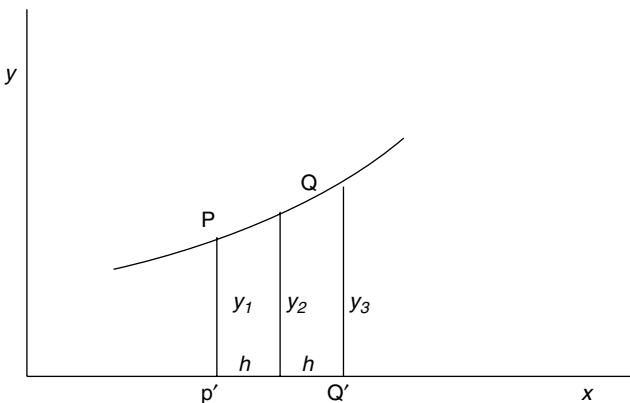


Figure H-1

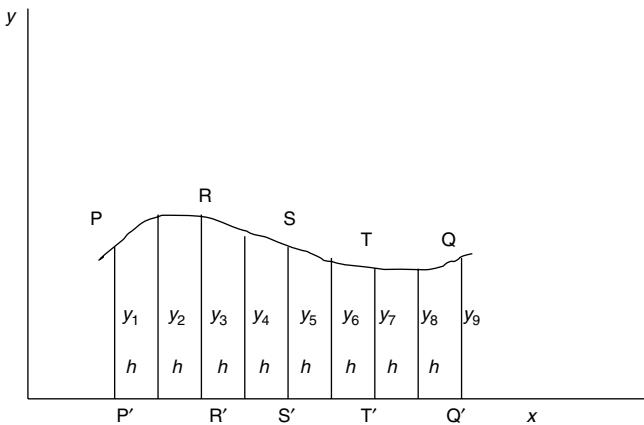


Figure H-2

$$\begin{aligned} \text{Total area} &= \frac{1}{3} h (y_1 + 4y_2 + y_3 + y_3 + 4y_4 + y_5 + y_5 + 4y_6 \\ &\quad + y_7 + y_7 + 4y_8 + y_9) \\ &= \frac{1}{3} h [y_1 + 4(y_2 + y_4 + y_6 + y_8) + 2(y_3 + y_5 + y_7) \\ &\quad + y_9] \end{aligned} \quad (\text{H-6})$$

The Simpson's 1/3 rule for a quadratic integrated over two  $\Delta x$  intervals that are of uniform width or panel is

$$\begin{aligned} \text{Area} &= I = \int_a^b f(x) dx \\ &= \frac{h}{3} (y_1 + 4y_2 + 2y_3 + 4y_4 + 2y_5 + \dots \\ &\quad + 2y_{n-1} + 4y_n + y_{n+1}) + E \\ &= \text{width} \times \text{average height} \end{aligned} \quad (\text{H-7})$$

### SIMPSON'S 3/8 RULE

Simpson's 3/8 rule is derived by integrating a third-order polynomial interpolation formula. For a domain  $(a, b)$  divided into three intervals, it is expressed as

$$\begin{aligned} \text{Area} &= I = \int_a^b f(x) dx = \frac{3h}{8} (y_1 + 3y_2 + 3y_3 + 2y_4 + 3y_5 + \\ &\quad + 3y_6 + \dots + 2y_{n-2} + 3y_{n-1} + 3y_n + y_{n+1}) + E \\ &= \text{width} \times \text{height} \end{aligned} \quad (\text{H-8})$$

**EXAMPLE H-1**

A tracer experiment was carried out in a nozzle type reactor of volume  $V = 5.13\text{ L}$  with liquid rate at  $2.9\text{ L/min}$ . Table H-1 shows data for the exit age distribution  $E(\theta)$  against the dimensionless residence time  $\theta$ . Determine the area under the distribution curve.

**TABLE H-1**

$\theta$	$E(\theta)$	$\theta$	$E(\theta)$
0.000	0.000	1.243	0.403
0.113	0.308	1.356	0.355
0.226	0.995	1.469	0.313
0.339	0.876	1.582	0.275
0.452	0.786	1.695	0.237
0.565	0.720	1.808	0.213
0.678	0.663	1.921	0.171
0.791	0.606	2.034	0.142
0.904	0.545	2.147	0.123
1.017	0.497	2.260	0.109
1.130	0.450	2.373	0.095

(Source: A.K. Coker, Ph.D., *Study of Fast Reactions in Nozzle Type Reactors*, Aston Univ., Birmingham, UK, 1985.)

A computer program PROG7 has been developed to determine the area under the distribution curve from the residence time distribution function  $E(t)$  and expressed as follows:

$$\int_0^\infty E(t) dt = 1 \quad (\text{H-9})$$

The average time spent by material flowing at a rate  $q$  through a volume  $V$

$$\text{equals } \bar{t} = \frac{V}{q} \quad (\text{H-10})$$

We can also express Eq. (H-9) in the form of dimensionless time where

$$\theta = \frac{tq}{V} \quad (\text{H-11})$$

A computer program PROG7 has been developed to determine the area under the curve. Simpson's rule is also easy to use in Microsoft Excel spreadsheet. We can carry this out by entering the  $x$ -values in one column, the  $y$ -values in the next column. We then follow this procedure by an additional column containing the  $y$ -values multiplied by their appropriate constants (i.e., multiplied by 4 or 2, except the initial and final values). We sum the last column to obtain the value of the integral and multiply this by  $\Delta x/3$ .

**H.2 NON-LINEAR EQUATIONS**

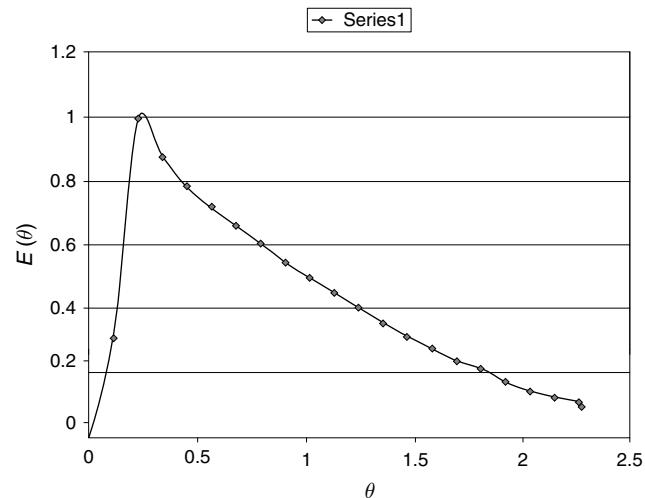
The Newton-Raphson's iterative method is a process for the determination of a real root of an equation  $f(x) = 0$ , given just one point close to the desired root (Figure H-4). If we let  $x_0$  represent the known approximate value of the root of  $f(x) = 0$ , and  $h$  be the difference between the true value  $\alpha$  and the approximate value, we have

$$\alpha = x_0 + h$$

and this becomes

$$\int_0^\infty E(\theta) d\theta = 1 \quad (\text{H-12})$$

Using the Excel spreadsheet program prog7a.xls or Fortran program PROG7, the area under the residence time distribution of the tracer response can be calculated. Table H-1 shows the computer results of the program, and Figure H-3 shows a residence time distribution from a tracer experiment studying the mixing characteristics of a nozzle type reactor that behaves non-ideally [1].

**Figure H-3** Residence time distribution to a tracer response.

From Taylor series,

$$f(x) = f(x_0) + hf'(x_0) + \frac{h}{2!}f''(x_0) + \cdots + \frac{h}{n!}f^n(x_0) \quad (\text{H-13})$$

about  $x_0$ , gives

$$f(\alpha) = f(x_0 + h) = f(x_0) + hf'(x_0) + \frac{h}{2!}f''(\xi) \quad (\text{H-14})$$

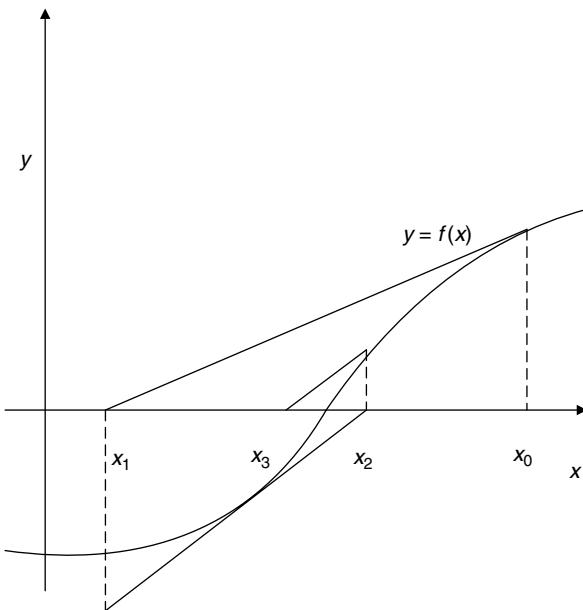
where

$\xi = x_0 + \theta h$ ,  $0 < \theta < 1$ , lies between  $\alpha$  and  $x_0$ . Ignoring the remainder term and writing  $f(\alpha) = 0$ , we have

$$f(x_0) + hf'(x_0) \approx 0 \quad (\text{H-15})$$

so that

$$h \approx -\frac{f(x_0)}{f'(x_0)} \quad (\text{H-16})$$



**Figure H-4** Newton–Raphson’s method.

Therefore, the next root that gives a better estimate than  $x_0$  is

$$x_1 = x_0 - \frac{f(x_0)}{f'(x_0)} \quad (\text{H-17})$$

Better approximations may be obtained by repetition (iteration) of the process. We may write this as

$$x_{n+1} = x_n - \frac{f(x_n)}{f'(x_n)} \quad (\text{H-18})$$

Each iteration provides the point at which the tangent at the original point cuts the  $x$ -axis as shown in Figure H-4. The equation of the tangent at the point  $[x_n, f(x_n)]$  is

$$y - f(x_n) = f'(x_n)(x - x_n) \quad (\text{H-19})$$

Therefore the point  $(x_{n+1}, 0)$  corresponds to

$$-f(x_n) = f'(x_n)(x_{n+1} - x_n) \quad (\text{H-20})$$

which gives

$$x_{n+1} = x_n - \frac{f(x_n)}{f'(x_n)} \quad (\text{H-21})$$

A developed computer program PROG6.FOR is used to solve non-linear (e.g., Colebrooke–White) equation using Newton–Raphson’s method of iteration in Chapter 9.

### SOLUTION OF NON-LINEAR EQUATIONS

Consider a set of  $N$  non-linear equations of the form

$$\begin{aligned} F(x, y, \dots) &= 0 \\ G(x, y, \dots) &= 0 \end{aligned} \quad (\text{H-22})$$

where  $x, y, \dots$  are the roots of the  $N$  equations. These equations can be solved explicitly for the roots. If we consider some points

$(x_1, y_1)$  near the root of definition for the functions  $F, G$ , we can expand both functions by an  $N$ -dimensional Taylor series about the point  $(x_1, y_1)$  as

$$\begin{aligned} F(x, y, \dots) = 0 &= F(x_1, y_1, \dots) + \frac{\partial F}{\partial x}|_0(x - x_1) \\ &\quad + \frac{\partial F}{\partial y}|_0(y - y_1) + \dots \\ G(x, y, \dots) = 0 &= G(x_1, y_1, \dots) + \frac{\partial G}{\partial x}|_0(x - x_1) \\ &\quad + \frac{\partial G}{\partial y}|_0(y - y_1) + \dots \end{aligned} \quad (\text{H-23})$$

Truncating the series after the first-order derivative and rewriting in matrix form will yield

$$\begin{bmatrix} \frac{\partial F}{\partial x}|_0 & \frac{\partial F}{\partial y}|_0 & \dots \\ \frac{\partial G}{\partial x}|_0 & \frac{\partial G}{\partial y}|_0 & \dots \\ \vdots & \vdots & \vdots \end{bmatrix} \begin{bmatrix} x - x_1 \\ y - y_1 \\ \vdots \end{bmatrix} = \begin{bmatrix} F - F_0 \\ G - G_0 \\ \vdots \end{bmatrix} \quad (\text{H-24})$$

We can solve for the roots  $x, y, \dots$  which give  $F = G = \dots = 0$  as

$$\begin{bmatrix} x \\ y \\ \vdots \end{bmatrix} = \begin{bmatrix} x_1 \\ y_1 \\ \vdots \end{bmatrix} - \begin{bmatrix} \frac{\partial F}{\partial x}|_0 & \frac{\partial F}{\partial y}|_0 & \dots \\ \frac{\partial G}{\partial x}|_0 & \frac{\partial G}{\partial y}|_0 & \dots \\ \vdots & \vdots & \vdots \end{bmatrix}^{-1} \begin{bmatrix} F_0 \\ G_0 \\ \vdots \end{bmatrix} \quad (\text{H-25})$$

for  $N = 1$

Equation (H-25) becomes

$$x = x_1 - \frac{F_0}{\frac{\partial F}{\partial x}|_0} \quad (\text{H-26})$$

This is Newton–Raphson’s method for finding the roots of an equation.

When  $N = 2$

$$\begin{aligned} x &= x_1 - \frac{F_0 \frac{\partial G}{\partial x}|_0 - G_0 \frac{\partial F}{\partial x}|_0}{\frac{\partial F}{\partial x}|_0 \cdot \frac{\partial G}{\partial y}|_0 - \frac{\partial F}{\partial y}|_0 \cdot \frac{\partial G}{\partial x}|_0} \\ y &= y_1 - \frac{-F_0 \frac{\partial G}{\partial y}|_0 + G_0 \frac{\partial F}{\partial y}|_0}{\frac{\partial F}{\partial x}|_0 \cdot \frac{\partial G}{\partial y}|_0 - \frac{\partial F}{\partial y}|_0 \cdot \frac{\partial G}{\partial x}|_0} \end{aligned} \quad (\text{H-27})$$

Equation (H-27) is a two-dimensional generalization of Newton’s method. This technique is often employed to solve large sets of non-linear algebraic equations. Care must be taken in choosing initial guess  $(x_1, y_1)$  quite close to the final roots, as the algorithm may diverge. A developed computer program PROG8 is used to solve non-linear equations as illustrated by the following example.

The computer program PROG8 can be used to solve any number of non-linear equations. The partial derivatives of the functions are estimated by the difference quotients when a variable is perturbed by an amount equal to a small value ( $\Delta$ ) used in the program to perturb the  $X$ -values. Table H-2 shows the computer results of Example H-2.

### H.3 SOLUTION OF SIMULTANEOUS, FIRST-ORDER, ORDINARY DIFFERENTIAL EQUATIONS

Analytical solutions to complex kinetic reactions in reactor systems are time-consuming and intractable. The designer must resort to

**EXAMPLE H-2**

A pair of reactions  $A + B \xrightarrow{k_1} 2C$  and  $A + C \xrightarrow{k_2} D$  are conducted in a four-stage continuous flow stirred tank reactors (CFSTR) with  $C_{AO} = 0.9 \text{ mol/m}^3$ ,  $C_{BO} = 0.3 \text{ mol/m}^3$ , and  $C_{CO} = C_{DO} = 0 \text{ mol/m}^3$ . The residence time in each stage is 10 min. Determine the exit concentrations of  $A$  and  $B$  in the four CFSTRs. The rate expressions are

$$(-r_A) = k_1 C_A C_B + k_2 C_A C_C \text{ and } (-r_B) = k_1 C_A C_B \text{ mol/m}^3 \text{ min}$$

where

$$k_1 = 0.3 \text{ m}^3/\text{mol min}$$

$$k_2 = 0.15 \text{ m}^3/\text{mol min.}$$

*Solution*

Figure H-5 shows a battery of CFSTR with  $C_{AO} = 0.9 \text{ mol/m}^3$  and  $C_{BO} = 0.3 \text{ mol/m}^3$  in the first tank and where,  $V_1 = V_2 = V_3 = V_4 = V_R$ ,  $u$  = volumetric flow rate, and the residence time,  $\bar{t} = V_R/u = 10 \text{ min}$ .

*General mass balance*

$$\text{Input by flow} = \text{output by flow} + \text{disappearance by reaction}$$

$$+ \text{accumulation}$$

Assuming that the operation is at steady state and, therefore, accumulation = 0

Mass balance on first CFSTR

The material balance on species A is:

$$uC_{AO} = uC_{A1} + (k_1 C_{A1} C_{B1} + k_2 C_{A1} C_C) V_R \quad (\text{H-28})$$

The stoichiometry between species A, B, and C is

$$\begin{aligned} C_C &= 3(C_{BO} - C_B) - (C_{AO} - C_C) \\ &= 0.9 - 3C_B - 0.9 + C_A \\ C_C &= C_A - 3C_B \end{aligned} \quad (\text{H-29})$$

Substituting Eq. (H-29) into Eq. (H-28) gives

$$C_{AO} = C_{A1} + C_{A1} \bar{t} [k_1 C_{B1} + k_2 (C_{A1} - 3C_{B1})]$$

or

$$0.9 = C_{A1} + 3C_{A1} C_{B1} + 1.5 C_{A1} (C_{A1} - 3C_{B1}) \quad (\text{H-30})$$

Material balance on species B is

$$uC_{B0} = uC_{B1} + k_1 C_{A1} C_{B1} V_R \quad (\text{H-31})$$

or

$$0.3 = C_{B1} + 3C_{A1} C_{B1} \quad (\text{H-32})$$

Second CFSTR Material balance on species A is

$$uC_{A1} = uC_{A2} + [k_1 C_{A2} C_{B2} + k_2 C_{A2} (C_{A2} - 3C_{B2})] V_R \quad (\text{H-33})$$

or

$$C_{A1} = C_{A2} + 10[0.3 C_{A2} C_{B2} + 0.15 C_{A2} (C_{A2} - 3C_{B2})] \quad (\text{H-34})$$

Material balance on species B is

$$uC_{B1} = uC_{B2} + k_1 C_{A2} C_{B2} V_R \quad (\text{H-35})$$

or

$$C_{B1} = C_{B2} + 3C_{A2} C_{B2} \quad (\text{H-36})$$

Rearranging Eq (H-35) and (H-36) yields

$$C_{A2} + 3C_{A2} C_{B2} + 1.5C_{A2} (C_{A2} - 3C_{B2}) = C_{A1}$$

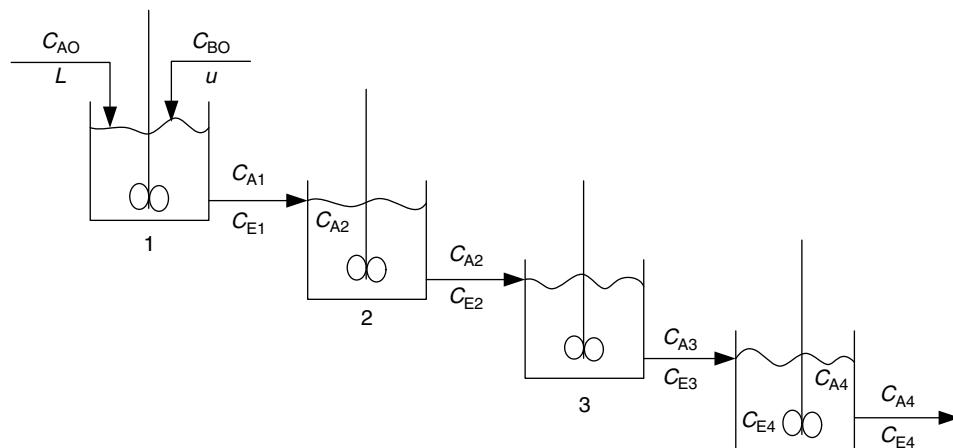
$$C_{B2} + 3C_{A2} C_{B2} = C_{B1} \quad (\text{H-37})$$

Third CFSTR

The material balances on species A and B are

$$C_{A3} + 3C_{A3} C_{B3} + 1.5C_{A3} (C_{A3} - 3C_{B3}) = C_{A2}$$

$$C_{B3} + 3C_{A3} C_{B3} = C_{B2} \quad (\text{H-38})$$



**Figure H-5** A battery of CFSTR.

(continued)

**EXAMPLE H-2 —(continued)**

Fourth CFSTR

The material balances on species A and B are

$$\begin{aligned} C_{A4} + 3C_{A4}C_{B4} + 1.5C_{A4}(C_{A4} - 3C_{B4}) &= C_{A3} \\ C_{B4} + 3C_{A4}C_{B4} &= C_{B3} \end{aligned} \quad (\text{H-39})$$

These are eight non-linear equations involving the material balances of species A and B in the four stirred tank reactors. Rearranging these equations yields the following:

$$\begin{aligned} 1.5C_{A1}^2 - 1.5C_{A1}C_{B1} + C_{A1} - 0.9 &= 0 \\ 3C_{A1}C_{B1} + C_{B1} - 0.3 &= 0 \\ 1.5C_{A2}^2 - 1.5C_{A2}C_{B2} + C_{A2} - C_{A1} &= 0 \\ 3C_{A2}C_{B2} + C_{B2} - C_{B1} &= 0 \\ 1.5C_{A3}^2 - 1.5C_{A3}C_{B3} + C_{A3} - C_{A2} &= 0 \\ 3C_{A3}C_{B3} + C_{B3} - C_{B2} &= 0 \\ 1.5C_{A4}^2 - 1.5C_{A4}C_{B4} + C_{A4} - C_{A3} &= 0 \\ 3C_{A4}C_{B4} + C_{B4} - C_{B3} &= 0 \end{aligned} \quad (\text{H-40})$$

A computer program PROG8 was developed using the Newton-Raphson's method to determine the outlet concentration of species A and B from the four stirred tank reactors. The eight equations (Eqs H-40) are supplied as functions in the subroutine of PROG8 as

$$\begin{aligned} F(1) &= 1.5 * X(1) * X(1) - 1.5 * X(1) * X(2) + X(1) - 0.9 \\ F(2) &= 3.0 * X(1) * X(2) + X(2) - 0.3 \end{aligned}$$

$$F(3) = 1.5 * X(3) * X(3) - 1.5 * X(3) * X(4) + X(3) - X(1)$$

$$F(4) = 3.0 * X(3) * X(4) + X(4) - X(2)$$

$$F(5) = 1.5 * X(5) * X(5) - 1.5 * X(5) * X(6) + X(5) - X(3)$$

$$F(6) = 3.0 * X(5) * X(6) + X(6) - X(4)$$

$$F(7) = 1.5 * X(7) * X(7) - 1.5 * X(7) * X(8) + X(7) - X(5)$$

$$F(8) = 3.0 * X(7) * X(8) + X(8) - X(6) \quad (\text{H-41})$$

where

$$\begin{aligned} C_{A1} &= X(1), C_{B1} = X(2), C_{A2} = X(3), C_{B2} = X(4) \\ C_{A3} &= X(5), C_{B3} = X(6), C_{A4} = X(7), C_{B4} = X(8). \end{aligned}$$

The exit concentrations of species A and B from the computer results and the initial guesses of

$C_{A1} = 0.1, C_{B1} = 0.1, C_{A2} = 0.1, C_{B2} = 0.1, C_{A3} = 0.1, C_{B3} = 0.1, C_{A4} = 0.1, C_{B4} = 0.1$ , are as follows.

$$\begin{array}{ll} C_{A1} = X(1) = 0.54599 & C_{B1} = X(2) = 0.11372 \\ C_{A2} = X(3) = 0.37030 & C_{B2} = X(4) = 0.05386 \\ C_{A3} = X(5) = 0.27118 & C_{B3} = X(6) = 0.02958 \\ C_{A4} = X(7) = 0.21107 & C_{B4} = X(8) = 0.01792. \end{array}$$

**TABLE H-2 Newton Method For Solving Non-linear Equations**

Initial roots

$$\begin{aligned} X(1) &= 0.1000 \\ X(2) &= 0.1000 \\ X(3) &= 0.1000 \\ X(4) &= 0.1000 \\ X(5) &= 0.1000 \\ X(6) &= 0.1000 \\ X(7) &= 0.1000 \\ X(8) &= 0.1000 \end{aligned}$$

Final roots

$$\begin{aligned} X(1) &= 0.54599 \\ X(2) &= 0.11372 \\ X(3) &= 0.37030 \\ X(4) &= 0.05386 \\ X(5) &= 0.27118 \\ X(6) &= 0.02958 \\ X(7) &= 0.21107 \\ X(8) &= 0.01792 \end{aligned}$$

numerical techniques with the aid of a computer for his or her solutions. The reactions taking place in batch and piston flow reactions involve a set of simultaneous, first-order, ordinary differential equations. Several numerical methods have been used for solving sets of equations, and the most popular method is the Runge-Kutta fourth order. The Runge-Kutta is a powerful integration technique that can be easily implemented on a personal computer. The only drawback is its instability if the step size is too large. In a set of equations, the Runge-Kutta algorithm uses the same step size for

each member of the set of equations. This causes practical problems if the set is *stiff* where some members of these equations have characteristic times much smaller than other members of the equations. An example is the free-radical kinetics reaction, which has rates that may differ by three orders of magnitude.

In general, a system of  $n$ th first-order equations will be of the form

$$\frac{dy_i}{dx} = f_i(x, y_0, y_1, y_2, \dots, y_{n-1}), \quad i = 0, 1, 2, 3, \dots, n \quad (\text{H-42})$$

with  $n$  initial conditions  $y_i(X_0) = A_i, i = 0, 1, 2, \dots, n$

Consider the system of two equations:

$$\frac{dy}{dx} = f(x, y, z)y(x_0) = y_0 \quad (\text{H-43})$$

$$\frac{dz}{dx} = g(x, y, z)z(x_0) = z_0 \quad (\text{H-44})$$

We may advance the solution of  $y$  and  $z$  to new values at  $x_1 = x_0 + h$  using any of the one-step or Runge-Kutta methods.

In general, our solutions will be advanced using expressions of the form

$$y(x_1) = y(x_0) + K \quad (\text{H-45})$$

$$z(x_1) = z(x_0) + L \quad (\text{H-46})$$

where the nature of  $K$  or  $L$  depends on the method being applied. For the Runge-Kutta fourth order

$$K = \frac{K_1 + 2K_2 + 2K_3 + K_4}{6} \quad (\text{H-47})$$

and

$$L = \frac{L_1 + 2L_2 + 2L_3 + L_4}{6} \quad (\text{H-48})$$

where

$$K_1 = hf(x_0, y_0, z_0) \quad (\text{H-49})$$

$$L_1 = hg(x_0, y_0, z_0) \quad (\text{H-50})$$

$$K_2 = hf\left(x_0 + \frac{1}{2}h, y_0 + \frac{1}{2}K_1, z_0 + \frac{1}{2}L_1\right) \quad (\text{H-51})$$

$$L_2 = hg\left(x_0 + \frac{1}{2}h, y_0 + \frac{1}{2}K_1, z_0 + \frac{1}{2}L_1\right) \quad (\text{H-52})$$

$$K_3 = hf\left(x_0 + \frac{1}{2}h, y_0 + \frac{1}{2}K_2, z_0 + \frac{1}{2}L_2\right) \quad (\text{H-53})$$

$$L_3 = hg\left(x_0 + \frac{1}{2}h, y_0 + \frac{1}{2}K_2, z_0 + \frac{1}{2}L_2\right) \quad (\text{H-54})$$

$$K_4 = hf(x_0 + h, y_0 + K_3, z_0 + L_3) \quad (\text{H-55})$$

$$L_4 = hg(x_0 + h, y_0 + K_3, z_0 + L_3) \quad (\text{H-56})$$

#### H.4 EXTENSION OF RUNGE-KUTTA METHODS

##### RUNGE-KUTTA-GILL METHOD

The Runge-Kutta-Gill method is the most widely used single-step method for solving ordinary differential equations. For the differential equation

$$\frac{dy}{dx} = f(x, y), y(x_n) = y_n \quad (\text{H-57})$$

$$y_{n+1} = y_n + \frac{1}{6} \left[ k_1 + 2 \left( 1 - \frac{1}{\sqrt{2}} \right) k_2 + 2 \left( 1 + \frac{1}{\sqrt{2}} \right) k_3 + k_4 \right] + O(h^5) \quad (\text{H-58})$$

where

$$k_1 = hf(x_i, y_i)$$

$$k_2 = hf\left(x_i + \frac{h}{2}, y_i + \frac{1}{2}k_1\right)$$

$$k_3 = hf\left(x_i + \frac{h}{2}, y_i + \left[ \frac{-1}{2} + \frac{1}{\sqrt{2}} \right] k_1 + \left[ 1 - \frac{1}{\sqrt{2}} \right] k_2\right)$$

$$k_4 = hf\left(x_i + h, y_i - \frac{1}{\sqrt{2}}k_2 + \left[ 1 + \frac{1}{\sqrt{2}} \right] k_3\right) \quad (\text{H-59})$$

##### THE RUNGE-KUTTA-MERSON METHOD

The Runge-Kutta-Merson method outlines a process for deciding the step size for better predetermined accuracy. For this method, five functions are evaluated at every step. The algorithm is

$$k_1 = hf(x_i, y_i)$$

$$k_2 = hf\left(x_i + \frac{h}{3}, y_i + \frac{k_1}{3}\right)$$

$$k_3 = hf\left(x_i + \frac{h}{3}, y_i + \frac{k_1}{6} + \frac{k_2}{6}\right)$$

$$k_4 = hf\left(x_i + \frac{h}{2}, y_i + \frac{k_1}{8} + \frac{3k_3}{8}\right)$$

$$k_5 = hf\left(x_i + h, y_i + \frac{k_1}{2} - \frac{3k_3}{2} + 2k_4\right)$$

$$y_{n+1} = y_n + \frac{1}{6}(k_1 + 4k_4 + k_5) + O(h^5) \quad (\text{H-60})$$

We can estimate the local error from a weighted sum of the individual estimate.

$$E = \frac{1}{30}(2k_1 - 9k_3 + 8k_4 - k_5) \quad (\text{H-61})$$

The following Example H-3 illustrates the use of Runge-Kutta fourth order with the Excel spreadsheet.

#### H.5 PARTIAL DIFFERENTIAL EQUATION

If two or more independent variables are involved in a differential equation, we can express the differential equation as partial differential equation (PDE).

We shall consider the second-order equation (PDE) of the form

$$A \frac{\partial^2 u}{\partial x^2} + B \frac{\partial^2 u}{\partial x \partial y} + C \frac{\partial^2 u}{\partial y^2} + D \left( x, y, u, \frac{\partial u}{\partial x}, \frac{\partial u}{\partial y} \right) = 0 \quad (\text{H-65})$$

where the coefficients  $A$ ,  $B$ ,  $C$ , and  $D$  are functions of  $x$ ,  $y$ , and  $u$ ;  $x$  and  $y$  are the independent variables and  $u$  is the dependent variable. We can classify Eq. (H-65) with respect to the sign of the discriminant  $\Delta = (B^2 - 4AC)$ . The equation is elliptical if  $\Delta < 0$ , hyperbolic if  $\Delta > 0$ , and parabolic type if  $\Delta = 0$ .

##### ELLIPTICAL EQUATION

$$\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} = -u \quad (\text{H-66})$$

and the Laplace equation

$$\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} = 0 \quad (\text{H-67})$$

The coefficients  $A = C = 1$ ,  $B = 0$  and  $B^2 - 4AC = -4$  are examples of elliptical equation.

##### HYPERBOLIC EQUATION

The wave equation

$$\frac{\partial^2 u}{\partial t^2} = \frac{Tg}{w} \frac{\partial^2 u}{\partial x^2} \quad (\text{H-68})$$

is a hyperbolic type where the coefficients  $A = 1$ ,  $B = 0$ ,  $C = -1$  and  $B^2 - 4AC = 4$

##### PARABOLIC EQUATION

The heat conduction equation

$$\frac{\partial u}{\partial t} = \frac{k}{cp} \frac{\partial^2 u}{\partial x^2} \quad (\text{H-69})$$

is a parabolic type where the coefficients  $A = 0$ ,  $B = 0$ ,  $C = 1$  and  $B^2 - 4AC = 0$ .

A method for solving the above partial differential equations is to replace the derivatives by difference quotients; that is, converting the equation to a difference equation. We can then write the difference equation that corresponds to each point at the intersections (nodes) of a grid work that subdivides the region of interest at which the function values are known. Solving these equations

### EXAMPLE H-3

Consider the dynamic response of component A in a CFSTR undergoing a first-order reaction represented by

$$(-r_A) = kC_A \left( \frac{\text{mol}}{\text{dm}^3 \text{ min}} \right)$$

The rate constant decay is

$$k = k_o - at^2 \text{ (min}^{-1}\text{)}$$

The first-order differential equation from the mass balance is:

$$\frac{dC_A}{dt} = \frac{u}{V_R} C_{A0} - \left( \frac{u}{V_R} + k_o - at^2 \right) C_A \quad (\text{H-62})$$

Spreadsheet programming: The Excel program (Prog8a.xls) is used to create numerical solutions using the fourth-order Runge-Kutta by rearranging the above equation to yield

$$\Delta C_A = C_A - C_{A0} = \left[ \frac{uC_{A0}}{V_R} - \left( \frac{u}{V_R} + k_o - at^2 \right) C_A \right] \Delta t \quad (\text{H-63})$$

The transient response concentration  $C_A$  is

$$C_A = C_{A0} + \left[ \frac{uC_{A0}}{V_R} - \left( \frac{u}{V_R} + k_o - at^2 \right) C_A \right] \Delta t \quad (\text{H-64})$$

Figure H-6 shows the spreadsheet snap shot of the numerical Runge-Kutta method. Further details are given by Coker, A.K. (*Hydroc. Proc.*, Dec 2004, pp. 77–85).

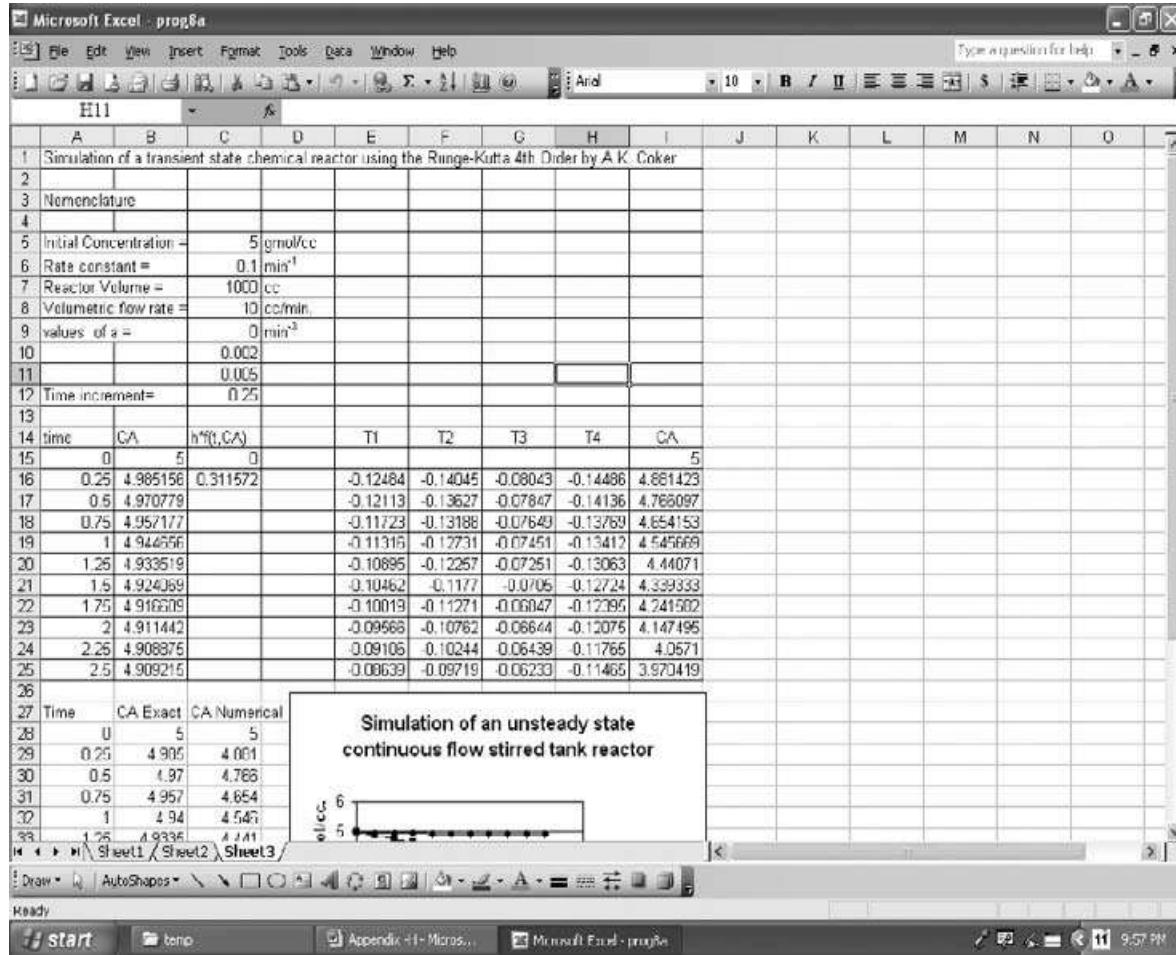


Figure H-6 Simulation of CFSTR reaction with the Excel program.

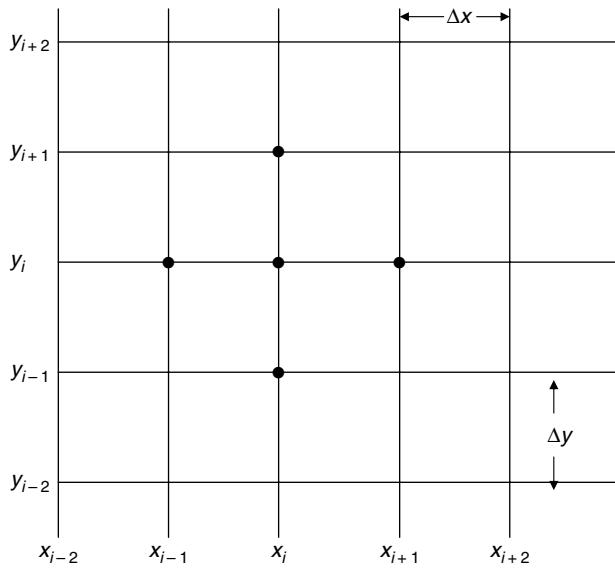


Figure H-7

simultaneously gives values for the function at each node that approximates the true values.

Let  $h = \Delta x$  equal spacing of grid work in the  $x$ -direction as shown in the Figure H-7.

From Taylor series,

$$\begin{aligned} f(x_n + h) &= f(x_n) + f'(x_n)h + \frac{f''(x_n)h^2}{2} + \frac{f'''(x_n)h^3}{6} \\ &\quad + \frac{f''''(x_n)h^4}{24} + \dots \text{ for } x_n < \xi_1 < x_n + h \end{aligned} \quad (\text{H-70})$$

and

$$\begin{aligned} f(x_n - h) &= f(x_n) - f'(x_n)h + \frac{f''(x_n)h^2}{2} - \frac{f'''(x_n)h^3}{6} \\ &\quad + \frac{f''''(x_n)h^4}{24} + \dots \text{ for } x_n - h < \xi_2 < x_n \end{aligned} \quad (\text{H-71})$$

It follows that

$$f(x_n + h) + f(x_n - h) = 2f(x_n) + f''(x_n)h^2 + \frac{f'''(\xi)h^3}{12} \quad (\text{H-72})$$

or

$$f''(x_n) + \frac{f'''(\xi)h^2}{12} = \frac{f(x_n + h) - 2f(x_n) + f(x_n - h)}{h^2} \quad \text{for } x_n - h < \xi < x_n + h \quad (\text{H-73})$$

Using the subscript notation, we have

$$f''(x_n) + O(h^2) = \frac{f_{n+1} - 2f_n + f_{n-1}}{h^2} \quad (\text{H-74})$$

where the subscripts of  $f$  indicate the  $x$ -values at which it is evaluated. The order relation  $O(h^2)$  shows that error approaches proportionality to  $h^2$  as  $h \rightarrow 0$ .

Similarly, the first derivative is approximated to

$$f(x_n + h) - f(x_n - h) = 2f'(x_n)h + O(h^2) \quad (\text{H-75})$$

or

$$f'(x_n) = \frac{f(x_n + h) - f(x_n - h)}{2h} \quad (\text{H-76})$$

$$= \frac{f_{n+1} - f_{n-1}}{2h} \quad (\text{H-77})$$

The first derivative could also be approximated by the forward or backward difference, but would have an error of  $O(h)$ . The central difference approximation gives the more accurate approximation.

When  $f$  is a function of both  $x$  and  $y$ , we can obtain the second partial derivative with respect to  $x$ ,  $\frac{\partial^2 u}{\partial x^2}$ , by holding  $y$  constant and evaluating the function at three points where  $x$  equals  $x_n$ ,  $x_n + h$ , and  $x_n - h$ . Correspondingly, the partial derivative  $\frac{\partial^2 u}{\partial y^2}$  is determined by holding  $x$  constant.

Consider the Laplace equation on a region in the  $xy$  plane. We subdivide the region with equispaced lines parallel to the  $x$ - and  $y$ -axes. Consider the region near  $(x_i, y_i)$ .

$$\nabla^2 u = \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} = 0 \quad (\text{H-78})$$

We can replace the derivatives by difference quotients which approximate the derivatives at point  $(x_i, y_i)$ . Then we have

$$\begin{aligned} \nabla^2 u(x_i, y_i) &= \frac{u(x_{i+1}, y_i) - 2u(x_i, y_i) + u(x_{i-1}, y_i)}{\Delta x^2} \\ &\quad + \frac{u(x_i, y_{i+1}) - 2u(x_i, y_i) + u(x_i, y_{i-1})}{\Delta y^2} = 0 \end{aligned} \quad (\text{H-79})$$

or

$$\nabla^2 u_{i,j} = \frac{u_{i+1,j} - 2u_{i,j} + u_{i-1,j}}{(\Delta x)^2} + \frac{u_{i,j+1} - 2u_{i,j} + u_{i,j-1}}{(\Delta y)^2} = 0 \quad (\text{H-80})$$

If we let  $\Delta x = \Delta y = h$ , the PDE becomes

$$\nabla^2 u_{i,j} = \frac{1}{h^2} (u_{i+1,j} + u_{i-1,j} + u_{i,j+1} + u_{i,j-1} - 4u_{i,j}) = 0 \quad (\text{H-81})$$

This is known as the standard five-point formula, as five points are involved in the relationship of Eq. (H-81) which points to the right, left, above, and below the central point  $(x_i, y_i)$ . We can write Eq. (H-81) as

$$u_{i,j} = \frac{1}{4} (u_{i+1,j} + u_{i-1,j} + u_{i,j+1} + u_{i,j-1}) \quad (\text{H-82})$$

Instead of Eq. (H-82), we may also use the formula

$$u_{i,j} = \frac{1}{4} (u_{i-1,j-1} + u_{i+1,j-1} + u_{i+1,j+1} + u_{i-1,j+1}) \quad (\text{H-83})$$

Therefore, the general procedure is to approximate the PDE by a finite difference transformation and then to obtain the solution at the mesh points, using the finite difference approximations. Other numerical methods of solution are the implicit Crank–Nicolson method or the alternating direction implicit scheme (ADI) by Peaceman and Rachford. Details of these methods are illustrated in Numerical analysis texts.

## H.6 THE EXPLICIT METHOD

The explicit method divides space and time into discrete uniform subintervals and replaces both time and space derivatives by finite difference approximations, permitting one to easily compute values of the function at a time  $\Delta t$  after the initial time. These values are then used to compute a second set of values and the process is repeated. An approach to solving parabolic partial differential equations by a numerical method is to replace the partial derivatives by finite-difference approximations. An example is the one-dimensional heat flow equation.

$$\frac{\partial^2 u}{\partial x^2} = \frac{c\rho}{k} \frac{\partial u}{\partial t} \quad (\text{H-84})$$

We can use the relations

$$\left. \frac{\partial^2 u}{\partial x^2} \right|_{t=t_j} = \frac{u_{i+1}^j - 2u_i^j + u_{i-1}^j}{(\Delta x)^2} + O(\Delta x)^2 \quad (\text{H-85})$$

and

$$\left. \frac{\partial u}{\partial t} \right|_{t=t_j} = \frac{u_i^{j+1} - u_i^j}{\Delta t} + O(\Delta t) \quad (\text{H-86})$$

The subscripts are used to denote the position and superscripts for time.

Substituting Eqs (H-85) and (H-86) into Eq. (H-84) and solving for  $u_i^{j+1}$  gives the equation for the forward-difference method:

$$u_i^{j+1} = \frac{k\Delta t}{c\rho(\Delta x)^2} (u_{i+1}^j - u_{i-1}^j) + \left(1 - \frac{2k\Delta t}{c\rho(\Delta x)^2}\right) u_i^j \quad (\text{H-87})$$

Solving for  $u_i^{j+1}$  in terms of the temperatures at time  $t_j$  in Eq. (H-87) for a parabolic partial-differential equation involves subdividing the length into uniform subintervals and applying the finite-difference approximation to Eq. (H-84) at each point where  $u$  is not known. Eq. (H-87) then gives the values of  $u$  at each interior point at  $t = t_1$  since the values at  $t = t_0$  are given by the initial conditions. It can be used to get values at  $t_2$  using the values at  $t_1$  as initial conditions, so the solution can be stepped forward in time. At the end points, the boundary conditions will determine  $u$ .

The relative size of the time and distance steps,  $\Delta t$  and  $\Delta x$ , affects Eq. (H-87). If the ratio of  $\Delta t/(\Delta x)^2$  is chosen so that  $k\Delta t/c\rho(\Delta x)^2 = \frac{1}{2}$ , the equation is simplified in that the last term vanishes and we have

$$u_i^{j+1} = \frac{1}{2} (u_{i+1}^j + u_{i-1}^j) \quad (\text{H-88})$$

If the value  $k\Delta t/c\rho(\Delta x)^2$  is chosen as less than one-half, there will be improved accuracy (limited by the errors dependent on the size of  $\Delta x$ ). If the value is chosen greater than one-half, which would reduce the number of calculations required to advance the solution through a given interval of time, the phenomenon of instability sets in. An example is used to illustrate the explicit method.

## H.7 INITIAL VALUE METHODS

Another method for solving boundary value problems is to treat them like initial value problems. Since a second-order equation can be reduced to two first-order equations, two initial conditions are required. One condition will be known at a boundary; simply assume a value for the other dependent variable at that same boundary, integrate to the other side, and check if the required boundary condition is satisfied. If not, change the initial value and repeat the integration. The result of this method depends upon the skill with which the iterations are employed in the program.

### FINITE DIFFERENCE METHOD IN EXCEL

The Microsoft Excel spreadsheet can be used to program the finite difference method and use the “Calculation” feature to handle the circular reference. First, turn off the iteration, prepare the spreadsheet, and then turn the calculation back on. Whether this converges depends upon the initial guess.

Consider a reaction and diffusion in a flat layer with a first-order reaction represented by

$$\frac{d}{dx} \left( D \frac{dc}{dx} \right) = kc, \text{ with boundary conditions } \frac{dc}{dx}(0) = 0, \quad c(R) = c_o \quad (\text{H-89})$$

where

$D$  = diffusivity

$c$  = concentration

$x$  = position (distance).

The idea in the finite difference method is that the differential equation, valid for all  $x$  positions, is replaced by a set of equations representing the equation only at the grid points. Using the finite difference method as derived above, the equation at the grid point is

$$D \frac{c_{i+1} - 2c_i + c_{i-1}}{\Delta x^2} = kc_i, \text{ where } c_i = c(x_i) \quad (\text{H-90})$$

Re-arranging Eq. (H-90) to the following form gives

$$c_i = \frac{c_{i+1} + c_{i-1}}{(2 + k\Delta x^2/D)} \quad (\text{H-91})$$

Identify the cells (B1, C1, and D1) in the spreadsheet with the value of concentration at a node  $i$  as shown below.

A1	B1	C1	D
$k\Delta x^2/D$	$c_{i-1}$	$c_i$	$c_{i+1}$

Put the value of  $k\Delta x^2/D$  in cell A1. The equation for cell C1 is represented by Eq. (H-92) as

$$= (\text{D1} + \text{B1})/(2 + \$\text{A\$1}) \quad (\text{H-92})$$

Next, copy this equation over a series of cells that correspond to the number of grid points. For the first and last cell (grid point) use a different equation appropriate to the boundary condition. Then turn on the iteration feature to get the solution.

The next step is to resolve the problem using more grid point and a smaller  $\Delta x$ . This entails that the spreadsheet must be programmed, which gives an indication of whether the answer changes much as the mesh is refined. This ensures that the problem is accurately solved.

## H.8 FINITE DIFFERENCE METHOD FOR ELLIPTIC EQUATIONS

Consider the heat condition equation with a heat generation term as:

$$k \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) = Q \quad (\text{H-93})$$

### EXAMPLE H-4

A large flat steel plate is 2 cm thick. If the initial temperatures ( $^\circ\text{C}$ ) within the plate are given, as a function of the distance from one face, by the equations

$$\begin{aligned} u &= 100x & \text{for } 0 \leq x \leq 1 \\ u &= 100(2-x) & \text{for } 1 \leq x \leq 2 \end{aligned}$$

find the temperatures as a function of  $x$  and  $t$  if both faces are maintained at  $0^\circ\text{C}$  [3].

Data:

$$k = 0.13 \text{ cal/s cm } ^\circ\text{C}$$

$$c = 0.11 \text{ cal/g } ^\circ\text{C}$$

$$\rho = 7.8 \text{ g/cm}^3$$

*Solution*

since the plate is large, the lateral flow of heat relative to the flow perpendicular to the faces can be neglected, and therefore Eq. (H-84) is used for heat flow in one direction. In order to use Eq. (H-88) as an approximation to the physical problem, we subdivide the total thickness into an integral number of spaces.

If we use  $\Delta x = 0.25$ , this gives eight subdivisions. From Eq. (H-88),  $\Delta t$  is fixed by the relation

$$\frac{k\Delta t}{\rho c(\Delta x)^2} = \frac{1}{2}$$

The finite difference form of Eq. (H-93) is solved on a worksheet by replacing the derivatives with central differences centered on grid point  $(i, j)$ .

$$\frac{T_{i+1,j} - 2T_{i,j} + T_{i-1,j}}{\Delta x^2} + \frac{T_{i,j+1} - 2T_{i,j} + T_{i,j-1}}{\Delta y^2} = \frac{Q}{k} \quad (\text{H-94})$$

where  $T_{i,j}$  is the temperature at the  $i$ th location in the  $x$ -direction and the  $j$ th location in the  $y$ -direction. Assuming that the grid spacing  $\Delta x = \Delta y$ , Eq. (H-94) is rearranged as

$$T_{i,j} = \frac{T_{i+1,j} + T_{i-1,j} + T_{i,j+1} + T_{i,j-1} - \Delta x^2 \frac{Q}{k}}{4} \quad (\text{H-95})$$

The spreadsheet is then arranged as shown below:

A4 - $\Delta x^2 Q/k$		C4 - $T_{i,j-1}$	
	B5 - $T_{i-1,j}$	C5 - $T_{i,j}$	D5 - $T_{i+1,j}$
		C6 - $T_{i,j+1}$	

When the value of  $\Delta x^2 Q/k$  is placed in cell A4, the equation for cell C5 is:

$$= (D5 + B5 + C6 + C4)/4 - \$A\$4 \quad (\text{H-96})$$

In the worksheet, the boundaries of the problem are set with the fixed or derivative values of the boundary conditions, and the interior points are set with Eq. (H-96). Then copy this for every internal grid point, set the boundary equations, and turn on the iteration feature to obtain the solution. This is further carried out again with a finer mesh to assess the accuracy. If the heat of generation term depends upon temperature, it is easy to include that complication just by inserting the formula in place of  $\$A\$4$ .

$$\Delta t = \frac{(0.11)(7.8)(0.25)^2}{(2)(0.13)} = 0.206 \text{ s}$$

The boundary conditions are

$$u(0, t) = 0 \quad u(2, t) = 0$$

The initial conditions are

$$\begin{aligned} u(x, 0) &= 100x & \text{for } 0 \leq x \leq 1 \\ u(x, 0) &= 100(2-x) & \text{for } 1 \leq x \leq 2 \end{aligned}$$

The computer program PROG9 uses the simple algorithm of Eq. (H-88), which indicates that at each interior point the temperature at any point at the end of a time step is just the arithmetic average of the temperatures at the adjacent points at the beginning of that time step. The end temperatures are given by the boundary conditions. Because the temperatures are symmetrical on either side of the center line, we calculate only for  $x \leq 1.0$ . The temperature at  $x = 1.25$  is the same as at  $x = 0.75$ . Table H-3 gives the results of the program.

(continued)

**EXAMPLE H-4—(continued)**

**TABLE H-3** Potential Values in One Dimension by Explicit Method

**TABLE H-3—(continued)**

VALUES AT T= 2.878						
0.000	7.048	12.820	16.717	19.083	21.153	
24.997	33.351	48.741	71.808	100.000		
VALUES AT T= 3.166						
0.000	6.452	11.942	16.007	18.987	22.081	
27.239	36.751	52.380	74.210	100.000		
VALUES AT T= 3.454						
0.000	6.452	11.942	16.007	18.987	22.081	
27.239	36.751	52.380	74.210	100.000		
VALUES AT T= 3.742						
0.000	6.002	11.280	15.518	19.091	23.134	
29.383	39.705	55.335	76.083	100.000		
VALUES AT T= 4.030						
0.000	6.002	11.280	15.518	19.091	23.134	
29.383	39.705	55.335	76.083	100.000		
VALUES AT T= 4.317						
0.000	5.666	10.803	15.233	19.364	24.245	
31.380	42.270	57.784	77.591	100.000		
VALUES AT T= 4.605						
0.000	5.666	10.803	15.233	19.364	24.245	
31.380	42.270	57.784	77.591	100.000		
VALUES AT T= 4.893						
0.000	5.423	10.486	15.124	19.767	25.372	
33.218	44.507	59.845	78.835	100.000		
VALUES AT T= 5.181						
0.000	5.423	10.486	15.124	19.767	25.372	
33.218	44.507	59.845	78.835	100.000		
VALUES AT T= 5.469						
0.000	5.262	10.304	15.159	20.268	26.488	
34.902	46.469	61.604	79.879	100.000		
VALUES AT T= 5.757						
0.000	5.262	10.304	15.159	20.268	26.488	
34.902	46.469	61.604	79.879	100.000		
VALUES AT T= 6.044						
0.000	5.168	10.236	15.312	20.838	27.577	
36.445	48.200	63.120	80.767	100.000		
VALUES AT T= 6.332						
0.000	5.168	10.236	15.312	20.838	27.577	
36.445	48.200	63.120	80.767	100.000		
VALUES AT T= 6.620						
0.000	5.131	10.261	15.557	21.454	28.632	
37.858	49.738	64.440	81.532	100.000		
VALUES AT T= 6.908						
0.000	5.131	10.261	15.557	21.454	28.632	
37.858	49.738	64.440	81.532	100.000		
VALUES AT T= 7.196						
0.000	5.141	10.361	15.873	22.100	29.646	
39.158	51.111	65.599	82.197	100.000		
VALUES AT T= 7.484						
0.000	5.141	10.361	15.873	22.100	29.646	
39.158	51.111	65.599	82.197	100.000		
VALUES AT T= 7.771						
0.000	5.189	10.520	16.242	22.762	30.618	
40.355	52.346	66.624	82.781	100.000		
VALUES AT T= 8.059						
0.000	5.189	10.520	16.242	22.762	30.618	
40.355	52.346	66.624	82.781	100.000		
VALUES AT T= 8.347						
0.000	5.267	10.725	16.649	23.430	31.548	
41.461	53.462	67.538	83.297	100.000		
VALUES AT T= 8.635						
0.000	5.267	10.725	16.649	23.430	31.548	
41.461	53.462	67.538	83.297	100.000		

*(continued)*

*(continued)*

**EXAMPLE H-4—(continued)****TABLE H-3—(continued)**

VALUES AT T= 8.923						
0.000	5.368	10.965	17.083	24.097	32.435	
42.486	54.476	68.358	83.756	100.000		
VALUES AT T= 9.211						
0.000	5.368	10.965	17.083	24.097	32.435	
42.486	54.476	68.358	83.756	100.000		
VALUES AT T= 9.498						
0.000	5.486	11.231	17.534	24.757	33.281	
43.438	55.401	69.097	84.168	100.000		
VALUES AT T= 9.786						
0.000	5.486	11.231	17.534	24.757	33.281	
43.438	55.401	69.097	84.168	100.000		
VALUES AT T= 10.074						
0.000	5.618	11.514	17.995	25.405	34.088	
44.326	56.250	69.769	84.539	100.000		
VALUES AT T= 10.362						
0.000	5.618	11.514	17.995	25.405	34.088	
44.326	56.250	69.769	84.539	100.000		
VALUES AT T= 10.650						
0.000	5.759	11.809	18.459	26.037	34.856	
45.155	57.031	70.381	84.876	100.000		
VALUES AT T= 10.938						
0.000	5.759	11.809	18.459	26.037	34.856	
45.155	57.031	70.381	84.876	100.000		
VALUES AT T= 11.225						
0.000	5.906	12.110	18.922	26.653	35.587	
45.931	57.754	70.942	85.184	100.000		
VALUES AT T= 11.513						
0.000	5.906	12.110	18.922	26.653	35.587	
45.931	57.754	70.942	85.184	100.000		
VALUES AT T= 11.801						
0.000	6.056	12.414	19.380	27.250	36.283	
46.659	58.424	71.458	85.465	100.000		
VALUES AT T= 12.089						
0.000	6.056	12.414	19.380	27.250	36.283	
46.659	58.424	71.458	85.465	100.000		
VALUES AT T= 12.377						
0.000	6.207	12.717	19.830	27.827	36.946	
47.343	59.048	71.935	85.724	100.000		
VALUES AT T= 12.665						
0.000	6.207	12.717	19.830	27.827	36.946	
47.343	59.048	71.935	85.724	100.000		
VALUES AT T= 12.952						
0.000	6.358	13.017	20.269	28.383	37.577	
47.987	59.629	72.378	85.963	100.000		
VALUES AT T= 13.240						
0.000	6.358	13.017	20.269	28.383	37.577	
47.987	59.629	72.378	85.963	100.000		
VALUES AT T= 13.528						
0.000	6.508	13.312	20.697	28.918	38.178	
48.594	60.174	72.789	86.185	100.000		
VALUES AT T= 13.816						
0.000	6.508	13.312	20.697	28.918	38.178	
48.594	60.174	72.789	86.185	100.000		
VALUES AT T= 14.104						
0.000	6.655	13.601	21.112	29.432	38.749	
49.167	60.684	73.173	86.391	100.000		

(continued)

**TABLE H-3—(continued)**

VALUES AT T= 14.392						
0.000	6.655	13.601	21.112	29.432	38.749	
49.167	60.684	73.173	86.391	100.000		
VALUES AT T= 14.967						
0.000	6.800	13.882	21.513	29.925	39.293	
49.709	61.163	73.531	86.583	100.000		
VALUES AT T= 15.255						
0.000	6.940	14.154	21.900	30.398	39.811	
50.221	61.613	73.867	86.763	100.000		
VALUES AT T= 15.543						
0.000	6.940	14.154	21.900	30.398	39.811	
50.221	61.613	73.867	86.763	100.000		
VALUES AT T= 15.831						
0.000	7.076	14.418	22.272	30.850	40.303	
50.705	62.038	74.183	86.931	100.000		
VALUES AT T= 16.118						
0.000	7.076	14.418	22.272	30.850	40.303	
50.705	62.038	74.183	86.931	100.000		
VALUES AT T= 16.406						
0.000	7.208	14.672	22.630	31.283	40.772	
51.164	62.438	74.480	87.089	100.000		
VALUES AT T= 16.694						
0.000	7.208	14.672	22.630	31.283	40.772	
51.164	62.438	74.480	87.089	100.000		
VALUES AT T= 16.982						
0.000	7.335	14.917	22.974	31.696	41.218	
51.599	62.817	74.760	87.238	100.000		
VALUES AT T= 17.270						
0.000	7.335	14.917	22.974	31.696	41.218	
51.599	62.817	74.760	87.238	100.000		
VALUES AT T= 17.558						
0.000	7.457	15.152	23.303	32.091	41.643	
52.012	63.175	75.023	87.378	100.000		
VALUES AT T= 17.845						
0.000	7.457	15.152	23.303	32.091	41.643	
52.012	63.175	75.023	87.378	100.000		
VALUES AT T= 18.133						
0.000	7.575	15.378	23.618	32.468	42.047	
52.403	63.513	75.273	87.510	100.000		
VALUES AT T= 18.421						
0.000	7.575	15.378	23.618	32.468	42.047	
52.403	63.513	75.273	87.510	100.000		
VALUES AT T= 18.709						
0.000	7.688	15.594	23.920	32.828	42.431	
52.775	63.834	75.508	87.635	100.000		
VALUES AT T= 18.997						
0.000	7.688	15.594	23.920	32.828	42.431	
52.775	63.834	75.508	87.635	100.000		
VALUES AT T= 19.285						
0.000	7.796	15.801	24.208	33.171	42.797	
53.128	64.138	75.731	87.752	100.000		
VALUES AT T= 19.572						
0.000	7.796	15.801	24.208	33.171	42.797	
53.128	64.138	75.731	87.752	100.000		
VALUES AT T= 19.860						
0.000	7.900	16.000	24.483	33.499	43.145	
53.463	64.426	75.942	87.864	100.000		
VALUES AT T= 20.148						
0.000	7.900	16.000	24.483	33.499	43.145	
53.463	64.426	75.942	87.864	100.000		

**FURTHER READING**

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2. Coker, A. K., *Hydroc. Proc.*, Dec 2004, pp. 77–85.
3. Coker, A. K., *Modeling of Chemical Kinetics and Reactor Design*, Gulf Publishing Co., Houston, TX, 2001.
4. Gerald, C. F. and P.O. Wheatley, *Applied Numerical Analysis*, 4th ed., Addison-Wesley Publishing Co., 1989.
5. Orvis, W. J., *Excel for Scientists and Engineers*, 2nd Ed., Sybex, 1993.
6. Finlayson, B. A., *Introduction to Chemical Engineering Computing*, John Wiley & Sons, Inc., 2006.

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## Appendix I

# SCREENSHOT GUIDE TO ABSOFT COMPILER GRAPHICAL USER INTERFACE

The graphical interface to the Absoft compiler can be started from Windows by selecting Programs from the Start menu and then compile from the Absoft Tools submenu. A command line version of an individual compiler can be started with one of the commands f90, f77, or cc. Both methods allow the user to compile source files quickly and easily. The Absoft compilers can be run either from a command line or from the Absoft Developer Tools Interface.

*Note:* It is assumed that the compiler has been installed on the C: drive. If this is not the case, substitute the correct drive letter in the examples as appropriate.

Selecting **Programs** from the **Start** menu and then **Compiler Interface** from the Absoft submenu opens the Absoft Developer Tools Interface. It can also be started by selecting **Run** from the **Start** menu and typing c:\Absoft\bin\Atools or by typing Atools at a console or command line window. The Windows NT

**Command Prompt** or Windows XP **MS-DOS Prompt** should be opened by selecting **Programs** from the **Start** menu and then **Development Command Shell** from the **Absoft** submenu. Opening the console window in this way automatically sets the environment variables by executing the c:\Absoft\bin\absvar.bat batch file.

Error may occur when compiling a program in the source file syntax or with the compiler options selected. All errors and warning messages are shown in the compiler output window in a format similar to the following:

```
error on line 38 of PROG1.FOR: spelling error?
```

The following steps as illustrated in Figures I-1 – I-12 are used to compile and execute the Fortran programs in this book using the Absoft compiler.

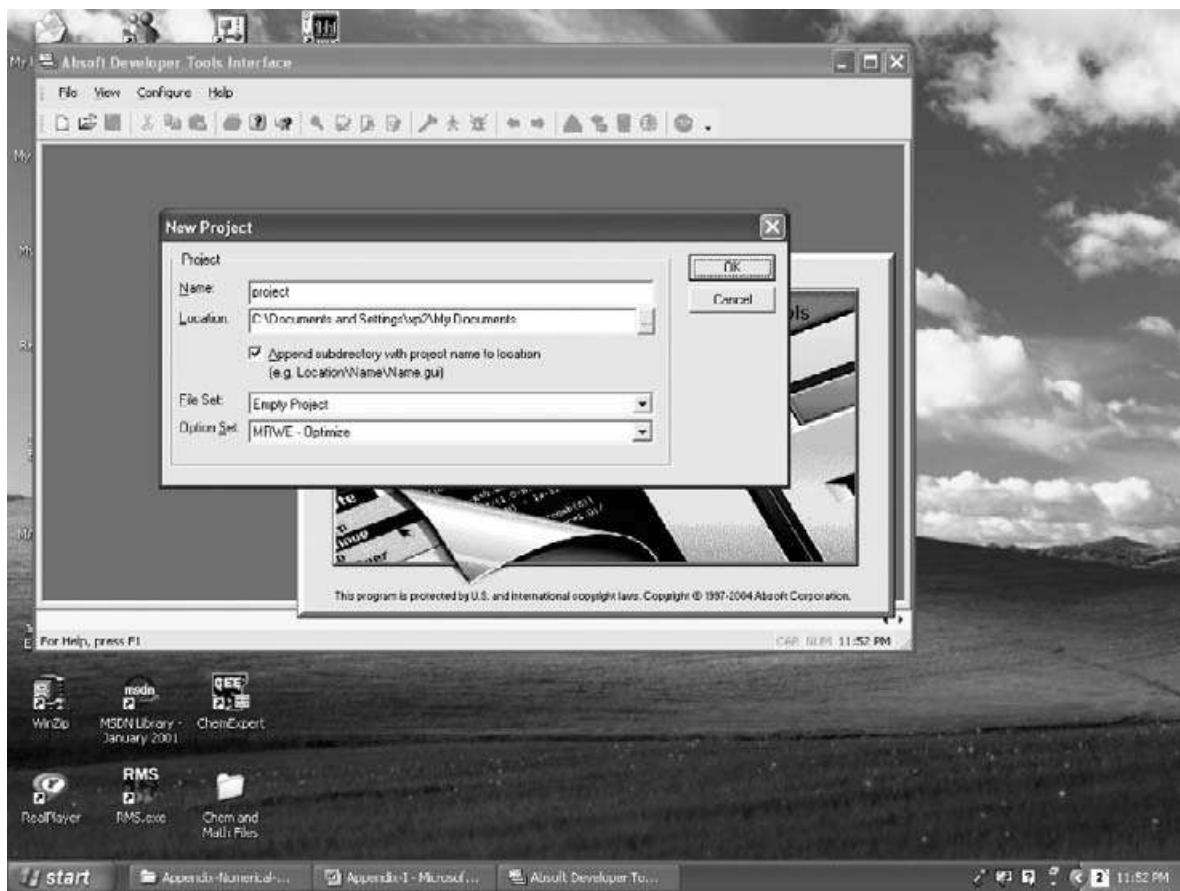
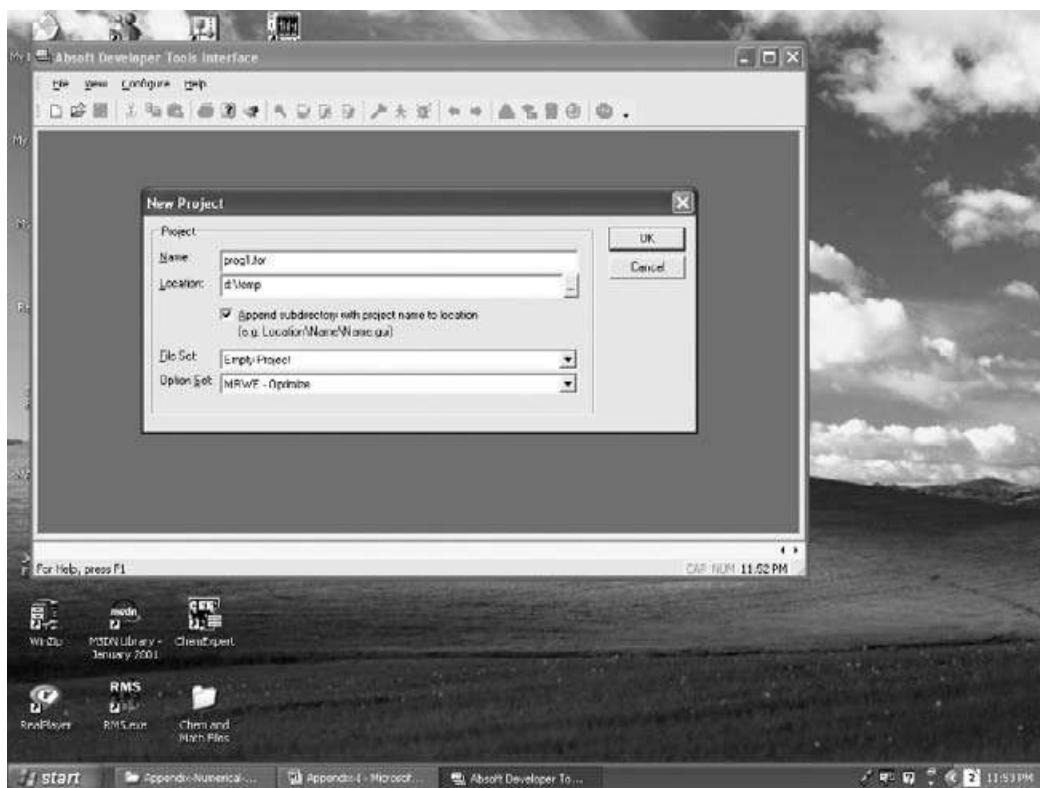


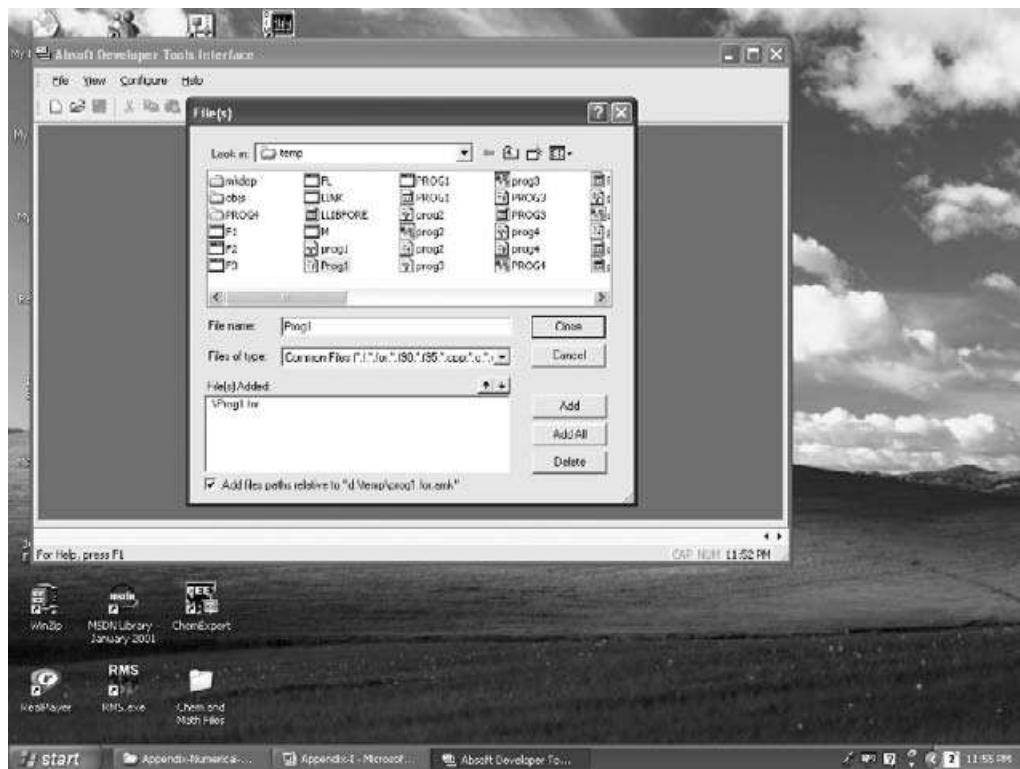
Figure I-1 Absoft compiler window.



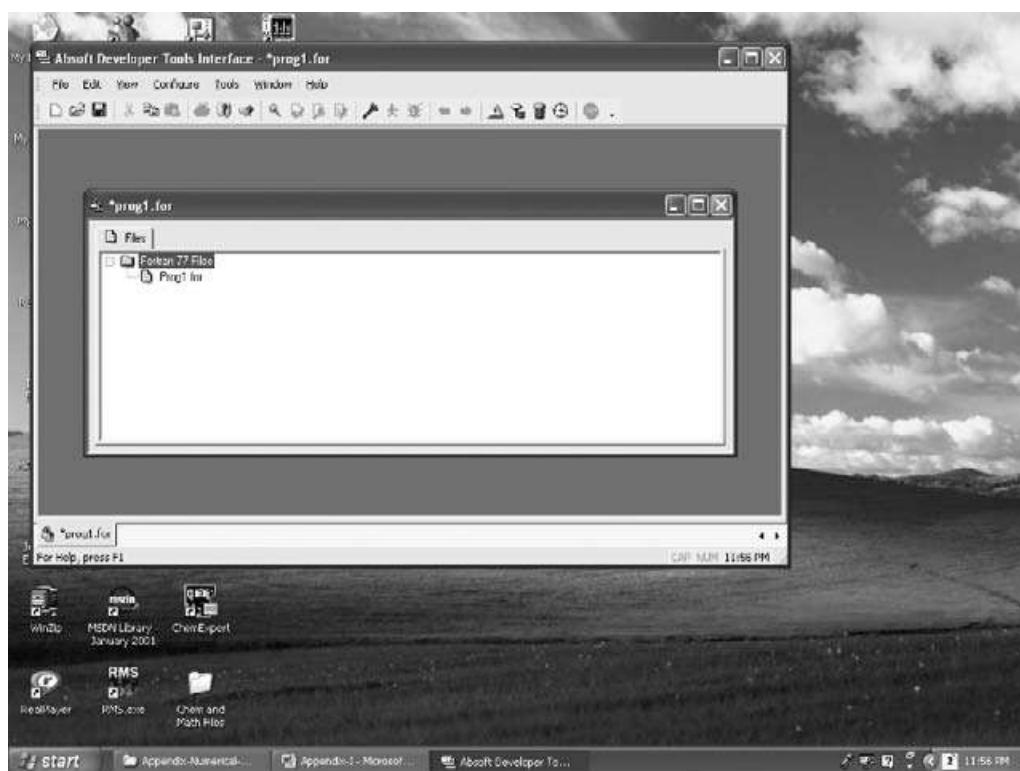
**Figure I-2** New Project window from File submenu.



**Figure I-3** Absoft Fortran 77 compiler dialog box.



**Figure I-4** Source file from Configure submenu.



**Figure I-5** Source file (e.g. PROG1) is selected and added to the File(s) Added box.

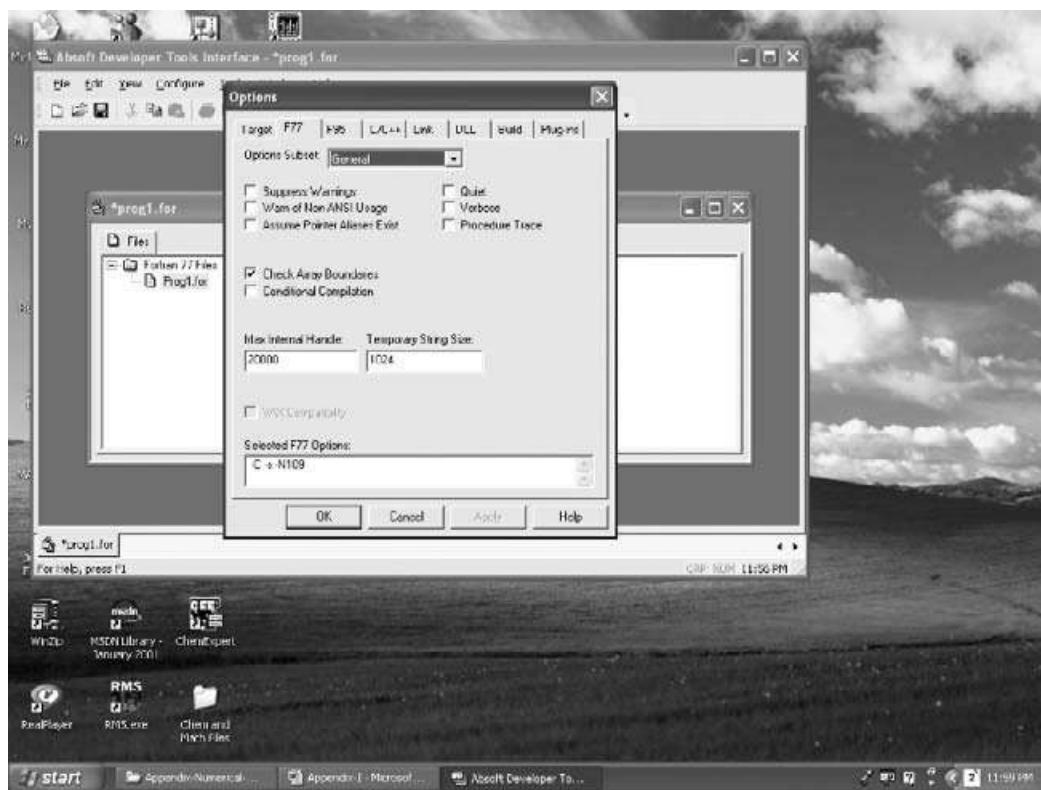


Figure I-6 f77 General Options.

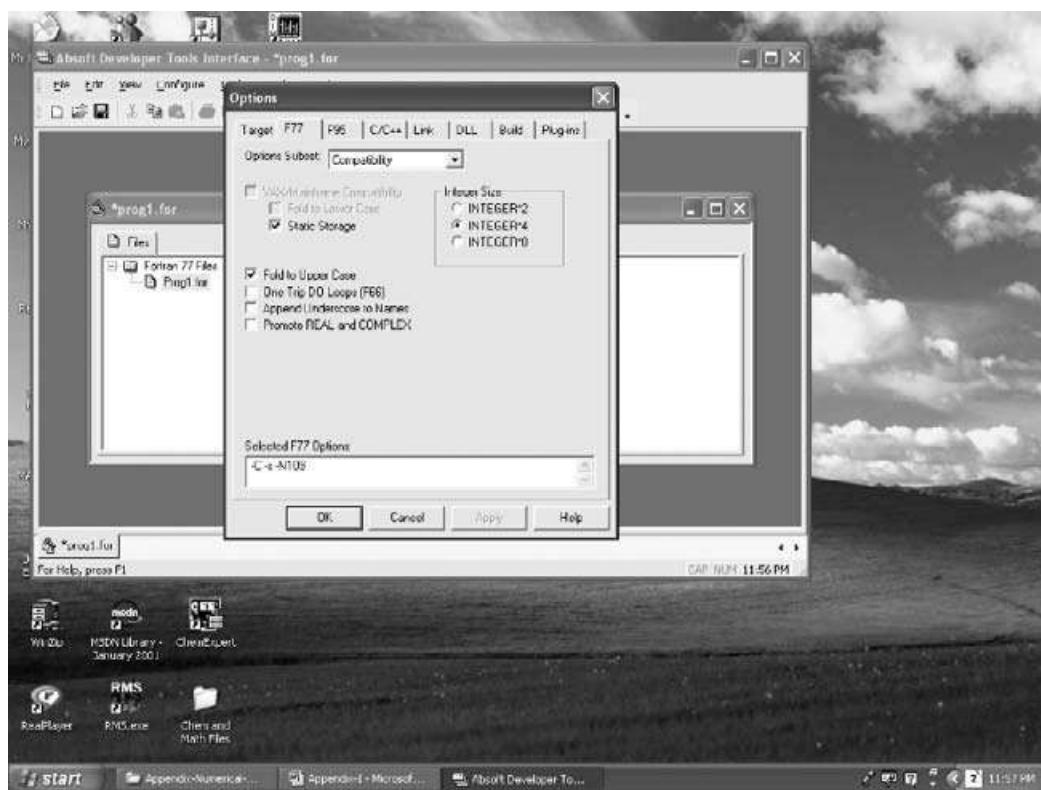


Figure I-7 f77 Compatibility Options.

The screenshot shows the Absoft Editor interface with three windows:

- Top Window:** Title bar says "Absoft Editor - d:\Temp\Prog1.for". Menus: File, Edit, View, Tools, Window, Help. Toolbar with various icons. Status bar at the bottom.
- Middle Window:** Title bar says "d:\Temp\Prog1.for". Subtitle bar says "PROGRAM PROG1". Text area contains:
 

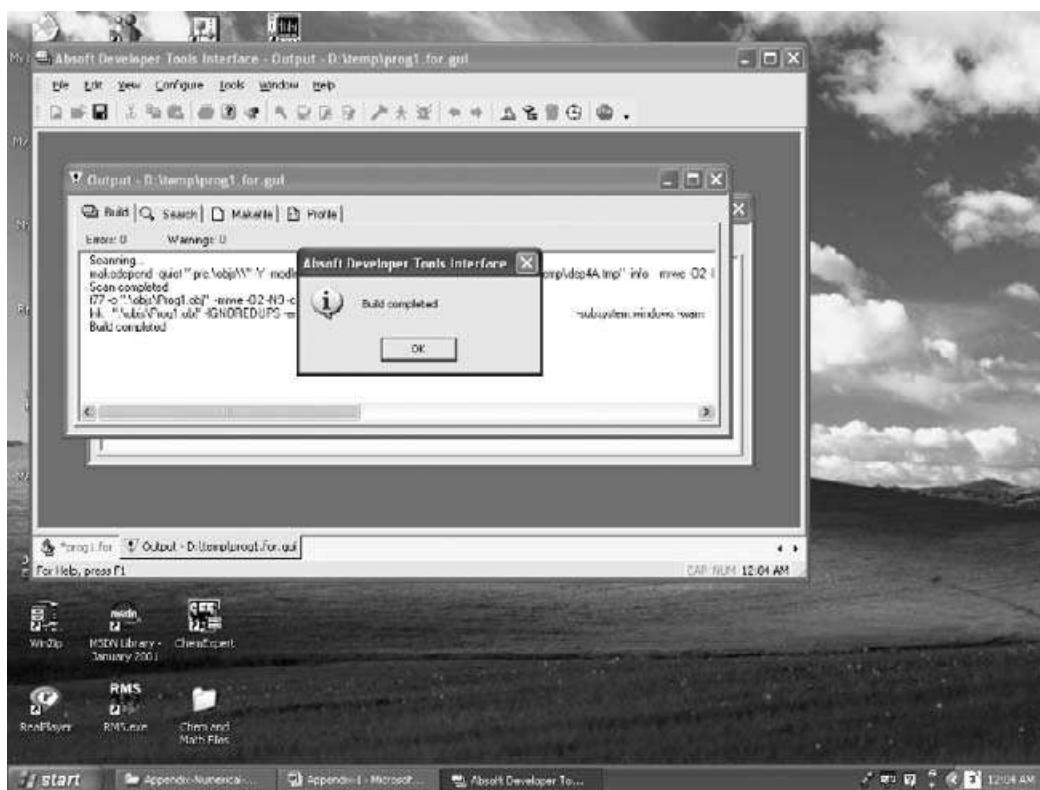
```

C IN A.K. CORER
C
C BUTTERDOETH-BEINHARH 2001
C GULF PROFESSIONAL PUBLISHING
C WOBURN, MA 01881-2641
C
C **** THE PROGRAM DETERMINES A RELATIONSHIP BETWEEN TWO VARIABLES ****
C * i.e., ONE DEPENDENT AND ONE INDEPENDENT VARIABLE. THE PROGRAM *
C * SELECTS THE CURVE OF BEST FIT BY TESTING THE DATA AGAINST *
C * EQUATIONS OF THE FORM: Y=a+b X(x) *
C * THE PROGRAM GIVES THE CONSTANTS FOR THE EQUATION AND THE *
C * CORRELATION COEFFICIENT *
C ****
C THIS PROGRAM WILL CORRELATE X-Y DATA FOR THE FOLLOWING EQUATIONS
C
C 1. Y=a+b X
C 2. Y=a+b X^2
C 3. Y=a+b X^0.5
C 4. Y=a+exp(b X)
C 5. Y=a+b ln(X)
C 6. Y=a X^b
C 7. Y=a+b (1/X)
C 8. Y=a+b (1/X)^b
C 9. Y=a+b X^3
C
C X = VECTOR OF INDEPENDENT VARIABLE
C Y = VECTOR OF DEPENDENT VARIABLE
C YCAT = VECTOR OF THE ESTIMATED DEPENDENT VARIABLE
C N = NUMBER OF DATA
C A, B = CONSTANTS FOR THE EQUATION
C COR = CORRELATION COEFFICIENT
      
```
- Bottom Window:** Title bar says "d:\Temp\Prog1.for". Menus: Start, Appendix-I-Numerical..., Appendix-I-Physical..., Absoft Developer Tools..., Absoft Editor - d:\Temp\Prog1...". Status bar at the bottom.

Figure I-8 Source code Window from Absoft Editor.



Figure I-9 Compiled source code from Tools submenu.



**Figure I-10** Errors and warning messages shown in the compiler output window.



**Figure I-11** Execute the compiled source file from Tools submenu.

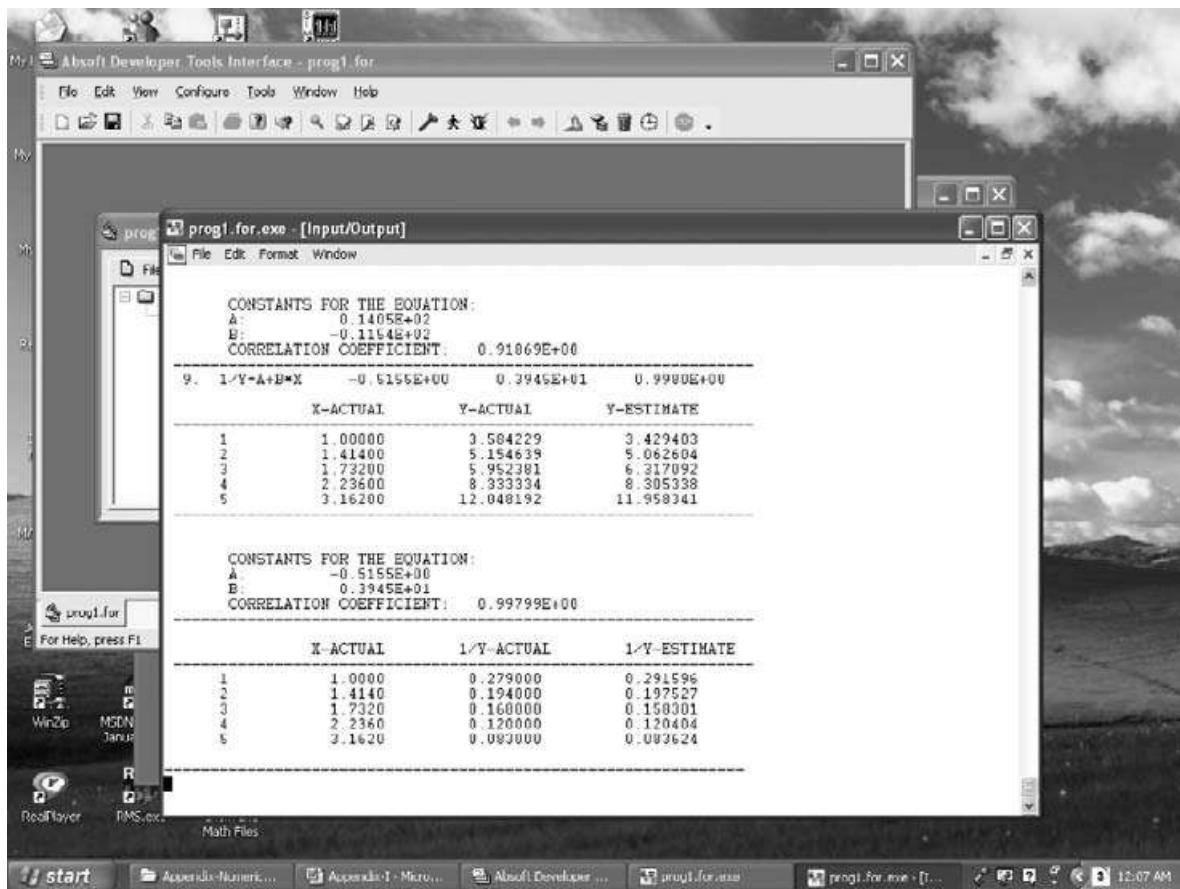


Figure I-12 Results of the compiled program.

What to do	How to perform task
1. Start up the interface to the compiler: Invoke the Absoft Developer Tools interface.	Select Programs from the Start menu, then select Compiler Interface from the Absoft Pro Fortran submenu. A windows appears entitled New Project windows, select a program file name (e.g. PROG1).
2. Set the options to create an MRWE application (a Windows program).	In the target type box, click on the MRWE application button; in the Options box, click the MRWE Events box. Then click on OK box.
3. Select the source file (e.g. PROG1.For) to compile	Click on Configure menu and select Add/Remove File(s) from the pull-down menu. Select the application source file from its folder (e.g. c:\absoft60), then add it to the Source File(s) Added box by clicking on the Add button. Click on the close box.
4. Select F77 compiler options	Click on Configure menu and select Set Options from the pull-down menu.
The options fall into five categories: General, Control, Compatibility, Miscellaneous, and Format.	Using General options subset, click on check Array Boundaries. Using Compatibility subset, click on Static storage and Fold to Upper Case boxes. Click on OK.
5. Invoke the Absoft compilers with all selected options and source files	Click on the Tools menu and Select Build from the pull-down menu. The Absoft compiler successfully compiles the source file and gives a message: "Build completed".

- 
- |                                                                                         |                                                                                                                                                                                                                                                                                                                                               |
|-----------------------------------------------------------------------------------------|-----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| 6. Execute the compiled application.                                                    | Click on the Tools menu and select Execute from the pull-down menu.                                                                                                                                                                                                                                                                           |
| 7. View/Print the result from Notepad, if the result file is created in the application | Click on Start menu, select programs, Accessories, Notepad in sequence. Click on file, select open from the pull-down menu. Select the absoft60 folder from the C directory. Select All files (*.*) from the File of type dialogue box. Double click on the result file (e.g. RES1). Click on Print menu to obtain a hard copy of the result. |
- 

The source codes and executable files of the various programs in the text are on the CD-ROM.

The Absoft Compiler can be purchased from

Absoft Corporation ([www.absoft.com](http://www.absoft.com))

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