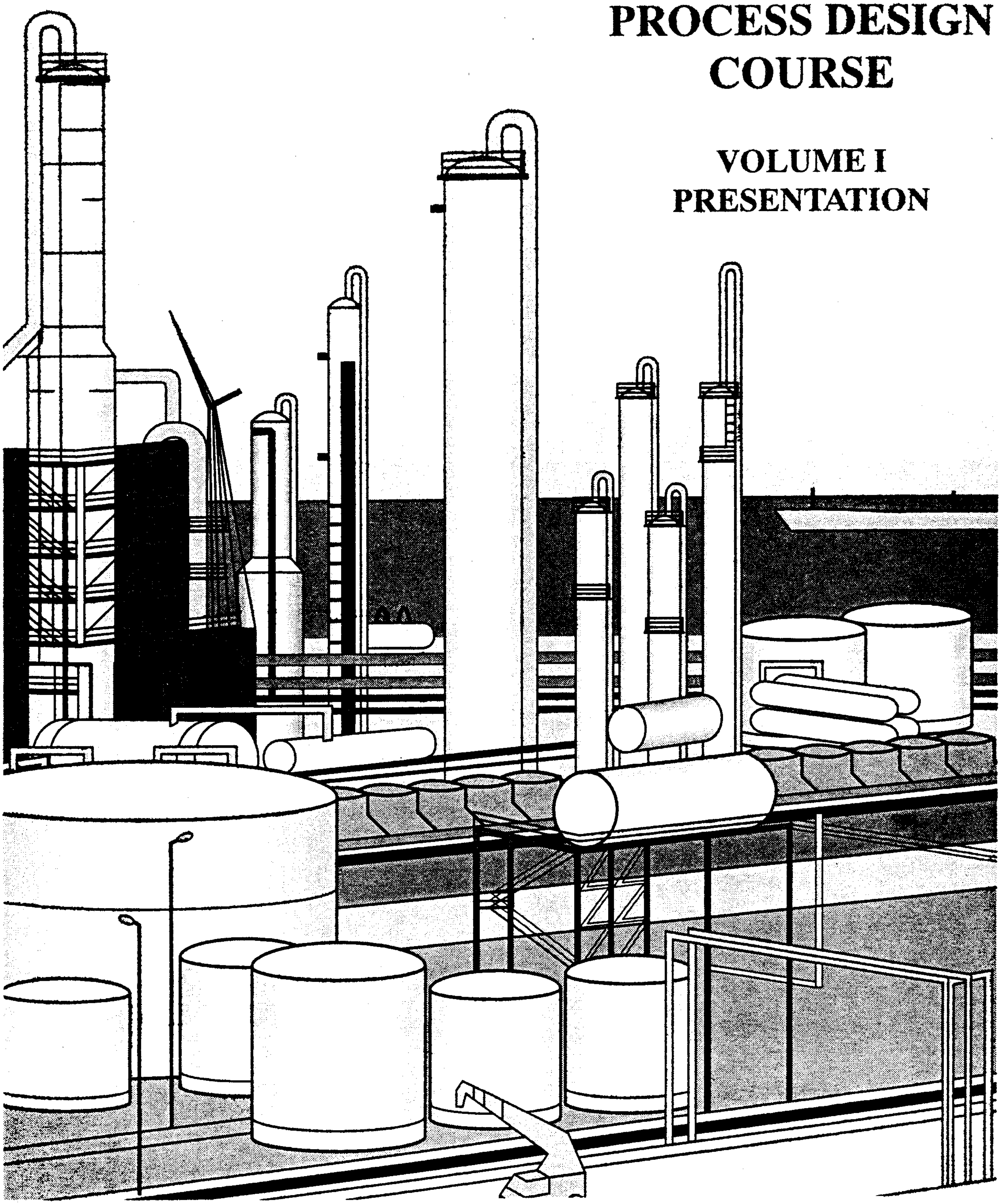
PROCESS DESIGN COURSE

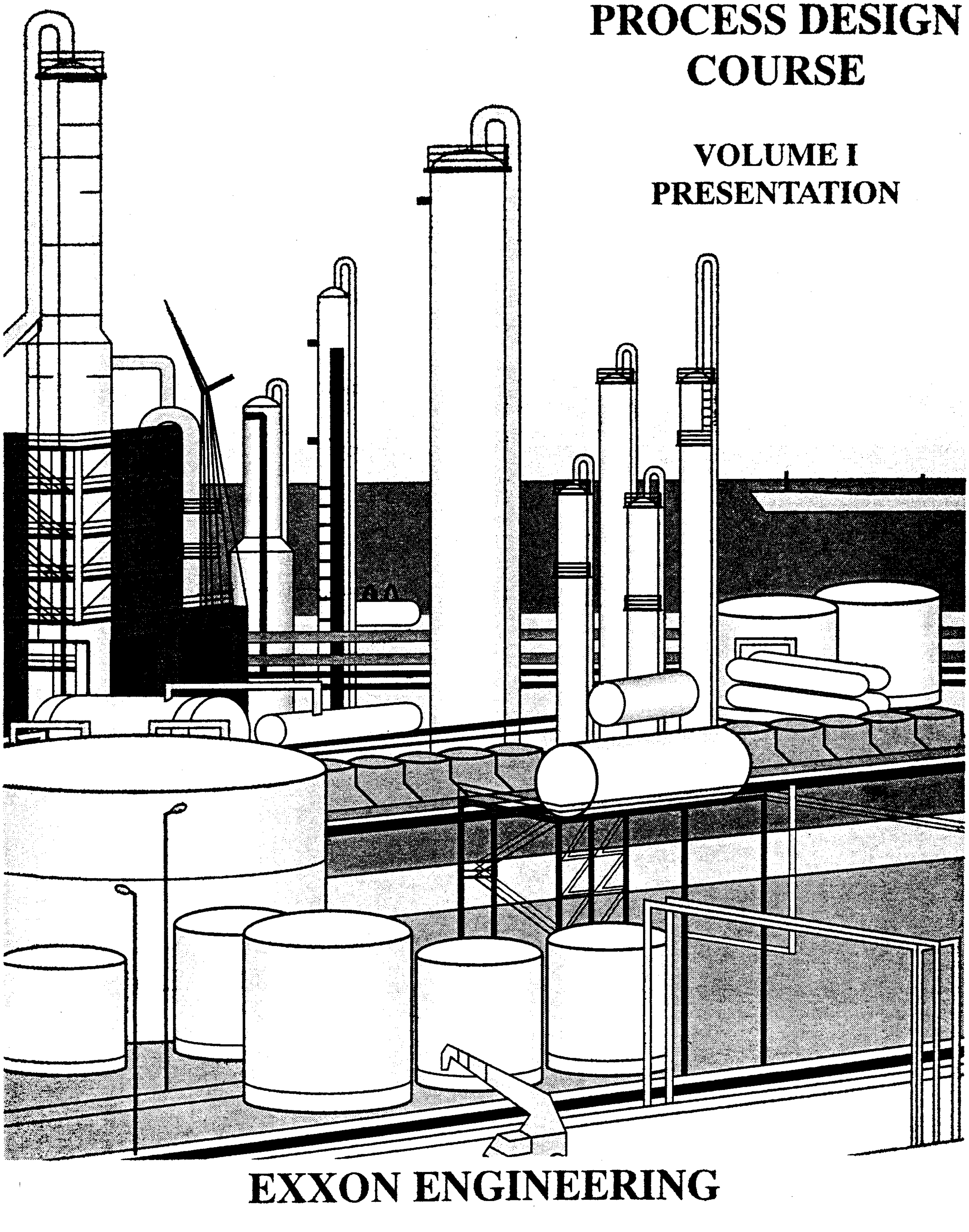
VOLUME I
PRESENTATION



EXXON ENGINEERING

PROCESS DESIGN COURSE

VOLUME I
PRESENTATION



EXXON ENGINEERING

TRAINING PERFORMANCE OBJECTIVES

EXXON ENGINEERING PROCESS DESIGN COURSE

I. DATA SOURCES

1. Given:

- A gas stream of three defined components
- The mol fraction of each component
- Temperature and pressure of gas stream

Determine the following properties of the stream:

a - Molecular weight
b - Weight and molal average boiling points
c - Pseudocritical temperature
d - Pseudocritical pressure
e - Absolute viscosity
f - Compressibility factor
g - Density in pounds/ft^3 at conditions.

2. Given:

- A liquid stream of three defined components
- Volume fraction of each component
- Temperature and pressure of liquid stream

Determine the following properties of the stream:

a - Molecular weight
b - Mean, molal, volume, and weight average boiling points
c - Characterization factor
d - True critical temperature
e - True critical pressure
f - Density in pounds/ft^3 at 60°F
g - Density in °API at 60°F

3. Given:

- A liquid stream of three defined components
- The weight fraction of each component
- Viscosity of each component at 100°F and 210°F

Determine:
a - The viscosity of the mixture at 100°F and 210°F via the Blending Index procedure.
b - The viscosity of the mixture at 150°F using viscosity-temperature charts.
c - The difference in enthalpy between the stream as a liquid at 100°F and 30 atmospheres and the stream as a vapor at 600°F and 1 atmosphere.
d - The latent heat of vaporization at 600°F.

4. Given:

- A gas stream of three defined components
- The mol fraction of each component
- Temperature and pressure of the gas stream
- Flow rate of stream in MegaSCF/SD

Determine:

a - The brake horsepower required to compress the stream adiabatically to twice the given pressure
b - Temperature of the compressed stream

5. Given

- A stream of three defined components
- The mol fraction of each component
- Pressure of the stream

Estimate the following properties of the stream using Vapor Pressures:

a - Bubble point temperature
b - Dew point temperature

6. Given:

- A stream of three defined components
- The mol fraction of each component
- The equilibrium instants for each component
- Temperature and pressure of the stream

Determine the following properties of the stream:

a - Mole percent liquid at given conditions
b - Liquid and vapor phase compositions

7. Given:

- ASTM Distillation of pipestill gas oil cut
- API gravity

Determine:

a - Specific heat of stream at 200°F, 300°F, and 400°F (correct for K_W)
b - Latent heat of vaporization of stream at 200°F, 300°F, and 400°F
c - 15/5 Distillation Curve
d - The "names" of an arbitrary number of pseudocomponents that will define the stream

II. DISTILLATION TECHNOLOGY AND TOWER DESIGN

1. Given:

   - Temperature, pressure, and rate of C_3-430°F refinery stream
   - Composition of the stream through C_5 material
   - Product specifications for C_4 in the bottoms and C_5 in the overhead
   - ASTM distillation of the C_5+ material
   - Conditions of utilities that are available

   Be able to

   - Determine a quickie preliminary design using Fenske-Gilliland procedure

2. Be able to define the following tray design terms:

a - Blowing	g - Single Pass Tray	m - Downcomer Clearance
b - Dumping	h - Bubble Area	n - Two Pass Tray
c - Jet Flooding	i - Hole Area	o - Antijump Baffle
d - Liquid Gradient	j - Perforated Area	p - Downcomer Seal
e - Weeping	k - Waste Area	q - Downcomer Filling
f - Ultimate Capacity	l - Free Area	

3. Be able to classify the following types of trays:

   - Bubble Cap
   - Sieve
   - Jet
   - Valve

   In order of increasing:

   a - Capacity
   b - Cost per Unit Area
   c - Fractionation Efficiency
   d - Flexibility

4. Given:

   - A liquid stream's--
     - Flow rate (pounds/hour) and composition
     - Surface tension
     - Viscosity
     - Temperature
   - A vapor stream's --
     - Flow rate (pounds/hour) and composition
     - Viscosity
     - Temperature
     - Pressure
   - Required flexibility of the service

   Be able to design a sieve tray by hand for this service

III. HEAT TRANSFER FUNDAMENTALS

1. Given:

   - Shell and tube exchangers for concurrent, countercurrent, and mixed flow

   Be able to:

   a - Identify each
   b - Explain the differences among them

2. Given:

   - Temperature in and out for both sides of S & T exchanger
   - Number of tube and shell side passes
   - No change in phase for either fluids

   Be able to:

   a - Calculate the ΔT_{LM}
   b - Calculate F_n
   c - State the temperature approach
   d - State the temperature cross
   e - Explain why one should not design an exchanger that results in outlet temperatures giving F_n <0.8

3. Given:

   - Temperatures in and out for both sides of a water condenser
   - T/Q curve for the HC being condensed

   Be able to:

   - Calculate the effective temperature driving force (Δte)

4. Given:

   - Temperatures in and out for both sides of S & T water cooler
   - Rate of HC on the shell side
   - API gravity and MABP of the HC
   - Exchanger geometry (tube sizes, area, etc.)

   Be able to:

   a - Calculate the individual film coefficients and the overall coefficient
   b - Calculate the total resistance due to fouling
   c - Calculate tube wall temperature on the water side
   d - Specify which, if either, film coefficient is controlling
   e - Estimate the effect on the overall coefficient if the rate of HC is doubled

IV. HEAT EXCHANGER DESIGN CONSIDERATIONS

1. Given:

   - An exchanger design with less than the maximum allowable ΔP on both sides

   Be able to:

   a - Explain why low pressure drop is undesirable
   b - List steps that could be taken to increase ΔP

2. Be able to list two advantages for double-pipe exchangers.

3. Be able to list the factors that enter into the choice between air-cooled and water-cooled exchangers.

4. Given:

   - The total duty of a S & T exchanger (MBtu/h)
   - The characteristics at the two fluids in the exchanger
   - The temperatures and pressures of both fluids
   - Type of service
   - Tube side and shell side fouling factors

   Be able to:

   a - Determine the minimum number of shells and explain reason for this minimum
   b - Determine which fluid should be on the tube side
   c - Specify whether a fixed tube sheet, U-tube, or pull-through floating head design should be used
   d - Specify the tube layout
   e - Specify tube type (bare or low fin)

5. Given:

   - The task of designing a S & T exchanger with specified shell side pressure drop

   Be able to:

   a - Specify minimum and maximum baffle spacing
   b - Explain reason for maximum limitation
   c - Specify the most desirable baffle orientation if the exchanger has condensing vapor on shell side
   d - List advantages and disadvantages for high pressure drop through the shell side

6. Given:

- Shell side hydrocarbon rate (no change of phase)
- Shell side fluid properties
- Temperatures and pressures
- Allowable pressure drop
- Water side inlet temperature
- Water side allowable ΔP

Be able to:

- <u>Estimate</u> the area of the exchanger

V. FURNACES

1. Given

   - The design of a furnace from a furnace supplier
   - The process information and requirements for the furnace
   - Type of fuel and percent excess air

   Be able to:

   a - Determine radiant heat flux
   b - Calculate heat transferred in the radiant section
   c - Calculate bridgewall temperature
   d - Calculate heat transferred in the convection section
   e - Calculate the stack temperature
   f - Calculate maximum tube metal temperature in radiant section
   g - Calculate fin-tip temperature in the convection section
   h - Calculate film temperatures

2. Given

   - Fuel gas type and rate to a furnace
   - Stack temperature
   - Flue gas analysis
   - Ambient air temperature

   Be able to:

   a - Calculate the percent excess air
   b - Calculate the furnace efficiency

3. The following information is given for a 100 M Btu/hour (heat to oil) furnace:

   - 75% of heat load is obtained with 15° API fuel oil
   - 25% of heat load is obtained with 1000 Btu/scf fuel gas
   - Oil burners operate at 25% excess air
   - Gas burners operate at 15% excess air
   - 0.6 pounds of atomizing steam (150 psig saturated) is required per pound of fuel oil fired
   - Stack temperature is 60°F

   Determine:

   a - Furnace efficiency
   b - Pounds per hour of flue gas

VI. FLUID FLOW, PUMPS, COMPRESSORS

1. Given

   - A piping system handling a liquid hydrocarbon stream
   - Stream flowrate and properties
   - Length and size of pipe
   - Number and type of fittings

   Be able to:

   a - Estimate equivalent length of pipe
   b - Estimate pressure drop

2. Be able to define the term "cavitation"

3. Given the following data for each of four pumping services:

   - Rate
   - Specific gravity and viscosity
   - NPSH available to the pumps
   - Solids content
   - Differential head required
   - Viscosity

   Be able to select from the following list the type of pump to be used in each service

   a - Centrifugal
   b - Rotary screw
   c - Reciprocating

4. Given the following data for a pumping service:

   - Horizontal centrifugal pump with motor driver
   - Suction drum is an overhead distillate drum
   - Elevation of drum and pump centerline
   - Size, length, and number of fittings in the suction line
   - Rate, specific gravity and viscosity of the liquid being pumped

   Be able to determine the NPSH available to the pump

5. Given the same data as TPO Number 4 plus

   - Pump characteristic, efficiency, and NPSH curves
   - Pump impeller diameter
   - Flow control valve pressure drop
   - Piping system downstream of pump
   - Downstream controlled pressure

   Be able to determine

   a - Brake horsepower required by pump
   b - System resistance curve
   c - Diameter to which impeller should be trimmed to allow for a control valve pressure drop equal to 20% of the line friction pressure drop
   D - Be able to list process information required to specify pump and spare

6. Given, for a centrifugal compressor:

   - Composition and rate of gas stream feeding the compressor
   - Discharge pressure
   - Suction pressure and temperature

   Be able to estimate:

   a - Temperature rise exponent
   b - Discharge temperature
   c - Polytropic compression exponent
   d - Brake horsepower

VII. PROCESS CONTROL

1. For Item under Fluid Flow Section, be able to determine the control valve size.

2. Given

   - A list of services such as feed from tankage to tower, fractionator reflux, etc.

   Be able to:

   a - List the four fundamental types of control
   b - Select the type or types of control which would be most frequently used for each service
   c - Select the control valve action on instrument air failure for each control valve

3. Given

   - An incomplete process design flowplow for a multipass process furnace
   - Properties and operating conditions of process stream
   - Fuel gas pressure
   - Pilot gas source is natural gas (pipeline)
   - Coil outlet temperature control is to be used
   - Furnace pressure drop

   Be able to prepare sketch showing typical instrumentation for controlling furnace operation

4. Given

   - Compressor suction knockout drum with automatic liquid drawoff to blowdown drum

   Be able to:

   - Sketch drum and compressor showing all instrumentation needed for the drum, including alarms and compressor shutdown

5. Be able to:

   a - Define the term rangeability
   b - State the maximum rangeability for a control valve (double seated, globe) and for an orifice meter.
   c - Specify when a rotameter, turbine meter and venturi are used

VII. DRUM DESIGN

Given:

- Existing P&ID for a tower that is to be expanded
- Detailed Heat and Material Balance for the new conditions

Be able to:

a - Calculate the maximum velocity permitted in the vapor-liquid distillate separator drum
b - Determine the volume required in the drum for the liquid
c - Determine the length and diameter of the drum
d - Prepare a detailed process design drawing of the drum suitable for submission to a vendor

IX. GENERAL PROCESS DESIGN CONSIDERATIONS

1. Be able to define the following terms:

   a - Operating temperature
   b - Maximum fluid temperature
   c - Design temperature
   d - Operating pressure
   e - Design pressure
   f - Short time design basis

2. Be able to explain why it is very important to minimize the difference between operating temperature and design temperature for carbon steel operating above 650°F

3. Given

   - The task of designing a depropanizer, a butane splitter and a gasoline rerun tower

   Be able to

   a - List the factors that should be considered in specifying the design pressures and the minimum design temperatures for the vessels.
   b - Explain how the vendor will know what internal pressure the bottom of the towers will have to withstand

4. Given

   - Temperatures and pressures in and out of two exchangers in series in discharge from pump
   - Pump curves
   - Normal and maximum operating conditions upstream of the pump
   - Exchangers have bypasses

   Be able to

   a - Specify the design temperatures for each exchanger
   b - Specify the design pressure for each exchanger

5. Given

   - A carbon steel piping system with uninsulated flanges
   - Design temperature
   - Design pressure

   Be able to

   a - Determine the flange rating for the pipe
   b - Calculate the maximum design pressure for the system using the selected flanges
   c - Calculate the maximum pressure to which the system may be subjected at the given temperature

6. In addition to the onsite process design considerations, be able to list at least three other systems that need to be considered in the complete design of a project.

7. In the preparation of a plot plan layout, be able to specify the desirable location for:

   a - Furnaces with respect to other equipment
   b - Cooling towers with respect to the unit
   c - Air fin exchangers with respect to fractionating towers and furnaces
   d - Pumps with respect to fractionating towers and furnaces

X. SAFETY CONSIDERATIONS

1. Be able to list the references used in the design of emergency release systems.

2. Be able to

   - Explain the meaning of the following terms:

   a - Autoignition Temperature
   b - Contingency
   c - Emergency
   d - Explosive Limits
   e - Flash Point
   f - High Flash Stocks
   g - Light Ends
   h - Low Flash Stocks
   i - Pyrophoric Material
   j - Single Risk
   k - Toxic Material
   l - Fire Zone
   m - Pressure Relief Valve
   n - Set Pressure
   o - Back Pressure
   p - Superimposed Back Pressure
   q - Built-up Back Pressure
   r - Spring Pressure
   s - MAWP
   t - Design Pressure
   u - Accumulation
   v - Open System
   w - Closed System
   x - Conventional Relief Valve
   y - Bellow Relief Valve
   z - Double Contingency

3. Given

   - The process flow sheet for a unit
   - Spacing standards

   Be able to

   - Determine the types of contingencies that should be considered for each vessel, heat exchanger, and furnace in the unit.

4. Given

   - A flow circuit including a centrifugal pump and downstream equipment
   - Maximum allowable working pressures for the downstream equipment
   - Pump curve
   - Normal and maximum pump suction pressures
   - Properties and rate of material being pumped

   Be able to

   a - Determine if a Pressure relief valve is needed on the pump discharge
   b - Size the relief valve if one is required
   c - Determine if a pressure relieving device should be considered for the low pressure side of any exchangers downstream of the pump

5. Given

- The pumping circuit in Item 4 but with no equipment design pressures

Be able to

a - Determine the design pressure for all downstream equipment not protected by a safety valve

6. Given

- The details of a horizontal drum, sphere, and fractionation tower
- The maximum allowable working pressures for each vessel
- The physical properties of material in the vessels

Be able to

a - Determine the fire load for each vessel
b - The size of the pressure relief valves required for the drum

7. Given

- A process flow sheet with all operating conditions
- Rates and compositions of each stream
- Design pressures for each piece of equipment
- Details on main safety release header

Be able to

a - Determine where pressure relief valves should be installed
b - Loads for each pressure relief valve installation
c - Size of valves for each contingency
d - Size of inlet and outlet lines for the PRV's
e - Explain reason for pressure drop limitation in inlet line
f - Choose between conventional and bellows valve based on built-up back pressure on valve
g - Determine which valves should discharge to a closed system and which may discharge to the atmosphere
h - Determine which PRV's should discharge to a flare and which to a condensible blowdown drum

INDEX

PROCESS DESIGN COURSE

LECTURES

DATA SOURCES

1. API Data Book
2. Maxwell—Data Book on Hydrocarbons
   (Within Exxon, "the Blue Book")
3. Tema Standards (Heat Exchangers)
4. Crane Technical Paper No. 410 (Fluid Flow)
5. API RP 520—Parts I & II (Safety)
6. API RP 521 (Safety)
7. Perry—Chemical Engineer's Handbook (General)
8. Marks—Handbook for Mechanical Engineers (General)
9. Previous Designs
10. Basic Practices of Refinery and/or Affiliate

1.01

BOILING POINTS

Type of Boiling Pt.	Abbreviation	Correlated Physical Property
1. Molal ABP	None, Spell Out	T_{PC}, Liquid Thermal Expansion
2. Weight ABP	WABP	T_C
3. Volume ABP	VABP	Liquid C_P, Viscosity
4. Mean ABP	MeABP, or MABP	MW, Watson K, ρ, P_{PC}, ΔH_{COMB}

- 1-3 Can Be Calculated by Linear Blends of Boiling Points of Fractions with Amount of Each Fraction.
- MeABP Correlates with Mol Wt.

1.02

MOLAL & WEIGHT AVERAGE BOILING POINTS

Comp.	Mol. Fr.	MW	B.P.	Mol. A B P Mol. Fr. x BP	lbs/Mol Mol. Fr. x MW	W A B P Wt. Fr. x BP
C_3	0.10	44.1	-43.8	-4.38	4.41	-3.41
IC_4	0.40	58.1	10.9	4.36	23.24	4.47
NC_4	0.50	58.1	31.1	15.55	29.05	15.93
	1.00			15.53	56.70	16.99

AVG MW → 56.70

VOLUME & MEAN AVERAGE BP

Comp.	Vol. Fr.	BP.°F	VABP (ΣVF x BP)	Lb/Gal	M.W.	Moles/ Gallon	Mole Fraction
NC_6	0.17	155.7	26.5	5.53	86.2	0.0109	0.189
NC_7	0.73	209.2	152.7	5.73	100.2	0.0417	0.721
NC_8	0.10	258.2	25.8	5.89	114.2	0.0052	0.090
	1.00		205.0	5.71		0.0578	1.000

Avg. Density = ΣVF x Lb/Gal = 5.71

S.G. = 5.71/8.33 = 0.685

$$°API = \frac{141.5}{S.G.} - 131.5 = 75.0$$

$$M.W. = \frac{Lb/Gal}{Moles/Gal} = \frac{5.71}{0.0578} = 98.8$$

MeABP = 185°F (Maxwell P. 21)
(Blue Book P. 3.20)

CHARACTERIZATION FACTOR

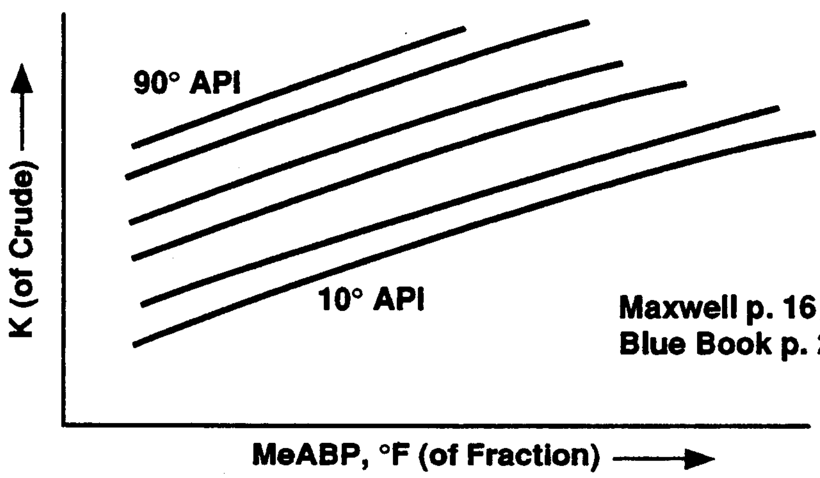
- An Index of Chemical Character:

$$K = \frac{\sqrt[3]{\text{MeABP, °R}}}{\text{SP. GR.}}$$

- Applies to Entire Boiling Range of theCrude.
- Use to Get Specific Gravity of Petroleum Fractions (Cuts).
- Use to Get Specific Heat of Petroleum Fraction Vapors.

1.05

USE OF CHARACTERIZATION FACTOR



Maxwell p. 16
Blue Book p. 2.20

1.06

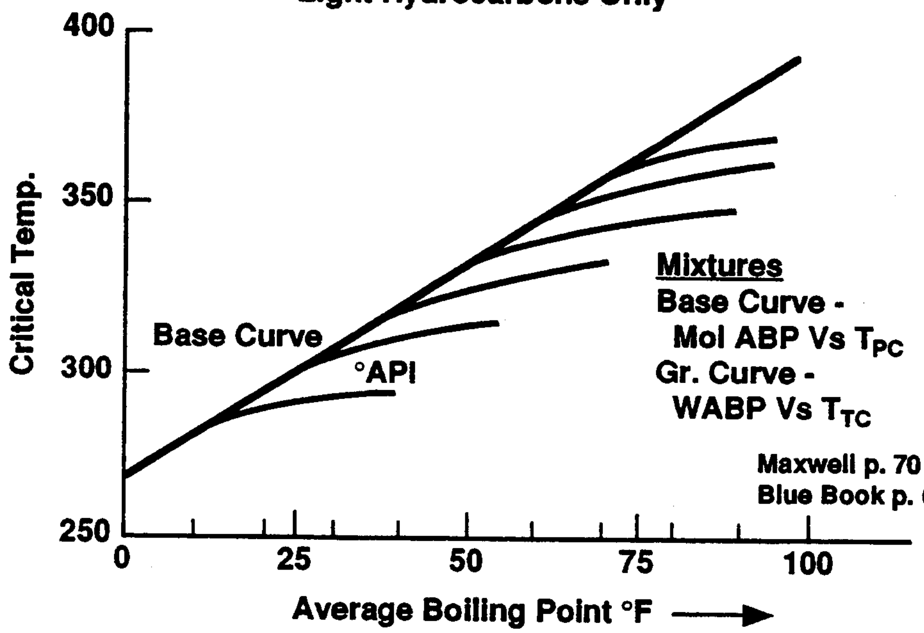
CRITICAL PROPERTIES

- Our Primary Use
  - — Compressibility of Gases
  - — Correlations for Equilibrium Constants
- True-Critical - for Pure Compounds
- Pseudo-Critical - for Mixtures

1.07

PSEUDO-CRITICAL TEMPERATURE (T_{PC})

Light Hydrocarbons Only

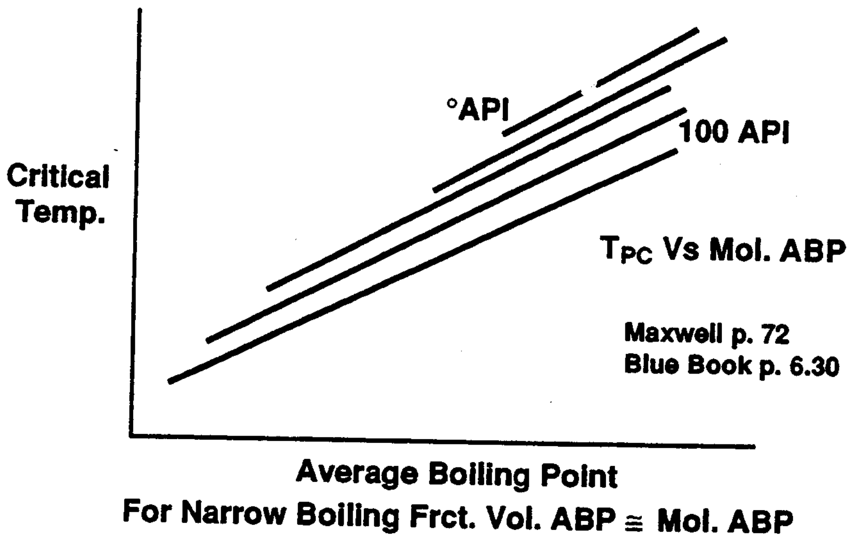


Maxwell p. 70
Blue Book p. 6.16

1.08

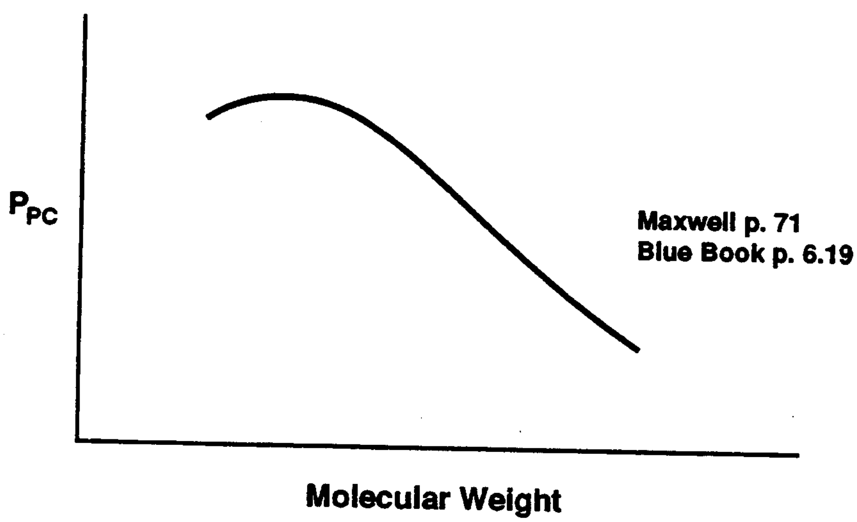
PSEUDO-CRITICAL TEMPERATURE (T_{PC})

Petroleum Fractions



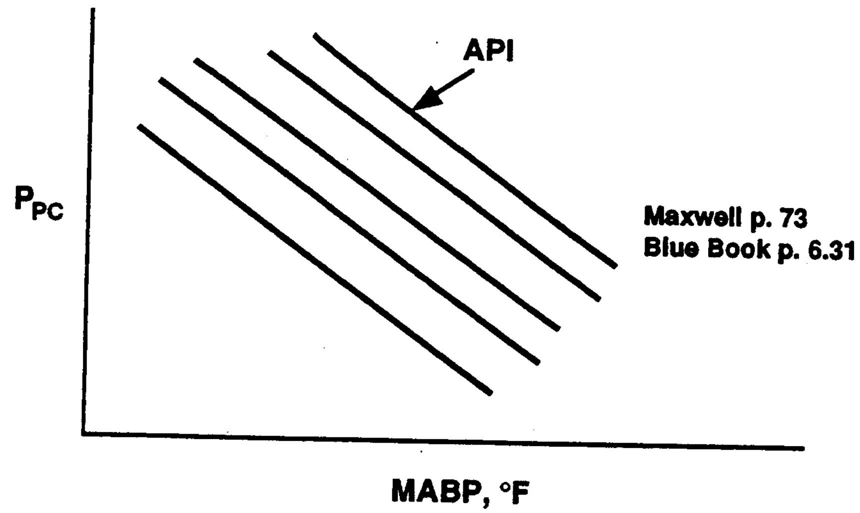
PSEUDO-CRITICAL PRESSURE (P_{PC})

Light Hydrocarbons - Normal Paraffin Mixtures



PSEUDO-CRITICAL PRESSURE (P_{PC})

Petroleum Fractions



1.11

GAS DENSITY

$PV = n\mu RT = \mu RT$ for 1 Mol

Volume of 1 Mol $= \frac{\mu RT}{P}$

Weight of 1 Mol = M

$$\rho = \frac{lb}{ft^3} = \frac{M}{\frac{\mu RT}{P}}$$

$$\rho = \frac{PM}{\mu RT} = \frac{lbs}{ft^3}$$

P = PSIA T = °R R = 10.731

1.12

DENSITY OF LIQUIDS

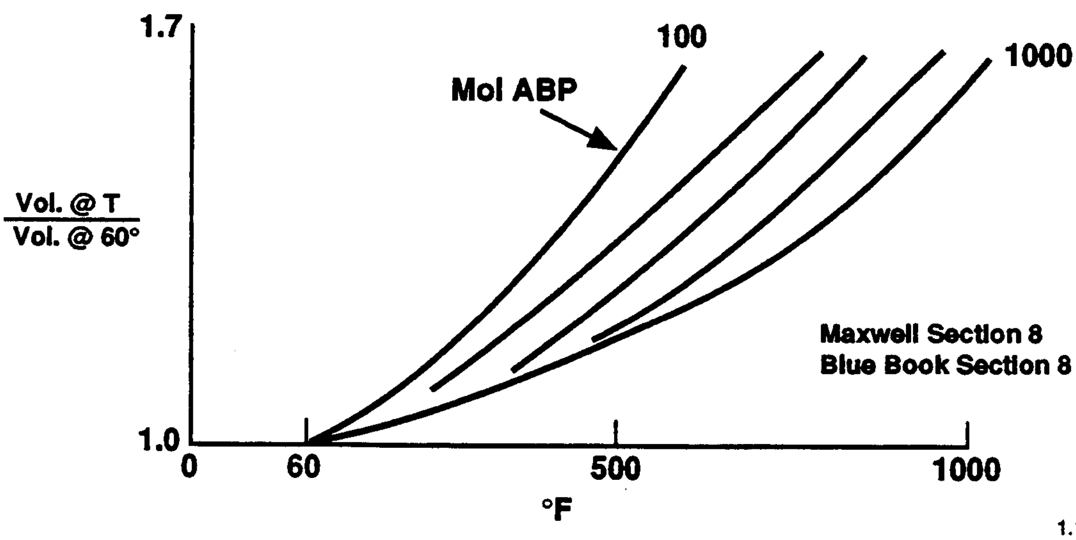
- **See Section 8 of Maxwell**
- **Density Calculations**

  Do Not Blend API.

  Blend Specific Gravity, or lbs/ft³

1.13

THERMAL EXPANSION OF LIQUIDS



1.14

THERMAL PROPERTIES

Section 7 of Maxwell and Blue Book

- Specific Heat
- Latent Heat
- Enthalpy

1.15

VISCOSITY

Section 9 of Maxwell and Blue Book

- Pure Compounds
- Pet. Fractions
- Visc. - Temp. Charts
- Visc. Conversions
- Visc. Blending
- VBI

1.16

VISCOSITY OF LIQUID MIXTURES

Page 173 Maxwell
Page 9.61 Blue Book

Comp.	Vol. Fr.	Visc. @ 50°F, CS	VBI
A	0.5	15.0	36.4
B	0.3	31.5	31.8
C	0.2	131.0	24.7
	1.0	ΣVF x VBI =	32.68
		Vol. Average	
		Viscosity, CS	26.7

VISCOSITY OF GAS MIXTURES

- Viscosity Charts - Section 9 of Maxwell & Blue Book
- Calculations:

$$Z = \frac{N_1 Z_1 \sqrt{M_1} + N_2 Z_2 \sqrt{M_2} + \cdots}{N_1 \sqrt{M_1} + N_2 \sqrt{M_2} + \cdots}$$

Z = Visc. of Mix. at Atmos. Press.

N_1, N_2 = Mol. Fr. of Indiv. Components

Z_1, Z_2 = Visc. of Indiv. Components

M_1, M_2 = Mol. Wt of Components

- Correct for Pressure (Maxwell Page 177, Blue Book 9.87)

VAPOR-LIQUID EQUILIBRIUM

- **Whenever a Liquid and Vapor Exist in Equilibrium, It Is Possible to Determine the Composition of Each of the Phases. V-L-E Relationships Are the Basis for Bubble Point, Dew Point and Flash Calculations.**

- **For Any Specific Component at V-L Equilibrium, its Concentration in the Vapor Phase Divided by its Concentration in the Liquid Phase is Known as its Equilibrium Constant, or K.**

IDEAL SYSTEM — EQUILIBRIUM CONSTANTS

Raoult's Law:	$\rho_i = P_i X_i$	ρ_i = **Partial Pressure of i in Vapor**
Dalton's Law:	$\rho_i = \pi Y_i$	P_i = **Vapor Pressure of i @ Temp.**
	$P_i X_i = \pi Y_i$	π = **Total System Pressure** X_i = **Mol. Frct. of i in Liquid** Y_i = **Mol. Frct. of i in Vapor**
	$\dfrac{Y_i}{X_i} = \dfrac{P_i}{\pi} = K_i$	**K = Equilibrium Constant**
	$K_i = f(T,P)$	**Good up to About 4 Atmospheres for Systems Comprised of Similar Compounds**

NON-IDEAL SYSTEM

- **Activity Coefficient Corrects for Non-Ideal Liquid Solutions**
  - **— Neccessary in Calculations Involving Polar Compounds; e.g. Sour Water Strippers**
- **Equilibrium Constants for Non-Ideal Vapor Phase Are Obtained from a Variety of Computerized Correlations, e.g.**
  - **— Benedict Webb Rubin**
  - **— Chao Seader**
  - **— Redlich Kwong**

FUGACITY FUNCTIONS

- **Fugacity Function May be Considered a Corrected Vapor Pressure.**
- **Hence** $K_i = \frac{f(F_i)}{\pi} = \frac{\text{Fugacity Function}}{\text{Total Pressure}}$
- **When K_i is Determined by a Rigorous Methods such as Chao - Seader, then the product of K_i and π may be Called a "Fugacity Function".**
- **Maxwell Chapter 5 Gives Graphs for Pure Component Fugacity Functions. These Are Only Approximate Values (Non Composition-Dependent). They Cannot be Used for Definitive Designs.**

BUBBLE POINT CALCULATION USING GAS LAWS

System Pressure = 120 PSIA = 8.17 ATM = π

(See Maxwell Chapter 4)

	Vapor Pressure = P_i			$K_i = P_i/\pi$		
	130°F	135°F	140°F	130°F	135°F	140°F
C_3	18.30	19.50	20.70	2.240	2.390	2.530
IC_4	7.46	8.00	8.51	0.913	0.979	1.040
nC_4	5.46	5.90	6.36	0.668	0.722	0.779

$Y_i = K_iX_i$

	X_i	130°F	135°F	140°F
C_3	0.0133	0.0298	0.0318	0.0336
IC_4	0.9516	0.8688	0.9316	0.9897
nC_4	0.0351	0.0234	0.0253	0.0273
	1.0000	0.9220	0.9887	1.0506

Linear Interpolation Gives Bubble Point = 136°F

$$\left.\begin{array}{l}\text{FUGB} = 135.9°\text{F}\\ \text{CHAO} = 137.5°\text{F}\\ \text{RKJZ} = 135.8°\text{F}\end{array}\right\}\ \text{Rigorous Methods}$$

DEW POINT CALCULATION USING IDEAL GAS LAWS

System Pressure = 132 PSIA = 8.982 ATM = π

	Vapor Pressure = P_i			$K_i = P_i/\pi$		
	150°F	155°F	160°F	150°F	155°F	160°F
IC_4	9.71	10.40	11.00	1.081	1.160	1.220
nC_4	7.27	7.75	8.30	0.809	0.863	0.924
IC_5	3.10	3.34	3.60	0.345	0.372	0.401

$X_i = Y_i/K_i$

	Y_i	150°F	155°F	160°F
IC_4	0.4590	0.425	0.396	0.376
nC_4	0.5393	0.667	0.625	0.584
IC_5	0.0017	0.005	0.005	0.004
	1.0000	1.097	1.026	0.964

Linear Interpolation Gives Bubble Point = 157°F

$$\left.\begin{array}{l}\text{FUGB} = 157.5°\text{F}\\ \text{CHAO} = 159.0°\text{F}\\ \text{RKJZ} = 157.2°\text{F}\end{array}\right\}\ \text{Rigorous Methods}$$

EQUATIONS FOR FLASH VAPORIZATION

F = Mols of Liquid Feed
V= Mols of Vapor Product
L = Mols of Liquid Product

z_i = Mol Fraction of i in Feed
y_i = Mol Fraction of i in Vapor
x_i = Mol Fraction of i in Liquid

- Material Balance: $Fz_i = Vy_i + Lx_i$
- For 1 Mol of Feed: $z_i = Vy_i + Lx_i$
- By Equilibrium: $x_i = \frac{y_i}{k_i}$

1.25

EQUATIONS FOR FLASH VAPORIZATION (Continued)

$z_i = Vy_i + Lx_i$ (Previous Slide)

Divide by Vy_i:

$$\frac{z_i}{Vy_i} = 1 + \frac{Lx_i}{Vy_i} = 1 + \frac{L}{VK_i}$$

$$z_i = Vy_i\left(1 + \frac{L}{VK_i}\right)$$

$$Vy_i = \frac{z_i}{1 + \frac{L}{VK_i}}$$

1.26

FLASH VAPORIZATION CALCULATION

$$V = \sum_{i=1}^{i=n} Vy_i = \sum_{i=1}^{i=n} \frac{Z_i}{1 + \frac{L}{VK_i}}$$

STEP 1. Assume a Value of V.
Then L = 1-V, and $\frac{L}{V} = \frac{1-V}{V}$

STEP 2. Calculate Vy_i for Each Component

STEP 3. Sum to get V. Iterate until $V_{assumed} = V_{calculated}$

Convergence Can be Difficult. Agreement of $V_{assumed}$ with $V_{calculated}$ should be within 0.1% to Assure a True Solution.

1.27

FLASH CALCULATION EXAMPLE

P = 200 PSIA T = 180°F

Assume V = 0.420; L/V = 1.381

Comp.	Z	K(1)	1 + L/VK	$\frac{Z}{1 + L/VK}$
C2	0.08	4.1710	1.3311	0.0601
C3	0.22	1.8650	1.7405	0.1264
NC4	0.53	0.8233	2.6774	0.1980
NC5	0.17	0.3760	4.6729	0.0364
	1.00			0.4209

Assume V = 0.429; L/V = 1.331

	1 + L/VK	$\frac{Z}{1 + L/VK}$
C2	1.3311	0.0601
C3	1.7405	0.1264
NC4	2.6774	0.1980
NC5	4.6729	0.0364
		0.4289 *OK!*

(1) K Values Are From Rigorous Calculation, Method RKJZ.

DISTILLATION CONVERSIONS

Definitions

• 15/5	— 15 Trays/5:1 Reflux — Approx. of Actual Boiling Point of Components
• TBP	— Ambiguous Connotation; Preferable Term is 15/5
• Stem Correction	— When Thermometers Are Used All the Stem is Not Immersed. Need to Correct for This.
• ASTM Distillations	— Handout 1.01
• GC Distillation	— Gas Chromatography ("GC's") has Largely Replaced the 15/5. For GC's, the IBP is Taken as 0.5 Vol.% Point; the FBP is Defined as the 99.5 Vol.% Point.

1.29

CONVERSION OF ASTM DISTILLATION TO A 15/5 DISTILLATION

- Establish Type of Data Reported
  - — Stem Corrected?
  - — Procedure Name?
- Plot Data to Arrive at Smooth Curve and Reject Sports
- Stem Correct the Smooth Curve, If Not Done by Laboratory
- Use Charts in Handout 1.01, Pages 2-3, to Convert This Stem Corrected Curve to 15/5

1.30

DISTILLATION CONVERSION EXAMPLE

Stem Corrected ASTM D-86 (Naphtha)

TBP

LV%	ASTM TEMP °F	50%-10% Increment °F ①	90%-50% Increment °F ②	15/5 Temp. °F
5	347	-33		314
10	356	-30		326
50	374	-1	0	374
90	401		16	417
95	410		17	427
FBP	425			

Call Avg = 0

50% - 10% = 374 - 356 = 18°F
90% - 50% = 401 - 374 = 27°F

① Addendum 1.01, Page 4
② Addendum 1.01, Page 5

PSEUDO - COMPONENTS

- If a Component Analysis is not Available, a Composition in Terms of "Pseudo" Components Can be Obtained from the Distillation Curve and Gravity.

- How Do We Represent the Components?

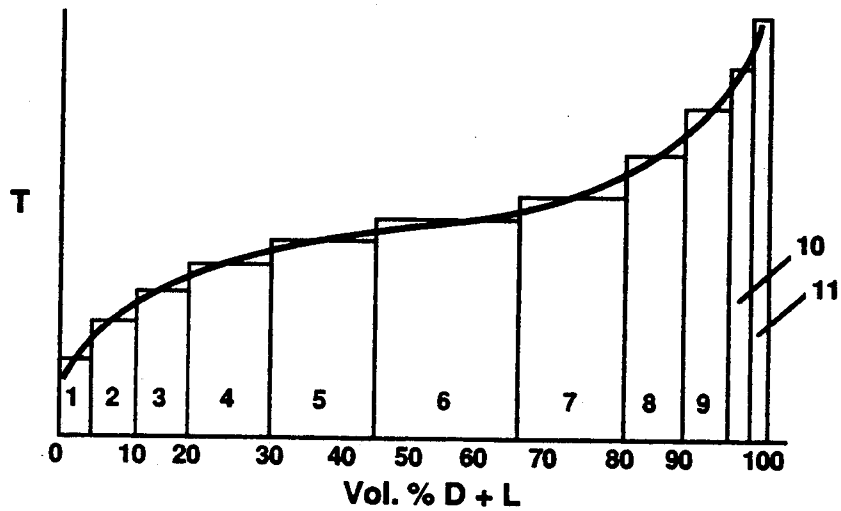
350 A 45.0   350 S 0.80

350 K 11.6

- How Do We Get Pseudo-Components?

PSEUDO-COMPONENT BOILING POINT

- If Only an ASTM Dist. Curve is Available, Convert It to a 15/5 Distillation. √
- Divide the Curve Into Rectangles, Each Rectangle Representing a Pseudo Component.
- Tops of the Rectangle are the Vol. Avg. BP's: 50% of each "Component" Boils Lower than the Top at Rectangle, 50% Boils Higher.



1.33

PSEUDO COMPONENT GRAVITY

- Have T_{VABP} and Vol% of Each Fraction
- Have Gravity of Total
- Calc. Char. Factor, K, of Total Feed

$$K = \frac{\sqrt[3]{MeABP,\ ^\circ R}}{SP.\ GR.}$$

(for Narrow Boiling Fractions, Can Assume $T_{VABP} = T_{MeABP}$)

- Use Same Equation, Solved for S.G., to Calculate S.G. of Each Pseudocomponent.

1.34

PRINCIPAL FEATURES OF CURRENT ASTM DISTILLATION METHODS

Method Number	Used For:	Thermo-meter[b] Range, °F	Boiling Point Limits °F Initial	Boiling Point Limits °F Final	Std. Oper. Pres-sure	Distillation Flask Capac. ml	Distillation Flask Insu-lation	Distillation Flask Shield Diam. In.	Probable Repeatability[a] °F IBP	Probable Repeatability[a] °F FBP	Probable Repeatability[a] °F Other	Probable Reproducibility[a] °F IBP	Probable Reproducibility[a] °F FBP	Probable Reproducibility[a] °F Other
ASTM D-86[c]	Naphthas, full Range or Heavy	30-580	No Spec.	<482	Atm.	125	None	1.5	2-4	2-3	2-4	6-10	6-8	5-10
ASTM D-86[c]	Light Distillates Hvy. Naphtha, Kerosene[d]	30-580	≤212	<482	Atm.	125	None	1.5	2-4	2-3	2-4	6-10	6-8	5-10
		30-580	>212	>482				2						
ASTM D-86[c]	Mid. Distillates[d] (gas oil)	30-760	>212	>482	Atm.	125	None	2	2-4	2-3	2-4	6-10	6-8	5-10
ASTM D-158	**Obsolete. (Use High Range D-86)**	30-760	No Spec.	No Spec.	Atm.	250	None	2¾	----	No Spec.	---	No Spec.	6	No Spec.
ASTM D-1160[e]	High Boiling Distillates and Residua	Thermo-couple[e]	No Spec.	620[g]	10 mm Hg	250	Vari-able[e]	None	----	------	10	-------	----	15-20
LIC 10.06[f]		30-760	No Spec.	620[g]	10 mm Hg	250	None	2¾	No Specification					

[a] Approximate range of values quoted by the A.S.T.M. for medium and wide-boiling materials, which apply only to official ASTM manual methods.

[b] Total immersion thermometer immersed 3-4 in. in vapor space, hence readings are not true temperatures. Thermocouples used for automatic D-86 methods are calibrated to give readings equal to total immersion thermometers specified.

[c] Automatic adaptations of these methods are widely used, which are normally calibrated to give practically the same results as the official ASTM methods. However, the automatic methods may give erroneous results on narrow-boiling (125°F) fractions.

[d] Note that these materials, if low boiling, may be run either with low or high range thermometers which will *not* give equivalent results.

[e] Although widely used, this vacuum method is not an acceptable standard, being too loosely defined and poorly designed. The massive, non-sensitive thermocouple specified normally exhibits a considerable lag in vapor temperature measurement, which increases as boiling range increases. Apparently, as a result partly of this variable and partly of variations permitted in the vapor zone assembly, the data are generally unreliable. Thus, reproducibility may be much poorer than claimed. The vapor zone may consist of one of two different flasks (long neck, 100 mm, or short neck, 25mm) combined with one of two different vapor heads, one insulated with asbestos, the other vacuum jacketed.

[f] Antiquated, poorly standardized vacuum method, commonly called 10 mm vacuum Engler. Data generally unreliable.

[g] At operating pressure.

STEM-CORRECTED THERMOMETER READINGS FOR ASTM D-86 NAPHTHA DISTILLATION

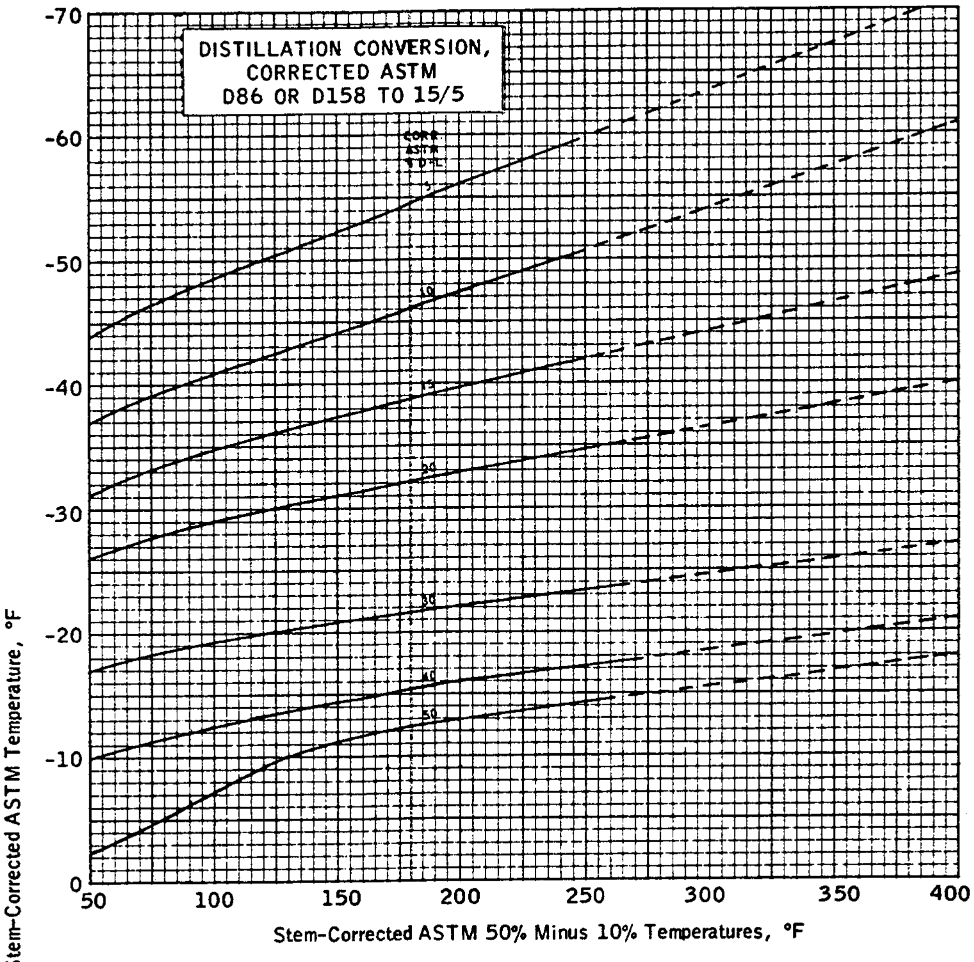
(LOW RANGE THERMOMETER, 30-580°F)

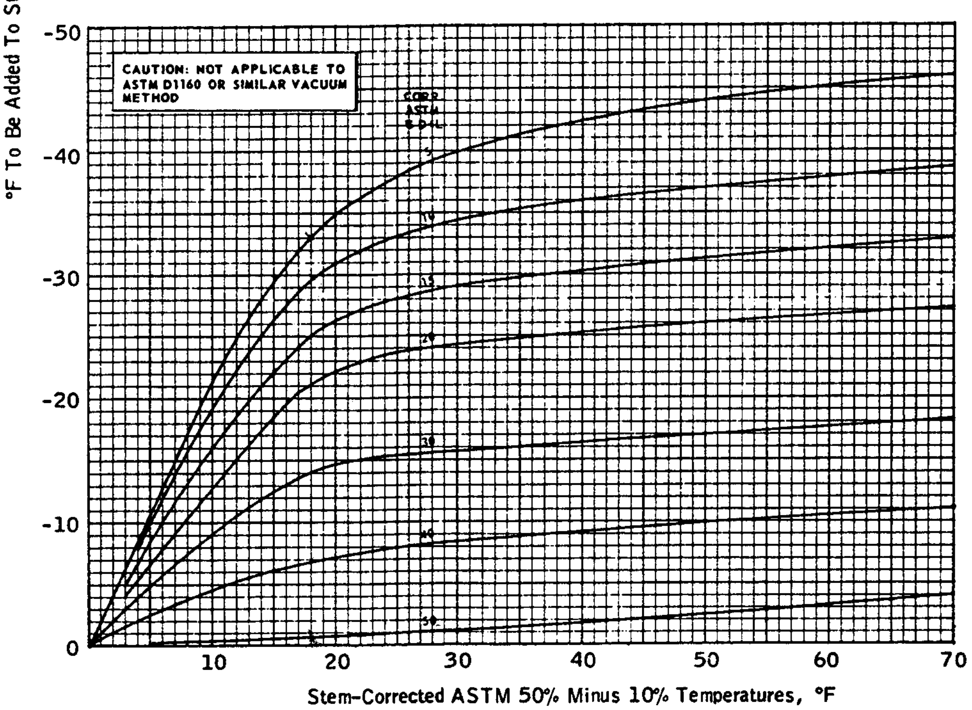
Standard Observed Reading °F	READING CORRECTED FOR THERMOMETRIC ERROR, °F (Correction is zero below 140°F)									
	0	1	2	3	4	5	6	7	8	9
140	141	142	143	144	145	146	147	148	149	150
150	**151**	**152**	**153**	**154**	**155**	**156**	**157**	**158**	**159**	**160**
160	161	162	163	164	165	166	167	168	169	170
170	171	172	173	174	175	176	177	178	179	180
180	181	182	183	184	185	186	188	189	190	191
190	192	193	194	195	196	197	198	199	200	201
200	**202**	**203**	**204**	**205**	**206**	**207**	**208**	**209**	**210**	**211**
210	212	213	214	215	216	217	218	219	220	221
220	222	223	224	225	227	228	229	230	231	232
230	233	234	235	236	237	238	239	240	241	242
240	243	244	245	246	247	248	249	250	251	252
250	**253**	**254**	**255**	**256**	**257**	**259**	**260**	**261**	**262**	**263**
260	264	265	266	267	268	269	270	271	272	273
270	274	275	276	277	278	279	280	281	282	283
280	284	285	286	288	289	290	291	292	293	294
290	295	296	297	298	299	300	301	302	303	304
300	**305**	**306**	**307**	**308**	**309**	**310**	**311**	**313**	**314**	**315**
310	316	317	318	319	320	321	322	323	324	325
320	326	327	328	329	330	331	332	333	334	336
330	337	338	339	340	341	342	343	344	345	346
340	347	348	349	350	351	352	353	354	355	357
350	**358**	**359**	**360**	**361**	**362**	**363**	**364**	**365**	**366**	**367**
360	368	369	370	371	372	373	374	376	377	378
370	379	380	381	382	383	384	385	386	387	388
380	389	390	391	392	394	395	396	397	398	399
390	400	401	402	403	404	405	406	407	408	409
400	**411**	**412**	**413**	**414**	**415**	**416**	**417**	**418**	**419**	**420**
410	421	422	423	424	425	426	428	429	430	431
420	432	433	434	435	436	437	438	439	440	441
430	443	444	445	446	447	448	449	450	451	452
440	453	454	455	456	458	459	460	461	462	463
450	**464**	**465**	**466**	**467**	**468**	**469**	**470**	**472**	**473**	**474**
460	475	476	477	478	479	480	481	482	483	484
470	486	487	488	489	490	491	492	493	494	495
480	496	497	499	500	501	502	503	504	505	506
490	507	508	509	510	512	513	514	515	516	517
500	**518**	**519**	**520**	**521**	**522**	**523**	**525**	**526**	**527**	**528**
510	529	530	531	532	533	534	535	537	538	539
520	540	541	542	543	544	545	546	547	549	550
530	551	552	553	554	555	556	557	558	559	561
540	562	563	564	565	566	567	568	569	570	571
550	**573**	**574**	**575**	**576**	**577**	**578**	**579**	**580**	**581**	**582**
560	583	585	586	587	588	589	590	591	592	593
570	594	596	597	598	599	600	601	602	603	604
580	605	607	608	609	610	611	612	613	614	615
590	616	617	619	620	621	622	623	624	625	626
600	**627**	**628**	**630**	**631**	**632**	**633**	**634**	**635**	**636**	**637**

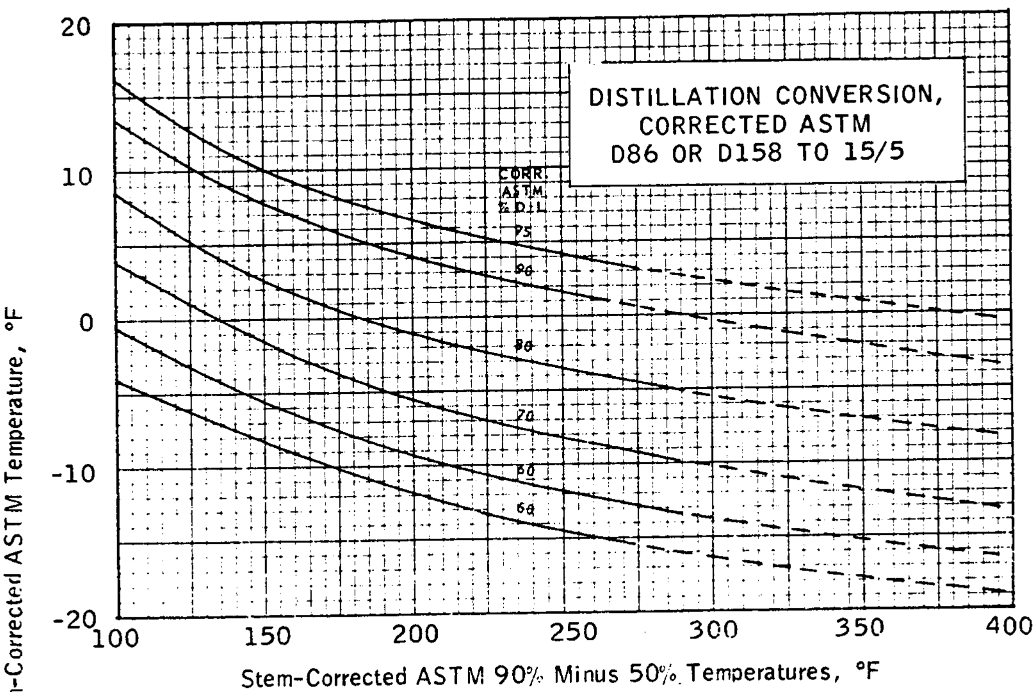
STEM-CORRECTED THERMOMETER READINGS FOR ASTM GAS OIL DISTILLATIONS, D-86 AND D-158

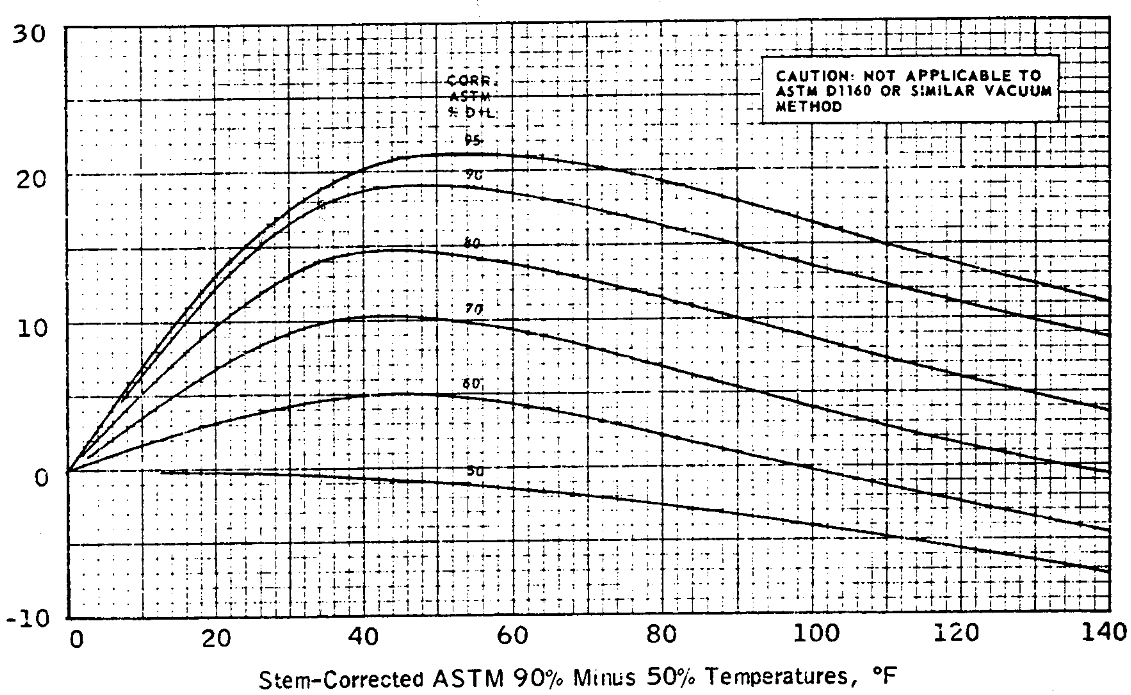
(HIGH RANGE THERMOMETER, 30-750°F)

Standard Observed Reading °F	READING CORRECTED FOR THERMOMETRIC ERROR, °F (Correction is zero below 220°F observed)									
	0	1	2	3	4	5	6	7	8	9
220	220	221	222	223	224	226	227	228	229	230
230	231	232	233	234	235	236	237	238	239	240
240	241	242	243	244	245	246	247	248	249	250
250	**251**	**252**	**253**	**254**	**255**	**256**	**257**	**258**	**259**	**260**
260	261	262	263	264	265	266	267	268	269	270
270	271	273	274	275	276	277	278	279	280	281
280	282	283	284	285	286	287	288	289	290	291
290	292	293	294	295	296	297	298	299	300	301
300	**302**	**303**	**304**	**305**	**306**	**307**	**308**	**309**	**310**	**311**
310	312	314	315	316	317	318	319	320	321	322
320	323	324	325	326	327	328	329	330	331	332
330	333	334	335	336	337	338	339	340	341	342
340	343	344	345	346	347	348	350	351	352	353
350	**354**	**355**	**356**	**357**	**358**	**359**	**360**	**361**	**362**	**363**
360	364	365	366	367	368	369	370	371	372	373
370	374	375	376	377	378	379	380	381	382	384
380	385	386	387	388	389	390	391	392	393	394
390	395	396	397	398	399	400	401	402	403	404
400	**405**	**406**	**407**	**408**	**409**	**410**	**411**	**412**	**413**	**415**
410	416	417	418	419	420	421	422	423	424	425
420	426	427	428	429	430	431	432	433	434	435
430	436	437	438	440	441	442	443	444	445	446
440	447	448	449	450	451	452	453	454	455	456
450	**457**	**458**	**459**	**460**	**461**	**462**	**464**	**465**	**466**	**467**
460	468	469	470	471	472	473	474	475	476	477
470	478	479	480	481	482	483	484	486	487	488
480	489	490	491	492	493	494	495	496	497	498
490	499	500	501	502	503	504	505	507	508	509
500	**510**	**511**	**512**	**513**	**514**	**515**	**516**	**517**	**518**	**519**
510	520	521	522	523	524	526	527	528	529	530
520	531	532	533	534	535	536	537	538	539	540
530	542	543	544	545	546	547	548	549	550	551
540	552	553	554	555	556	558	559	560	561	562
550	**563**	**564**	**565**	**566**	**567**	**568**	**569**	**570**	**571**	**572**
560	574	575	576	577	578	579	580	581	582	583
570	584	585	586	588	589	590	591	592	593	594
580	595	596	597	598	599	601	602	603	604	605
590	606	607	608	609	610	611	612	614	615	616
600	**617**	**618**	**619**	**620**	**621**	**622**	**623**	**624**	**626**	**627**
610	628	629	630	631	632	633	634	635	636	638
620	639	640	641	642	643	644	645	646	647	649
630	650	651	652	653	654	655	656	657	658	660
640	661	662	663	664	665	666	667	668	669	671
650	**672**	**673**	**674**	**675**	**676**	**677**	**678**	**679**	**681**	**682**
660	683	684	685	686	687	688	689	691	692	693
670	694	695	696	697	698	699	701	702	703	704
680	705	706	707	708	710	711	712	713	714	715
690	716	717	719	720	721	722	723	724	725	727
700	**728**	**729**	**730**	**731**	**732**	**733**	**734**	**736**	**737**	**738**
710	739	740	741	742	744	745	746	747	748	749
720	750	752	753	754	755	756	757	759	760	761
730	762	763	764	765	767	768	769	770	771	772
740	773	775	776	777	778	779	780	782	783	784
750	**785**	**786**	**787**	**788**	**790**	**791**	**792**	**793**	**794**	**795**

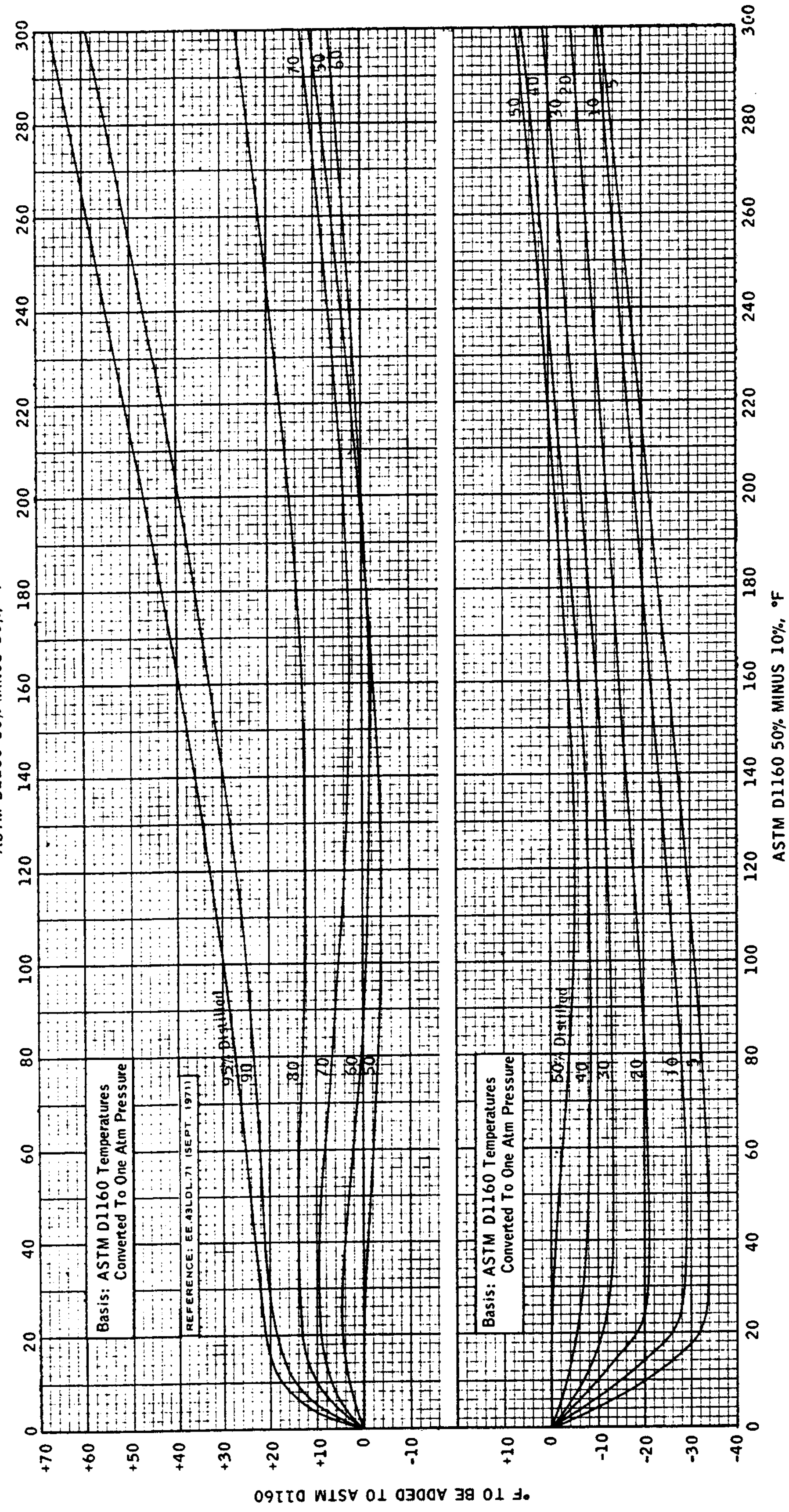

DISTILLATION CONVERSION,
CORRECTED ASTM
D86 OR D158 TO 15/5
CORR ASTM % D+L
°F To Be Added To Stem-Corrected ASTM Temperature, °F
Stem-Corrected ASTM 50% Minus 10% Temperatures, °F


CAUTION: NOT APPLICABLE TO ASTM D1160 OR SIMILAR VACUUM METHOD
CORR ASTM % D+L
Stem-Corrected ASTM 50% Minus 10% Temperatures, °F

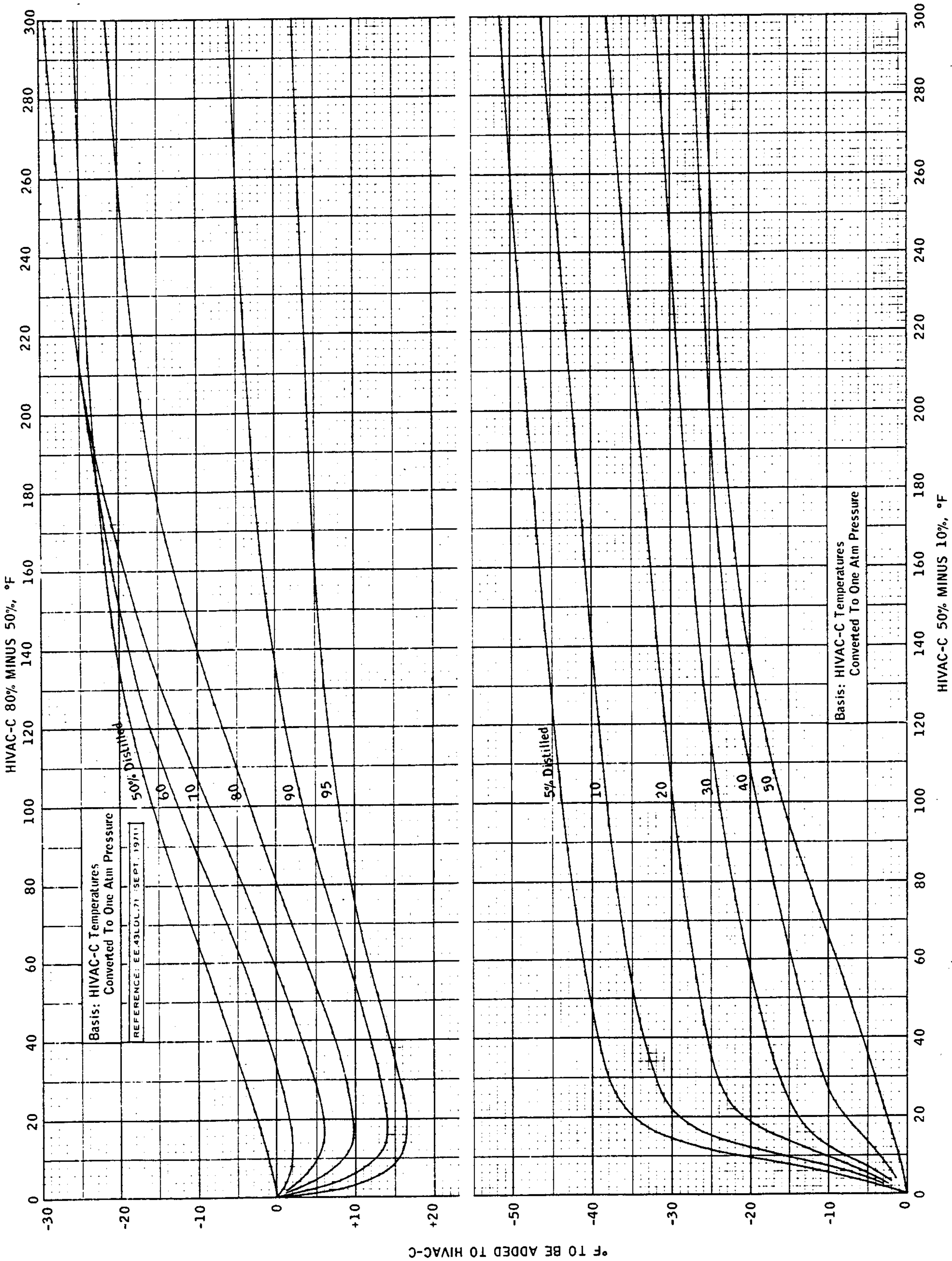

DISTILLATION CONVERSION,
CORRECTED ASTM
D86 OR D158 TO 15/5
CORR.
ASTM
% DIL
95
90
80
70
60
50
20
10
0
-10
-20
100
150
200
250
300
350
400
Stem-Corrected ASTM 90% Minus 50% Temperatures, °F
°F To Be Added To Stem-Corrected ASTM Temperature, °F

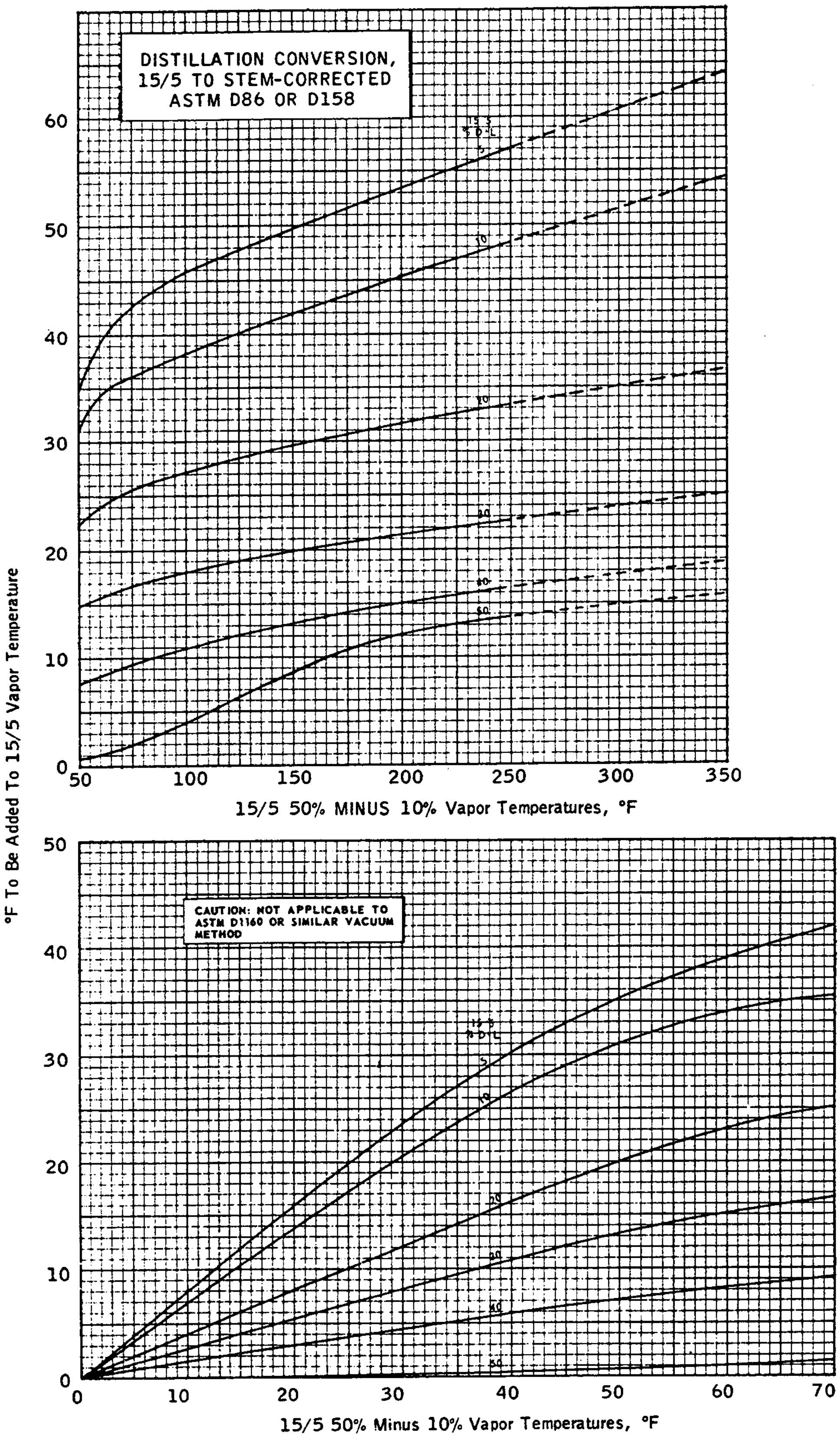

CAUTION: NOT APPLICABLE TO ASTM D1160 OR SIMILAR VACUUM METHOD
CORR.
ASTM
% DIL
95
90
80
70
60
50
30
20
10
0
-10
0
20
40
60
80
100
120
140
Stem-Corrected ASTM 90% Minus 50% Temperatures, °F

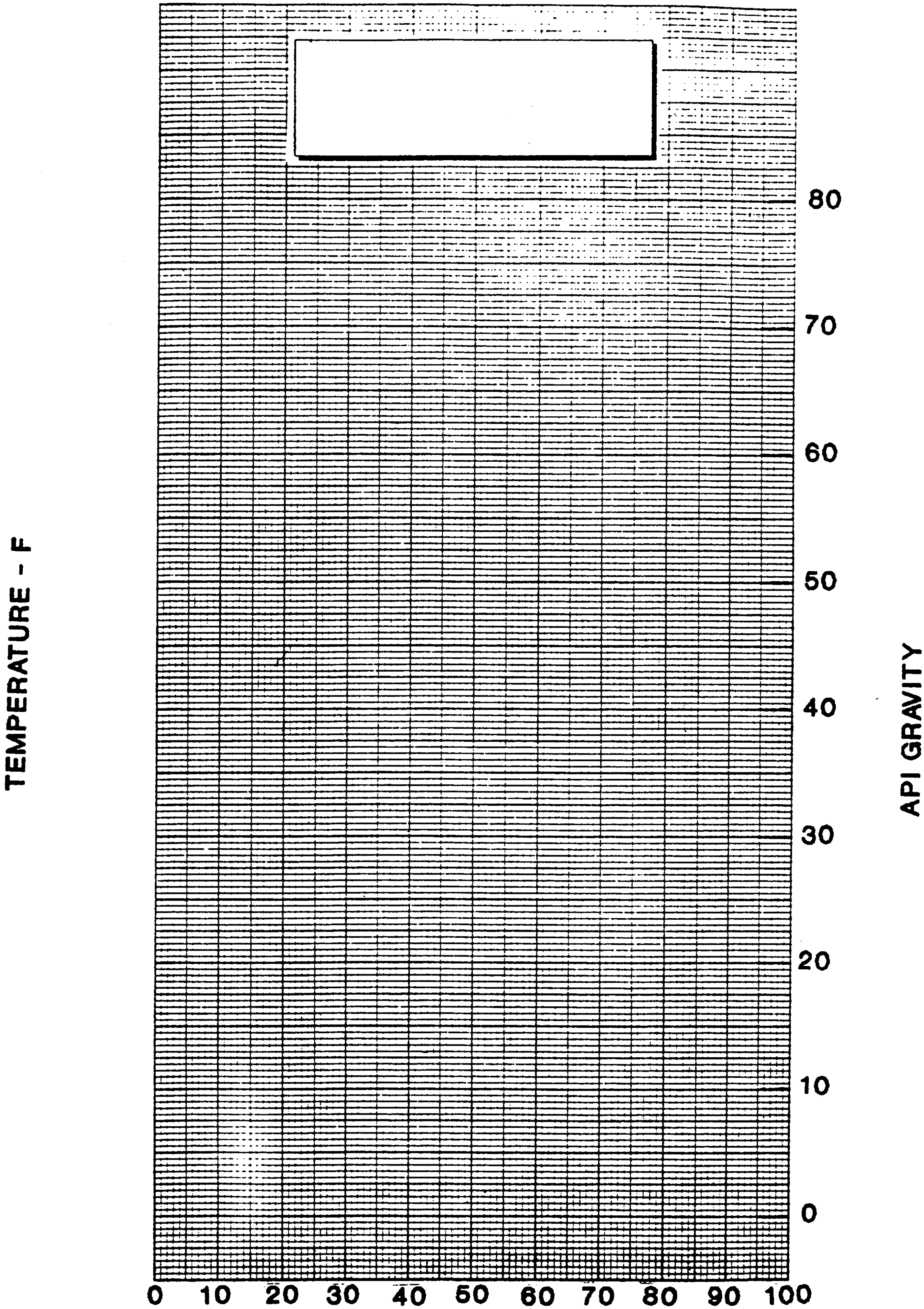
VACUUM DISTILLATION INTERCONVERSION
STANDARD ASTM D1160 TO 15/5

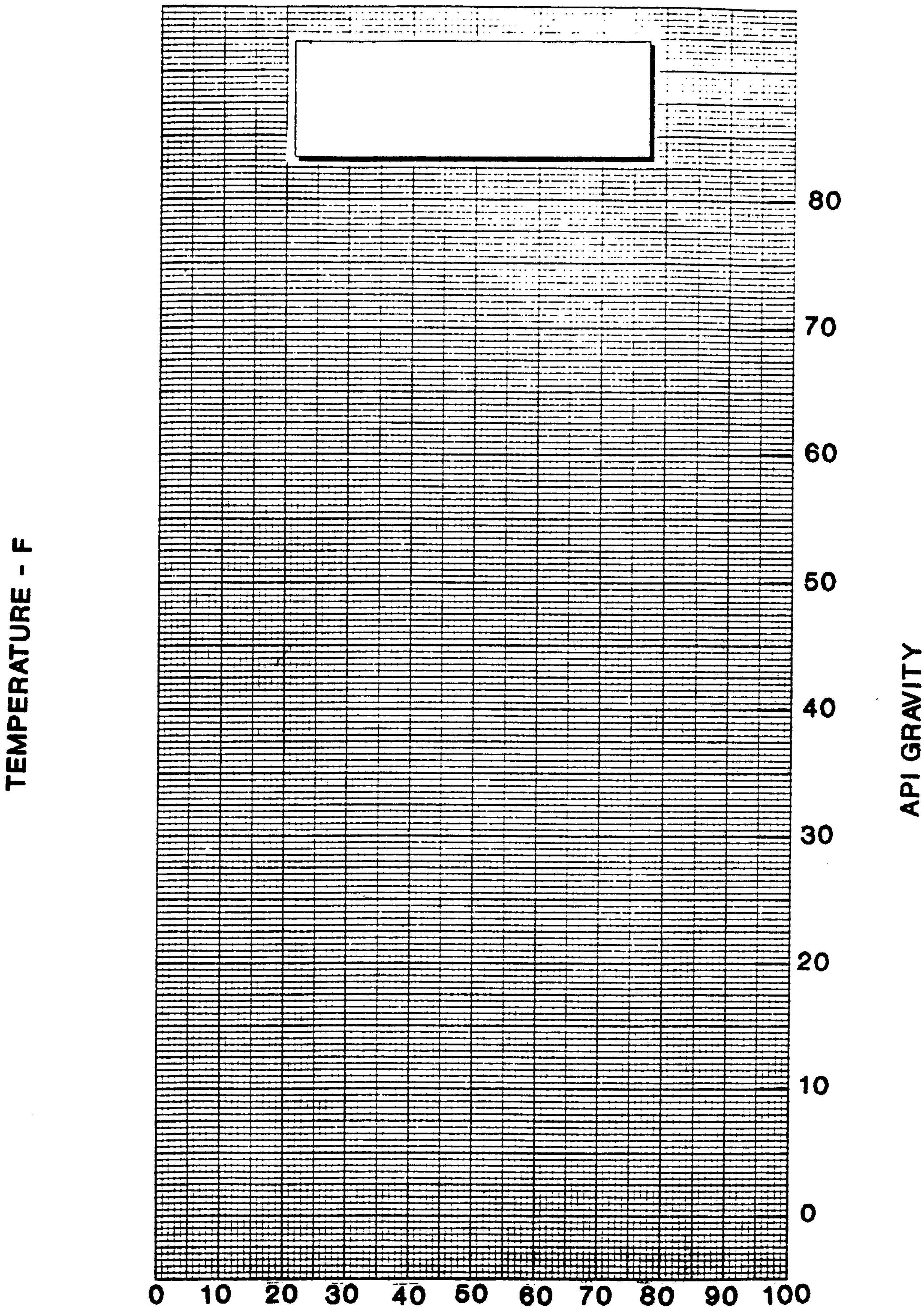


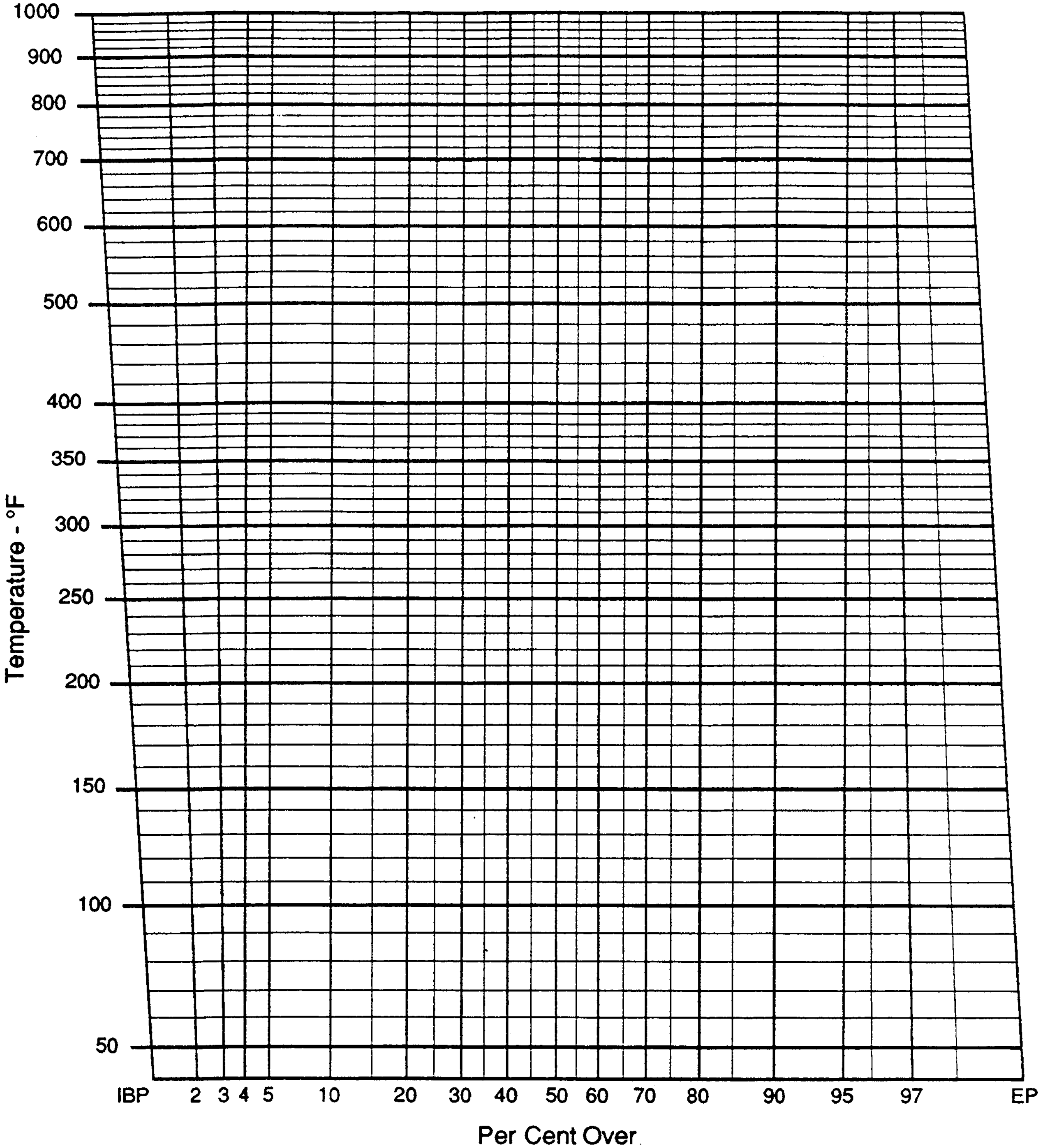
VACUUM DISTILLATION CON[illegible]N, HIVAC-C TO 15/5

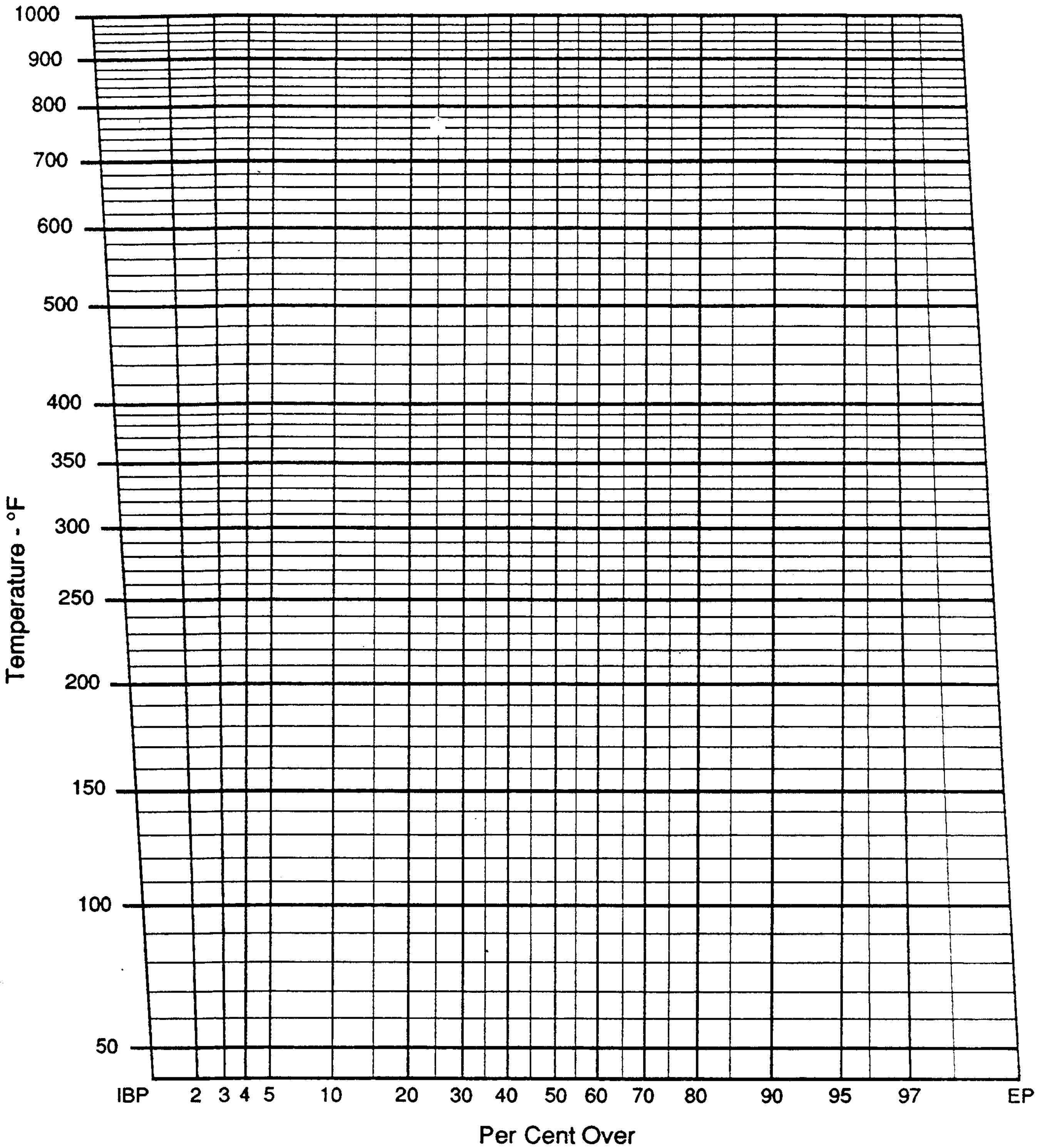



DISTILLATION CONVERSION,
15/5 TO STEM-CORRECTED
ASTM D86 OR D158
°F To Be Added To 15/5 Vapor Temperature
15/5 50% MINUS 10% Vapor Temperatures, °F
CAUTION: NOT APPLICABLE TO
ASTM D1160 OR SIMILAR VACUUM
METHOD
15/5 50% Minus 10% Vapor Temperatures, °F


TEMPERATURE - F
API GRAVITY
80
70
60
50
40
30
20
10
0
0 10 20 30 40 50 60 70 80 90 100


TEMPERATURE - F
API GRAVITY
80
70
60
50
40
30
20
10
0
0 10 20 30 40 50 60 70 80 90 100


Temperature - °F
1000
900
800
700
600
500
400
350
300
250
200
150
100
50
IBP
2
3
4
5
10
20
30
40
50
60
70
80
90
95
97
EP
Per Cent Over


Temperature - °F
1000
900
800
700
600
500
400
350
300
250
200
150
100
50
IBP
2
3
4
5
10
20
30
40
50
60
70
80
90
95
97
EP
Per Cent Over

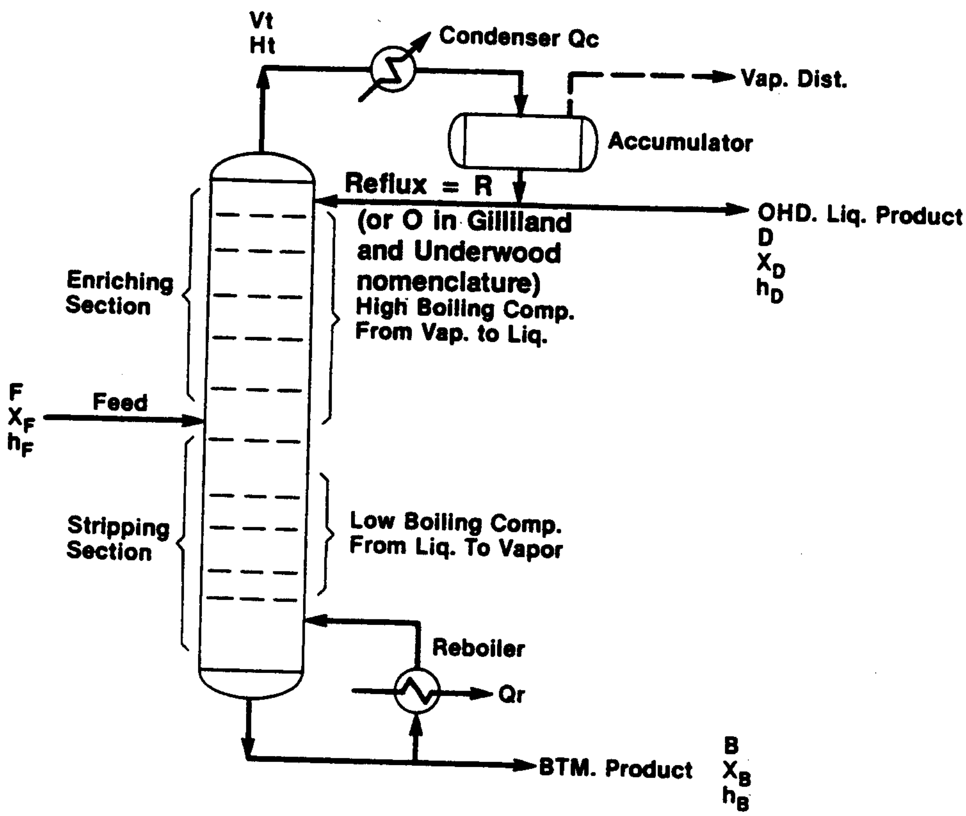
SECTION 2
DISTILLATION

GENERAL PROBLEMS
(Ref. EEDP—Section III)
Maxwell-Chapter 14

1. **Design of Fractionation Column to Make Desired Separation of Feed**
   A. **Screening**
   B. **Definitive Design**
2. **Checking a Given Column for Desired Separation of Feed**
   A. **For New Service**
   B. **For Existing Service**

   If We Can Handle the 1st Class, We Will Not Have Trouble With the Second Type.

FRACTIONATION COLUMN
(Mole Flow Rates)



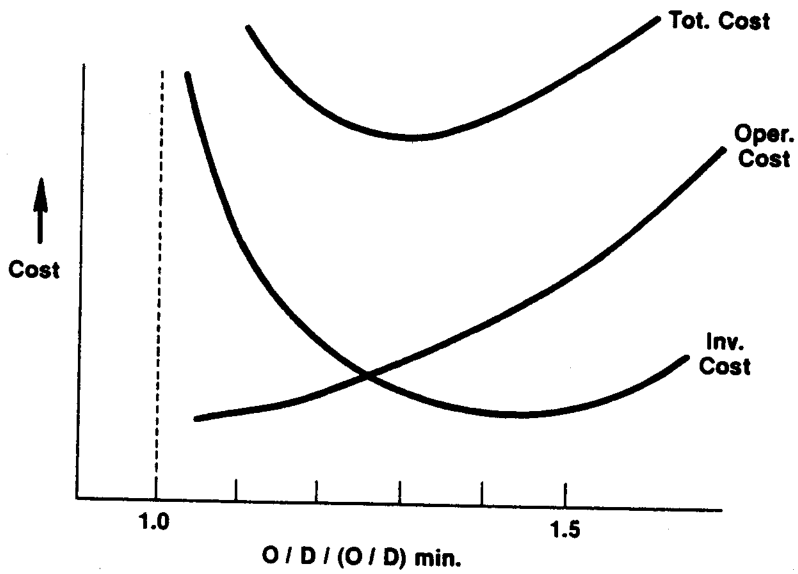
DEFINITIONS

1. Key Components
   — Major Feed Components Between Which Split is to be Made.
2. Minimum Reflux
   — Reflux Required for Desired Separation With an Infinite Number of Plates.
3. Total Reflux
   — Reflux at Infinite Reflux Ratio. Reflux Required for Separation With Minimum Number of Trays.
4. Minmum Cost Requirement
   — Cost Optimization for Best Return on Investment.

2.03

PLATES vs. REFLUX

There is an economic balance between plates and reflux.



Operating Reflux Ratio/Minimum Reflux Ratio

2.04

PLATES VS. REFLUX (Continued)

- In standard Tower Systems, Economics Not Too Sensitive. Sensitive for Complex System.

  Can Use Shortcut Procedures
  Can Use Simplified Plate-to-Plate Calc.

- With Utility Cost Going Up, the Optimum Favors More Trays and Lower Reflux.

BASIC STEPS IN COLUMN DESIGN

Know—

- Feed Rate & Composition
- Desired Separation—Recovery & Purity
- Battery Limit Conditions on Feed, Products, and Utilities

STEPS IN COLUMN DESIGN (Continued)

1. Determine Key Components Which Represent the Desired Separation
2. Determine Operating Conditions
3.* Use Short Cut Distillaton Techniques to Define the Extremes—Min. Stages & Min. Reflux
4.* Select an Operating Reflux Ratio
5.* Use Gilliland or Similar Correlation to Determine No. of Theoretical Stages
6. Perform Plate-To-Plate Calc. on Computer to Define the Tower
7. Optimize Tower Design
   —Reflux
   —Number of Stages
   —Level of Feed Preheat
   —Feed Tray Location

2.07

INITIALIZING COMPUTER PLATE-TO-PLATE CALCULATIONS TO MODEL THE TOWER

- Steps 3 to 5 May Be Eliminated by Use of Information From a Similar Operations
- PROCESS—COLUMN Has a Short Cut Distillation Procedure to Help Initialize a More Detailed Plate-To-Plate Tower Simulation. (Uses a Generalized Fenske Method)
- Addendum 2.2 Provides More Details on Short Cut Distillation Techniques

OPERATING PRESSURE

FACTORS TO CONSIDER

1. Disposition of the Overhead Product
2. Is the Overhead to be Totally Condensed?
3. Temperature of Available Cooling Medium to Overhead Condensers.

   If Cooling Water Available at 95°F is Used, the Min. Pressure for BP Distillate Would be That Corresponding to the Pressure at a BP Temperature of About 110°F (15° Approach).

OPERATING PRESSURE (Continued)

4. The Max. Pressure Might be Limited by

   A. Tower Bottoms Temperature. A Very High Pressure Could Result in Excessive Bottoms Temp. (Cracking, Color)

   B. Temperature of Heating Medium for Reboiler

   C. A Bottoms Temperature too Close to the Critical Temperature Results in Poor Separation–Bottoms Temperature Should be at Least 50°F Below T_c

OPERATING TEMPERATURE

1. Fixed Once the Separation and Pressure are Specified.
2. Temperature Estimates.
   - A. BP Calculation on OHD at Accumulator Pressure Gives Reflux Temperature for Total Condenser. Flash Calculation if Not a Total Condenser.
   - B. DP Calculation on Tower Overhead Will Give Tower Overhead Temperature.
   - C. BP Calculation on Bottoms Will Give Bottom Temperature.
   - D. Flash Calculation on Feed to Get Temperature & Fraction Vaporized.

FEED CONDITION

Factors to Consider—

1. Balancing of Tower Loadings to Take Advantage of Available Column Diameter Above and Below the Feed
2. Cost of Column Reflux (Water or Refrigeration)
3. Cost of Column Reboiling
4. Heat Available From Exchange With Products or Overhead Vapor
5. Not Unreasonable to Start With BP Liquid

RELATIVE VOLATILITY (USED TO MODEL SEPARATION)

	Comp.	K_I	$\alpha = K_I/K_C$
	A	K_A	K_A/K_C
	B	K_B	K_B/K_C
Heavy Key Component →	C	K_C	1.0
	D	K_D	K_D/K_C

$$\alpha_{AC} = \frac{K_A}{K_C} \cong \frac{P_A/\pi}{P_C/\pi} \cong P_A/P_C = \frac{\text{Vapor Pressure Component A}}{\text{Vapor Pressure Component C}}$$

MINIMUM REFLUX (UNDERWOOD'S EQN)

$$\left(\frac{O}{D}\right)_{MIN} + 1 = \sum_{i=1}^{i=N} \frac{\propto iX_{ID}}{\propto i - \theta} = \left({}^{R}/_{D}\right)_{MIN} + 1$$

Min.
Reflux
Ratio

Where: $\propto_{HK} < \theta < \propto_{LK}$

$$1 - Q = \sum_{i=1}^{i=N} \frac{\propto iZ_I}{\propto i - \theta}$$

$$\text{(O/D) MIN} = \text{Minimum Reflux} = \frac{\text{Moles Reflux}}{\text{Moles Distillate}}$$

Q = Mole Fraction Liquid in Feed
X_{ID} = Moles Fraction I in Distillate
Z_I = Mole Fraction I in Feed
$\propto_I$ = Relative Volatility of i
= $(\propto_{IB} X \propto_{IF} X \propto_{ID})^{1/3}$ ← Geometric Mean Average

MINIMUM STAGES (FENSKE EQN)

$$S_M = \frac{LN\left[\left(\frac{X_{LKD}}{X_{LKB}}\right)\left(\frac{X_{HKB}}{X_{HKD}}\right)\right]}{LN\ \alpha_{LK}}$$

Uses:
—Estimate Minimum No. of Trays Required
—Estimate Feed Tray Location

S_M = Minimum Number of Stages (At Total Reflux)
X_{LKD} = Mole Fraction Light Key in Distillate
X_{LKB} = Mole Fraction Light Key in Bottoms
X_{HKD} = Mole Fraction Heavy Key in Distillate
X_{HKB} = Mole Fraction Heavy Key in Bottoms
α_{LK} = Relative Volatility of Light Key
(Averaged Over the Entire Tower)

$$= (\underset{\substack{\uparrow \\ \text{Tower} \\ \text{Bottoms}}}{\alpha_{LKB}} \times \underset{\substack{\uparrow \\ \text{Tower} \\ \text{Feed}}}{\alpha_{LKF}} \times \underset{\substack{\uparrow \\ \text{Tower} \\ \text{Distillate}}}{\alpha_{LKD}})^{1/3} \Leftarrow \text{Geometric Mean Average}$$

2.15

ESTIMATING ACTUAL REFLUX RATE AND ACTUAL NUMBER OF STAGES

General "Rule of Thumb"—

$$\left(\frac{O}{D}\right)_{ACTUAL} = (1.15 \text{ to } 1.35) \times \left(\frac{O}{D}\right)_{MIN}$$

Use Gilliland Correlation to Calculate Actual Theoretical Stages for Selected O/D (P. 244 Maxwell). Use Safety Factor of 10% or 3 Trays, Whichever is Larger.

2.16

FEED TRAY LOCATION

1. Locate so That Ratio of Key Components in Feed Liquid is About Same as Liquid on Feed Tray. (Rule of Thumb)
2. Compare With Plate-To-Plate Printouts of Similar System
3. Use of Fenske EQN. at Total Reflux
4. Parametric Study on Plate-To-Plate Calculation (e.q. PROCESS-COLUMN) or Parametric Study Using a "Short Cut" Distillation Tower Model (e.g. PROCESS-SHORTCUT)

FEED TRAY LOCATION (MODIFIED FENSKE EQN.)

$$S_{AF} = \frac{LN\left[\left(\frac{X_{LKD}}{X_{HKD}}\right)\left(\frac{X_{HKF}}{X_{LKF}}\right)\right]}{LN\ \alpha'_{LK}}$$

Approximate Feed Tray Location Set by the Ratio– $\frac{S_{AF}}{S_M}$

S_{AF} = Number of Stages Above Feed (At Total Reflux)
X_{LKD} = Mole Fraction Light Key in Distillate
X_{LKF} = Mole Fraction Light Key in Feed
X_{HKD} = Mole Fraction Heavy Key in Distillate
X_{HKF} = Mole Fraction Heavy Key in Feed
α'_{LK} = Relative Volatility of Light Key (Averaged Over the Enriching Section of the Tower)
= $(\alpha_{LKF} \times \alpha_{LKD})^{1/2}$

$$(\underbrace{\alpha_{LKF}}_{\text{Tower Feed}} \times \underbrace{\alpha_{LKD}}_{\text{Tower Distillate}})^{1/2}$$

FEED TRAY LOCATION (KIRKBRIDE METHOD)

This is the Correlation Used by the Shortcut Distillation Technique in PROCESS/PRO II

$$\log \frac{S_{AF}}{S_{BF}} = 0.206 \log \left[{}^{B}/_{D} \left(\frac{X_{HKF}}{X_{LKF}}\right)\left(\frac{X_{LKB}}{X_{HKD}}\right)^{2} \right]$$

$$S_{AF}/S_{BF} = (K_1)^{0.206} \text{ where } K_1 = \left[{}^{B}/_{D} \left(\frac{X_{HKF}}{X_{LKF}}\right)\left(\frac{X_{LKB}}{X_{HKD}}\right)^{2} \right]$$

COMPARISON OF KIRKBRIDE METHOD WITH FENSKE METHOD: (MOLE FLOW RATES AND COMPOSITIONS)

$X_{LKF}=0.25$ $\alpha_{LK}=1.41$ (Entire Tower) $X_{LKD}=0.60$ $X_{LKB}=0.03$
$X_{HKF}=0.05$ $\alpha_{LK}=1.50$ (Enriching Section) $X_{HKD}=0.03$ $X_{HKB}=0.10$
$B/D=1.53$

FENSKE METHOD

$S_M=12.22$ $S_{AF}=3.42$

$S_{BF}=S_M-S_{AF}=8.80$

$$S_{AF}/S_M=\frac{3.42}{12.22}=0.28$$

$$S_{AF}/S_{BF}=\frac{3.42}{8.80}=\underline{0.39}$$

KIRKBRIDE METHOD

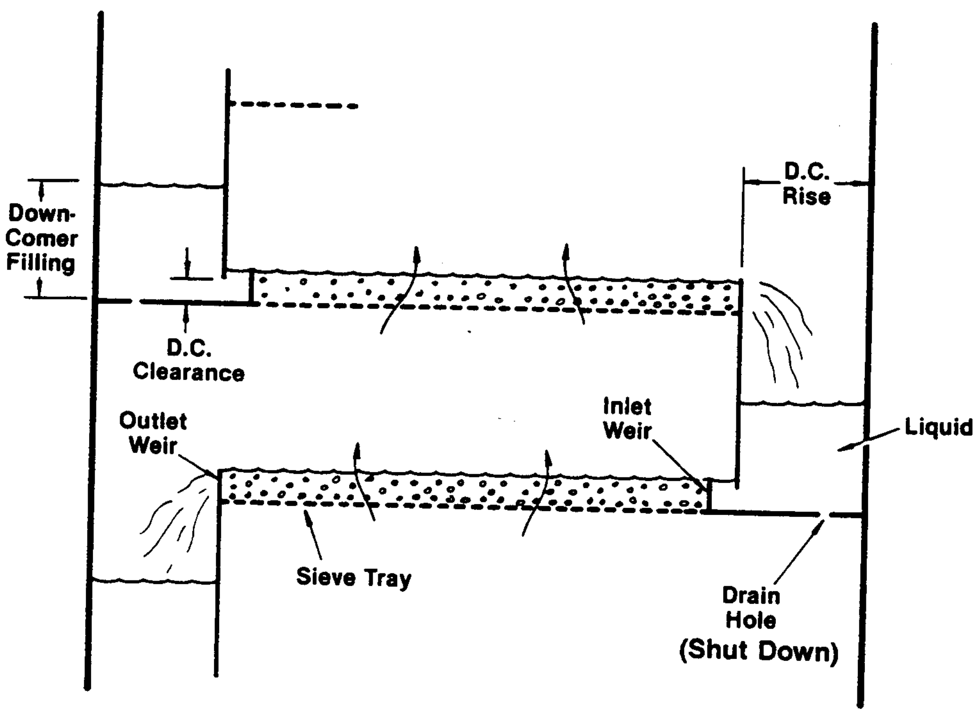
$$\log\frac{S_{AF}}{S_{BF}}=0.206\log[(1.53)(0.20)(1.0)^2]$$

$$S_{AF}/S_{BF}=\underline{0.78}$$

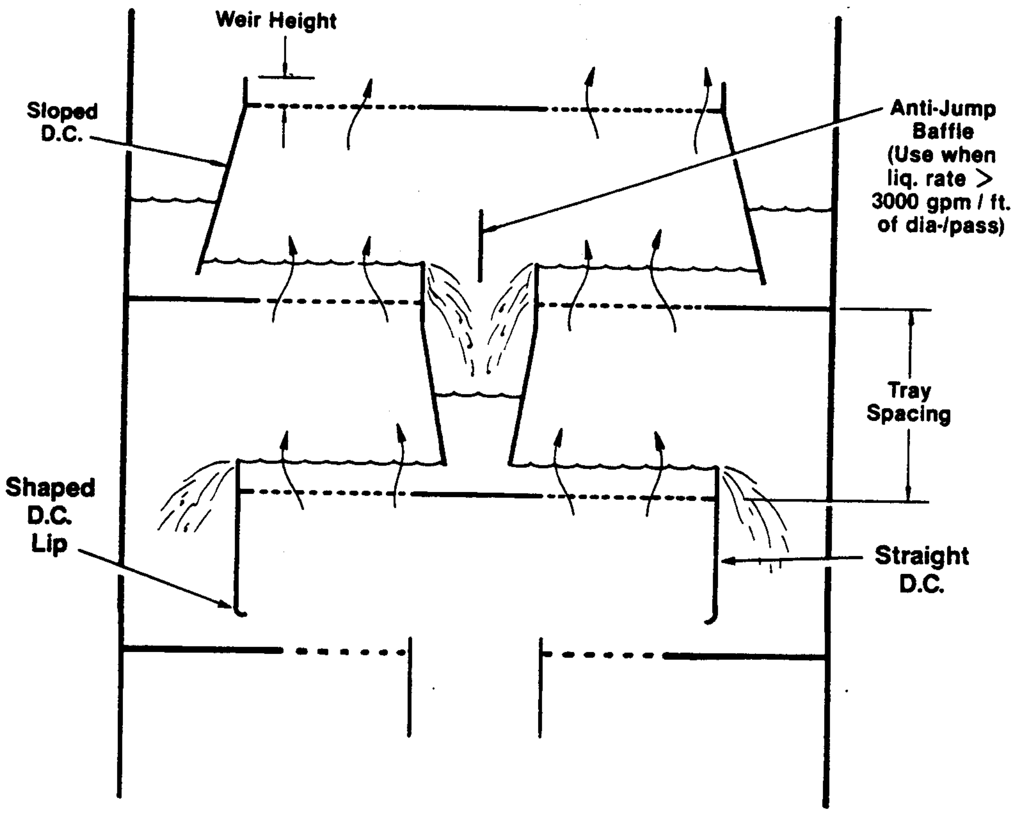
TRAY CHARACTERISTICS
(REF. EEDP—SECTION III-A, TABLE 1)

Type	Capacity	Efficiency	Cost	Flex.
Sieve	Med. to High	High	Lowest	Med. to Good 3/1
B.C.	Low to Med.	Med. to High	High 200% Sieve	Med. to Good 3/1
Valve	Good as Sieve	Good as Sieve	Med. 10–20% > Sieve	High Up to 5/1
Jet	Highest-HP Med.-LP	Low to Med.	5% > Sieve	Low. 1.5/1 or 2/1

SINGLE PASS

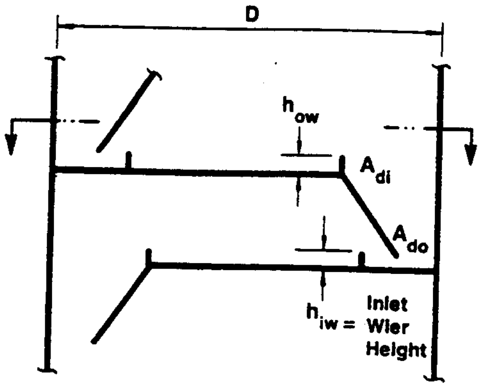


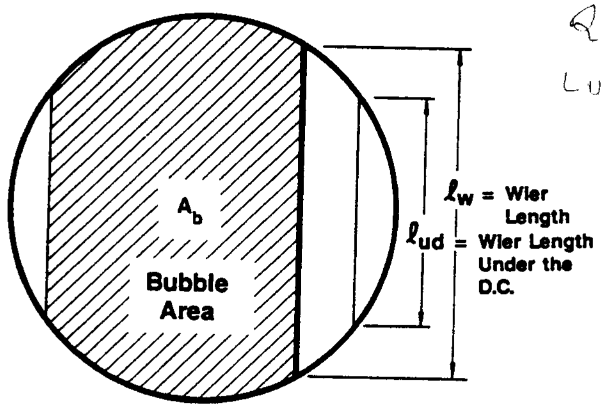
TWO-PASS TRAY



2.23

TRAY NOMENCLATURE — SINGLE PASS TRAYS
(SLOPED DOWNCOMERS SHOWN)





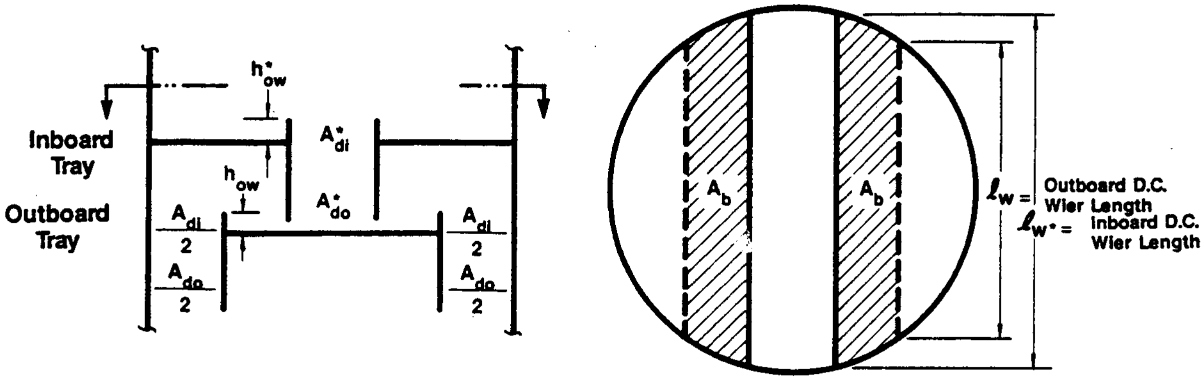
$$A_b = A_s - A_{di} - A_{do} - A_w$$

$$A_n = A_s - \frac{A_{di} + A_{do}}{2} - A_w$$

NOMENCLATURE

A_n = Free (Net) Area → Relates to Jet Flood Point
A_b = Bubble Area
A_s = Tower Cross Sectional Area
A_{di} = Down Comer Inlet Area
A_{do} = Down Comer Outlet Area
A_w = Waste Area

TRAY NOMENCLATURE — TWO PASS TRAYS
(CHORDAL DOWNCOMERS SHOWN)



$$A_b = A_s - A^*_{di} - A_{do} - A_w$$

~~For outboard tray~~, $A_n = A_s - \frac{A^*_{di} + A^*_{do}}{2} - A_w$

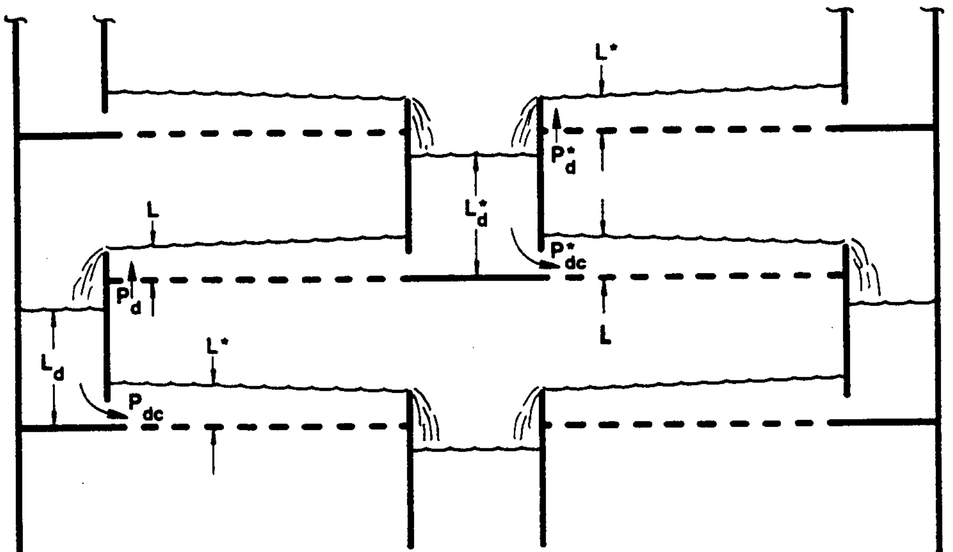
~~For inboard tray~~, $A_n = A_s - \frac{A_{di} + A_{do}}{2} - A_w$

For Design Calcs Use the Smaller of the Two.

$*$ → Inboard Tray

2.25

PRESSURE BALANCE FOR A TWO PASS SIEVE TRAY



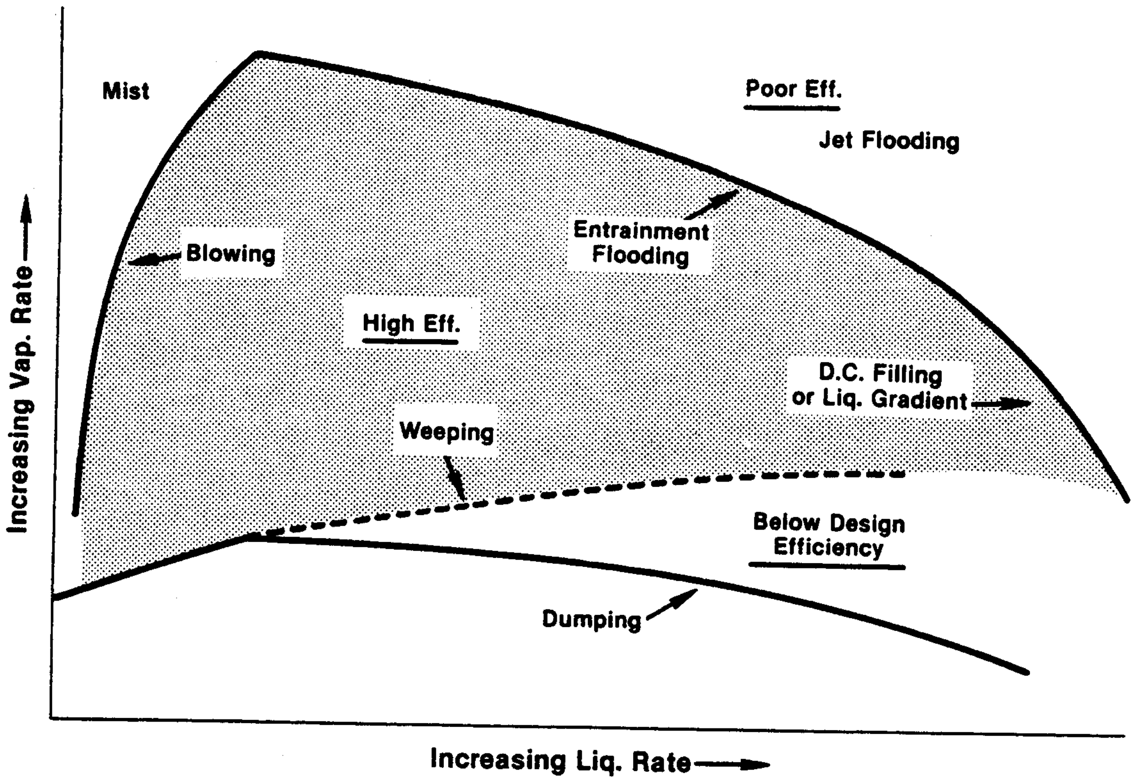
Pressure Balance for Inboard Downcomer Filling:

$$L^*_d = P^*_{dc} + P^*_d + L^* + L$$

Pressure Balance for Outboard Downcomer Filling:

$$L_d = P_{dc} + \underbrace{P_d}_{\text{Total Tray Pressure Drop}} + L + L^*$$

TYPICAL TRAY PERFORMANCE DIAGRAM



2.27

TOWER SIZING

For Sieve Trays

A. Calc. a Trial Tower Diameter With Assumed Tray Spacing
B. Check Liquid Carrying Capcity
C. Repeat (A) if Necessary
D. Size D.C. & Set Other Tray Details
E. Calculate Tray Hydraulics
F. Calculate Perforated Area
G. Calculate Flexibility
H. Calculate Efficiency

2.28

COMPUTER PROGRAM SELECTION FOR PLATE-TO-PLATE CALCULATIONS

- PROCESS-COLUMN or COPE-TOWER
  - — Good for Simple and Complex Systems
  - — Direct Access to EDL
  - — Can Set Product Specs
  - — Can Run With Minimum Data
  - — Maybe Difficult to Converge With Complex Systems Depending on Nature of the Complex System, Either TOWER or COLUMN May be Preferred
- ASPECT II
  - — Fast
  - — Can Use Steam Stripping
  - — Wider Range of Temperature and Pressure
  - — Good for Heavy Hydrocarbon Systems (Vacuum Pipe Stills)
- CHEMDIST
  - — For Severely Non-Ideal Systems
  - — Limited in Number of Components

2.29

USE OF PACKINGS

- Vacuum Systems Where Low ΔP is Critical
- Debottlenecking Existing Trayed Towers
- Light Ends Towers at High Liquid Loadings
- Absorption and Stripping Services (High Liquid Loadings)
- Small Diameter Towers (Tower Diameters Under About Three Feet)
- Corrosive Services Where Ceramic Materials or Plastic Materials are Attractive. Corrosive Services Where Packing Cost is Less Than Alloy Tray Cost

2.30

PACKING CHARACTERISTICS AND TYPES
(REF. EE DP-SECTION III-A, TABLE 2)

Type	Capacity	Efficiency	Cost	Flex.
Random Dumped Packing (eg. Pall Rings, Metal Intallox, Nutter Rings, etc.)	Med.	Med. to High	Med. to Low	Med. to Good > 3/1
Structured Packing (eg. Flexipac, Montz, Gempak, etc.)	Med. to Very High	Med. to Very High	High (at Least 2X RDP Cost)	Med. to Good > 3/1
Grids (eg. Glitsch Grid, Flexigrid, Snapgrid)	Very High	Poor (Good For Entrain-ment Removal and Heat Transfer)	Med. to High	Low < 2/1

Based on Efficiency, Capacity, Turndown and Cost, the Two Inch Metal Pall Ring is an Economic Packing for Most General Petroleum Distillation Services.

Section 2
DISTILLATION

Nomenclature

A = Area, ft^2
A_b = Bubble area, ft^2
A_c = Downcomer clearance area, in^2
A_{di} = Downcomer inlet area, ft^2
A_{do} = Downcomer outlet area, ft^2
A_f = Average tower free area, ft^2 (superficial area minus arithmetic average of inlet and outlet areas of downcomer (s) above the tray minus the waste area); for multipass trays, use smallest value of A_f.
A_o = Hole area, ft^2
A_s = Superficial (total) tower area, ft^2
A_w = Waste area, ft^2 (normally zero for sieve trays)
c = Clearance between tray and downcomer at tray inlet, inches
D = Diameter, ft
D_t = Trial diameter, ft
E_o = Overall efficiency, %
H = Tray spacing, ft
h_c = Clear liquid height on tray, inches of hot liquid
h_d = Downcomer filling, inches of hot liquid
h_{ed} = Effective dry tray pressure drop, inches of hot liquid
h_i = Tray inlet head, inches of hot liquid
h_t = Total tray pressure drop, inches of hot liquid
h_{ud} = Head loss under downcomer, inches of hot liquid
h_{wi} = Inlet weir height, inches
h_{wo} = Outlet weir height, inches
K_{HL} = Tray spacing—liquid rate capacity factor, dimensionless
K_{hp} = Liquid height—pressure drop factor, dimensionless
$K_{\sigma\mu}$ = Surface tension—viscosity capacity factor, dimensionless
$K_{\phi p}$ = Hole diameter—pressure drop factor, dimensionless
K_1 = Clear liquid height—density difference weeping factor, dimensionless
K_2 = Tray spacing—percent hole area weeping factor, dimensionless
K_3 = Surface tension—hole diameter weeping factor, dimensionless
L = Liquid rate, gph/ft of weir per pass
L' = Liquid rate, gph/ft of diameter per pass
L_L = Liquid load, ft^3/s at conditions
$L_{L(Min)}$ = Minimum liquid load, ft^3/s at conditions

Nomenclature (Continued)

ℓ_{fp} = Flow path length (distance between inlet and outlet downcomers), feet

ℓ_i = Inlet weir length, inches

ℓ_o = Outlet weir length, inches

ℓ_{ud} = Length of bottom edge of downcomer, inches

M_L = Liquid loading, lb/s

M_v = Vapor loading, lb/s

N_p = Number of liquid passes

Q_L = Liquid rate, gpm at conditions

V_{di} = Allowable velocity of clear liquid in downcomer inlet, ft/s

V_f = Vapor velocity based on average tower free area, ft/s

V_L = Design vapor load = $\left(\frac{ft^3}{s}\right)\sqrt{\frac{\rho_V}{\rho_L - \rho_V}}$ at conditions

$V_{L(Lim)}$ = Ultimate vapor load dependent on system properties, ft^3/s at conditions

V_o = Vapor velocity through holes, ft/s

$V_{o(Min)}$ = Vapor velocity through holes at minimum vapor rate, ft/s

$V_{o(Weep)}$ = Calculated vapor velocity through holes at which tray will start to weep, ft/s

β = Capacity factor, $1.4\sqrt{\frac{\rho_L - \rho_V}{\rho_V}}$

μ_L = Liquid viscosity at conditions, cP

ρ_L = Liquid density at conditions, lb/ft^3

ρ_V = Vapor density at conditions, lb/ft^3

σ_L = Liquid surface tension at conditions, mN/m

σ_{STD} = Standard liquid surface tension, mN/m

= $10^{\left(1.68 - \frac{0.244}{(\mu_L)^{0.55}}\right)}$

ϕ = Hole diameter, inches

Section 2
DISTILLATION

In order to simplify the work involved in making stepwise calculations for the rectification of binary and multicomponent systems, Gilliland[1] has presented an empirical correlation between theoretical steps and reflux ratio. To use the Gilliland correlation to predict the number of theoretical plates for a given reflux ratio, the minimum number of steps at total reflux and the minimum reflux ratio are required.

Minimum Number of Theoretical Steps

When a separation is specified with respect to only two components of a multi-component mixture, the lower boiling of these two components is designated the *light key component* and the higher boiling the *heavy key component*, and the minimum number of steps can be calculated by the well-known Fenske equation[2] as follows:[3]

$$S_M = \frac{\log\left(\frac{X_{LKD}}{X_{LKW}}\right)\left(\frac{X_{HKW}}{X_{HKD}}\right)}{\log \alpha_{LK}} \tag{1}$$

or

$$[\alpha_{LK}]^{S_M} = \left(\frac{X_{LKD}}{X_{LKW}}\right)\left(\frac{X_{HKW}}{X_{HKD}}\right) \tag{1a}$$

After equation (1) is solved for S_M, the latter may be substituted in this equation along with the distribution of either key component to predict[4] the distribution of the other components, or

$$\log\left(\frac{X_{LD}}{X_{LKW}}\right)\left(\frac{X_{HKW}}{X_{HKD}}\right) = S_M \log \alpha_L \tag{2}$$

Likewise,

$$\log\left(\frac{X_{HW}}{X_{HD}}\right)\left(\frac{X_{LKD}}{X_{LKW}}\right) = S_M \log\left(\frac{\alpha_{LK}}{\alpha_H}\right) \tag{3}$$

In any of the above equations, moles per 100 moles of feed may be replaced by total moles, or volume or weight units since in any of these conversions the multiplying factors cancel out.

When the degree of separation is specified for more than two components, equation (1) must be applied to all critical combinations of these components and the maximum S_M determined for the most difficult case. If the separation is specified with respect to the total quantity of two or more components, as in the case of Example 1, trial and error is required for the solution of S_M.

It should be pointed out that the concentrations calculated by equations (2) and (3) actually apply only to the separation at total reflux and, with the exception of the two key components, there will be some variation of the degree of separation with the reflux ratio. As the reflux ratio decreases, there is some improvement in separation between light and heavy components boiling outside the range of the key components and some deterioration in the separation of components boiling intermediate between the key components. However, in so far as the present procedure is concerned, the distillate and bottoms compositions for other reflux ratios are assumed to be the same as those calculated for total reflux.

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[1]Gilliland, *Ind. Eng. Chem. 32,* 1220 (1940).
[2]Fenske, *Ind. Eng. Chem. 24,* 482 (1932).
[3]A table of nomenclature is given on page 10.
[4]This equation may be used for any pair of components.

Minimum Reflux Ratios

Gilliland[5] has proposed several different formulas for predicting minimum reflux ratio and all have the disadvantage of being composed of a number of complex terms in addition to requiring trial and error for solution. Although all these equations appear to give satisfactory results, the terms are so complex that it is difficult to be certain that there are no numerical errors in their application.

In order to apply the Gilliland method with greater facility, the following equation was developed for predicting the minimum reflux ratio of a multicomponent system:

$$(O/D)_M + 1 = \left(\frac{\alpha_{LK} l_{LK} + 1}{\alpha_{LK} - 1}\right)\left(\frac{X_{LKD}}{l_{LK}} - X_{HKD}\right)$$

$$+ \sum_L \frac{\alpha_L}{\alpha_L - 1}\left(X_{LD} - l_L X_{XKD}\right) + \sum_H \frac{\alpha_H}{\alpha_{LK} - \alpha_H}\left(\frac{X_{LKD}}{l_H} - Z_{HD}\right) \qquad (4)$$

$(O/D)_M$ can be calculated for two arbitrary states of feed vaporization:

1. "Liquid" feed, corresponding to vaporization of the feed equivalent to the fraction of the feed lighter than the light key component. For the components lighter than the light key, $l_L = Z_L/\alpha_L$ and for the light key and heavier components, $l_{LK} = Z_{LK}$, and $l_H = Z_H$.[6]

2. "Vapor" feed, corresponding to vaporization of the feed equivalent to the fraction of the feed consisting of the heavy key component and lighter. For the components lighter than the heavy key, $l_L = Z_L/\alpha$ and $l_{LK} = Z_{LK}/\alpha_{LK}$ and for the components heavier than the heavy key, $l_H = Z_H$.[6]

After the minimum reflux ratios have been calculated for "liquid" and "vapor" feeds, the minimum reflux ratio for the actual vaporization of the feed can be calculated by direct interpolation or extrapolation. However, extrapolation beyond 50% of the difference between "liquid" and "vapor" feed may lead to serious deviations.

The first term of the right-hand side of equation (4) is the same as for binary mixtures, and the equation reduces to the equivalent of a binary mixture when all light components other than the light key have infinite volatility and all heavy components other than the heavy key have zero volatility. Under these circumstances the equation is exact when l_{LK} is taken as the ratio of the two components in the liquid phase of the feed. That is, if the feed is introduced as a liquid at its bubble point, $l_{LK} = Z_{LK}$, which is the ratio of the two components in the feed, if the feed is introduced as a vapor at its dewpoint, $l_{LK} = Z_{LK}/\alpha_{LK}$, which is the ratio of two components in the equilibrium liquid. For intermediate stages of vaporization l_{LK} can be calculated from the flash vaporization formula, although direct interpolation of the minimum reflux ratio of the basis of percentage vaporization between the saturated liquid and saturated vapor feeds gives values only slightly in error on the conservative side.

In the case of multicomponent mixtures, equation (4) is semi-empirical since it was necessary to make simplifing approximations in its derivation. Furthermore, the exact values of the various l's cannot be calculated directly from the composition and state of vaporization of the feed, since the liquid on the feed plate is not identical to the liquid phase of the feed as in the case of a binary mixture. As a result, it was necessary to define the l's empirically for two states of feed vaporization, arbitrarily chosen to simulate a binary mixture of the two key components, and then interpolate or extrapolate to the minimum reflux ratio corresponding to the actual vaporization of the feed.

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[5]Gilliland, *Ind. Eng. Chem. 32,* 1001 (1940).

[6]If components, intermediate between the two key components, are present, they are considered either *light* or *heavy* components depending upon which key their volatility more nearly approaches. In the case of "liquid" feed, $l_L = Z_L$ and $l_H = Z_H$ for these intermediate components; in the case of "vapor" feed, $l_L = Z_L/\alpha_L$ and $l_H = Z_{H\alpha H}/\alpha_{LK}$.

Equation (4) has been checked for a number of multicomponent systems on which the minimum reflux ratio was determined by stepwise trial and error calculations. Generally, unusual systems were chosen with respect to composition and relative volatility in order to reveal the maximum deviations ever likely to be encountered in practice. The agreement was quite satisfactory as the average deviation was less than $\pm 5\%$ and the maximum about 10%. The latter occurred at the limit of extrapolation relative to the arbitrary feed states.

Also, the minimum reflux ratio was calculated for these same systems by the Colburn method[7] with about the same degree of accuracy. It should be pointed out that the latter gave better results than equation (4) when the relative volatilities and compositions were not so abnormal as the system selected. However, under these circumstances both methods were quite accurate as the deviations seldom exceeded a few percent, and the present equation has a distinct advantage in that it is explicit and does not require trial and error.

Both methods are quite sensitive to the selection of key components, and the selection of the wrong key components can lead to a much greater error than is inherent in either method. If the desired separation is between adjacent components, there is usually no doubt about selecting these as the key components. However, if there are additional specifications relative to other components, it may be necessary to try two or more combinations of key components to make sure that the minimum reflux ratio is sufficient to fulfull all specified conditions.

Correlation of Theoretical Steps With Reflux Ratio

As mentioned at the beginning of this section, Gilliland correlated the results of a large number of stepwise calculations on various binary and multicomponent mixtures by plotting

$$[S - S_N] / [S + 1] = \phi(S) \text{ against } [(O/D) - (O/D)_N] / [O/D + 1] = F(O/D)$$

and found that all points could be represented by a single curve irrespective of the type or degree of separation. These points, along with about half again as many additional points, were replotted, and the best curve through them was essentially the same as Gilliland's original correlation.

In arriving at the coordinates for the additional points the minimum reflux ratio was calculated by equation (4); therefore these points are a criterion of the present method as well as the curve itself. In no case did the deviations exceed either 3 theoretical steps or 15%, and the average deviation was less than 1 theoretical step and also less that $\pm 5\%$. To take care of the maximum deviation it is recommended that in any design the number of theoretical steps predicted from the correlation on page 13 be increased by either 3 theoretical steps or 10%, whichever is greater.

Plate Efficiency

Because of the large number of factors which undoubtedly influence the plate efficiency of a fractionating tower, any fundamental formula accounting for even the most important variables must necessarily be quite involved. For this reason, a simple empirical correlation of the limited data on hydrocarbon mixtures seemed to be the most promising method of predicting plate efficiency.

Gunness[8] correlated the results of several tests on petroleum mixtures on the basis of vapor pressure of the liquid. As he points out, this is a method of indirectly correlating plate efficiency with liquid viscosity since viscosity of pure hydrocarbons and narrow boiling fractions is an approximate function of vapor pressure over a fairly wide range of vapor pressures.

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[7]Colburn, *Trans. Am. Inst. Chem. Engrs. 37,* 805 (1941).
[8]Gunness, *Sc.D. Thesis, Mass, Inst. Tech.* (1936).

In view of the consistent results obtained by Gunness, plate efficiency was plotted directly against fluidity (reciprocal viscosity) for a number of tests on commercial towers including those upon which Gunness based his curve. The curve on page 12 represents this correlation. While the over-all plate efficiency exceeds 100% at fluidities greater than 9 cm^{-1}, this is not inconsistent as the flow of the liquid across the plates results in concentration gradients which may achieve a greater degree of fractionation than predicted by stepwise calculations in which the liquid is assumed to leave the plate in equilibrium with the composite vapor. Lewis[9] has shown theoretically that different combinations of liquid and vapor concentration gradients across the plate may give over-all plate efficiencies as high as 200-300% when based on stepwise calculations.

There is no reason to believe that this correlation applies to mixtures other than hydrocarbons, and with the exception of alcohol-water mixtures there are too little data available to afford a comparison. Although there is considerable variation in the alcohol-water data, there is some indication that plate efficiencies are somewhat greater than for hydrocarbons of the same viscosity.

Location of the Feed Plate

As a simple approximation for locating the feed plate, it may be assumed that the proportion of actual plates above the feed will be the same as that required to effect the same separation between the key components at total reflux. That is, the number of theoretical steps at total reflux is calculated for the concentration then assumed that the ratio of this to the total number of theoretical steps at an infinite reflux ratio is the same as the ratio of actual plates above the feed is to the total number of plates. Application of this method is illustrated by Example 1.

In some cases where there are critical components other than the two key components, it may be necessary to check the total reflux steps above and below the feed on the basis of these components, since the optimum location of the feed plate will be different with each pair of components. Usually the separation of components other than the key components is so complete that only the latter need be considered.

Example

At an operating pressure of 100 psig determine the number of plates and reflux ratio required to separate the mixture given below so that the bottoms contain at least 90% of the butenes-2 present in the feed and at the same time have an isobutene content not greater than 5%:

Component	Feed (Mole %)
$i\text{-}C_4H_{10}$	40.0
$i\text{-}C_4H_8$	20.0
$C_4H_8\text{-}1$	15.0
C_4H_{10}	5.0
$t\text{-}C_4H_8\text{-}2$	10.0
$c\text{-}C_4H_8\text{-}2$	10.0
	100.0

[9] Lewis, *Ind. Eng. Chem. 28,* 399 (1936).

(1) Dewpoint of Distillate and Bubble Point of Bottoms

In order to calculate the average volatilities, the dewpoint of the distillate and bubble point of the bottoms must be found by trial and error using assumed compositions. These are tabulated below.

Component	Moles Per 100 Moles of Feed			Mole Fraction	
	Feed	Distillate	Bottoms	Distillate	Bottoms
i-C_4H_{10}	40.0	39.3	0.7	0.530	0.027
i-C_4H_8	20.0	18.7	1.3	.253	.050
C_4H_8-1	15.0	13.0	2.0	.176	.077
C_4H_{10}	5.0	1.0	4.0	.014	.154
t-C_4H_8-2	10.0	1.5	8.5	.020	.327
c-C_4H_8-2	10.0	0.5	9.5	.007	.365
	100.0	74.0	26.0	1.000	1.000

As a first trial, assume the dewpoint of the distillate is 140°F at 7.8 atm (114.7 psia).

Component	y_D	First Trial		
		α'_D* 140°F	F** 140°F	I $\pi y/F$
i-C_4H_{10}	0.530	1.29	8.4	0.493
i-C_4H_8	.253	1.155	7.5	.263
C_4H_8-1	.176	1.13	7.35	.187
C_4H_{10}	.014	1.00	6.5	.017
t-C_4H_8-2	.020	0.97	6.3	.025
c-C_4H_8-2	.007	0.91	5.9	.009
	1.000			0.994

*Relative volatilities to C_4H_{10} or (α')'s are used as a matter of convenience; then, the (α'_{av})'s are converted to (α_{av})'s, the relative volatilities to t-C_4H_8-2, which will be selected as the heavy key component.

**Computed from the fugacity function of butane multiplied by the relative volatilies.

Since the sum of the x's is 0.994 instead of 1.000, the assumed temperature should be lower slightly, but the difference would be so small (less than 1°F) that the change in the (x'_D)'s would be imperceptible. Consequently, 140°F will be used as the dewpoint of the distillate.

The bubble point of the bottoms is assumed to be 165°F at 8.0 atm[10] for the first trial.

Component	X_w	First Trial			Second Trail		
		$\alpha'w^*$ 165°F	F** 165°F	y Fx/π	$\alpha'w^*$ 160°F	F** 160°F	y
$i\text{-}C_4H_{10}$	0.027	1.26	10.7	0.036	1.265	10.25	0.035
$i\text{-}C_4H_8$	.050	1.14	9.7	.061	1.145	9.3	.058
$C_4H_8\text{-}1$	.077	1.115	9.5	.091	1.12	9.1	.088
C_4H_{10}	.154	1.00	8.5	.164	1.00	8.1	.156
$t\text{-}C_4H_8\text{-}2$	.327	0.97	8.25	.337	0.97	7.85	.321
$c\text{-}C_4H_8\text{-}2$	.365	0.915	7.8	.356	0.915	7.4	.338
	1.000			1.045			0.996

*Relative volatilities to C_4H_{10} or (α')'s are used as a matter of convenience; then, the (α'_{av})'s are converted to (α_{av})'s, the relative volatilities to $t\text{-}C_4H_8\text{-}2$, which will be selected as the heavy key component.

**Computed from the fugacity function of butane multiplied by the relative volatilies.

The bubble point of the bottoms will be taken as 160°F. The relative volatilities are averaged and converted to $t\text{-}C_4H_8\text{-}2$ as the heavy key in the following table:

Component	α'_D 140°F 7.8 atm	α'_W 160°F 8.0 atm	α'_A 150°F 7.9 atm	α'_{av} $(\alpha'_D\alpha'_W\alpha'_A)^{1/2}$	α_{av}
$i\text{-}C_4H_{10}$	1.29	1.265	1.275	1.275	1.315
$i\text{-}C_4H_8$	1.155	1.145	1.15	1.15	1.185
$C_4H_8\text{-}1$	1.13	1.12	1.125	1.125	1.16
C_4H_{10}	1.00	1.00	1.00	1.00	1.03
$t\text{-}C_4H_8\text{-}2$	0.97	0.97	0.97	0.97	1.00
$c\text{-}C_4H_8\text{-}2$	0.91	0.915	0.91	0.91	0.94

(2) Minimum Theoretical Steps (Total Reflux)

The minimum number of theoretical steps by which the desired separation can be accomplished is calculated as follows:

Let t = moles of $t\text{-}C_4H_8\text{-}2$ in the distillate per 100 moles of feed

$10 - t$ = moles of $t\text{-}C_4H_8\text{-}2$ in the bottoms per 100 moles of feed

Since 90% of the butenes-2 must be retained in the bottoms, the cis-butene-2 content of the distillate and bottoms will be:

and
$(2 - t)$ moles in the distillate per 100 moles of feed
$(8 + t)$ moles in the bottoms per 100 moles of feed

[10]After allowing 3 lb/sq in. as the approximate pressure drop through the tower.

Using the previously assumed values of 18.7 moles of isobutene in the distillate and 1.3 moles in the bottoms, the following equations must be satisfied:

$$\left(\frac{18.7}{1.3}\right)\left(\frac{10-t}{t}\right) = (1.185)^{S_M}$$

$$\left(\frac{18.7}{1.3}\right)\left(\frac{8+t}{2-t}\right) = \left(\frac{1.185}{0.94}\right)^{S_M}$$

A trial and error solution of these equations shows that they are satisfied by $S_M = 25.5$ and $t = 1.62$.

The distribution of the other components can be calculated from S_M and the distribution of t-C_4H_8-2.

t-C_4H_{10}: Let u = moles of t-C_4H_{10} in bottoms

$$\left(\frac{40-u}{u}\right)\left(\frac{8.38}{1.62}\right) = (1.315)^{25.5} = 1075$$

$$= 0.19 \text{ moles of t-}C_4H_{10} \text{ in the bottoms}$$

C_4H_8-1: Let v = moles of C_4H_8-1 in the bottoms

$$\left(\frac{15-v}{v}\right)\left(\frac{8.38}{1.62}\right) = (1.16)^{25.5} = 44$$

$$v = 1.58 \text{ moles of } C_4H_8\text{-2 in the bottoms}$$

C_4H_{10}: Let w = moles of C_4H_{10} in the bottoms

$$\left(\frac{5-w}{w}\right)\left(\frac{8.38}{1.62}\right) = (1.03)^{25.5} = 2.12$$

$$w = 3.55 \text{ moles of } C_4H_{10} \text{ in the bottoms}$$

The percentage of t-C_4H_8 in the bottoms will be:

$$\left(\frac{1.3}{0.19+1.3+1.58+3.55+8.38+9.62}\right) = 100 = 5.3\%$$

In order to meet a maximum of 5.0% t-C_4H_8 specified for the bottoms, it is necessary to reduce the 1.3 moles to 1.22 moles in the bottoms. This would require an increase in S_M to 25.8 which would modify the distribution of the other components. However, the latter change is so slight that it can be neglected. The composition of the overhead and bottoms will then be:

Component	Moles Per 100 Moles of Feed			Mole Fraction	
	Feed	Distillate	Bottoms	Distillate	Bottoms
i-C_4H_{10}	40.0	39.81	0.19	0.528	0.008
i-C_4H_8	20.0	18.78	1.22	.229	.050
C_4H_8-1	15.0	13.42	1.58	.178	.064
C_4H_{10}	5.0	1.45	3.55	.019	.145
t-C_4H_8-2	10.0	1.62	8.38	.021	.342
c-C_4H_8-2	10.0	0.38	9.62	.005	.391
		75.46	24.54	1.000	1.000

(3) Minimum Reflux Ratio

Since the critical separation is between isobutene and the butenes-2, the former is naturally selected as the light key component and trans-butene-2, since it is more volatile than the cis-butene-2, as the heavy key component. Butene-1 is considered a light intermediate component because of the proximity of its relative volatility to that of isobutene; normal butane is considered a heavy intermediate component since its relative volatility is nearer to the heavy key than the light key. The following tabulation gives the necessary information for calculating the minimum reflux ratios for the two arbitrary states of feed vaporization:

Component	Type	Mole Fraction			α_{av}	I	
		Feed	Distillate	Bottoms		"Liquid" Feed	"Vapor" Feed
$i\text{-}C_4H_{10}$	L	0.400	0.528	0.008	1.315	3.04	3.04
$I\text{-}C_4H_8$	LK	.200	.249	.050	1.185	2.00	1.69
$C_4H_8\text{-}1$	L	.150	.178	.064	1.16	1.50	1.29
C_4H_{10}	H	.050	.019	.145	1.03	4.00	3.48
$t\text{-}C_4H_8\text{-}2$	HK	.100	.021	.342	1.00	—	—
$c\text{-}C_4H_8\text{-}2$	H	.100	.005	.391	0.94	2.00	2.00
		1.000	1.000	1.000			

"Liquid" feed—40% vaporized

$$(O/D)_N + 1 = \left(\frac{1.185 \times 2.00 + 1.0}{1.185 - 1.0}\right)\left(\frac{0.249}{2.00} - 0.021\right)$$

$$+ \left(\frac{1.315}{0.315}\right)(0.528 - 3.04 \times 0.021) + \frac{1.16}{0.16}(0.178 - 1.50 \times 0.021)$$

$$+ \frac{1.03}{1.185 - 1.03}\left(\frac{0.249}{4.00} - 0.019\right) + \frac{0.94}{1.185 - 0.94}\left(\frac{0.249}{2.00} - 0.05\right)$$

$$(O/D)_M = 1.88 + 1.94 + 1.07 + 0.29 + 0.46 - 1 = 4.64$$

"Vapor" feed—90% vaporized

$$(O/D)_N + 1 = \frac{1.185 \times 1.69 + 1.0}{1.185 - 1.0}\left(\frac{0.249}{1.69} - 0.021\right)$$

$$+ 1.94 + \frac{1.16}{0.16}(0.178 - 1.29 \times 0.021)$$

$$+ \frac{1.03}{1.185 - 1.03}\left(\frac{0.249}{3.48} - 0.019\right) + 0.46$$

$$(O/D)_M = 2.06 + 1.94 + 1.10 + 0.35 + 0.46 - 1 = 4.91$$

Assume that the feed is sufficiently preheated to vaporize a percentage equivalent to the distillate or 75.46%. By interpolation, the minimum reflux ratio corresponding to this feed vaporization is:

$$(O/D)_M = 4.64 + \left(\frac{75.46 - 40}{90 - 40}\right)(4.91 - 4.64) = 4.83$$

(4) Theoretical Steps vs Reflux Ratio

Using the values determined in preceding sections for minimum theoretical steps, 25.8, and for minimum reflux ratio, 4.83, the number of theoretical steps for various reflux ratios can be predicted from the correlation on page 13.

O/D	F(O/D)	ϕ(S)	S	Theoretical Plates*
4.83	—	—	∞	∞
5.25	0.067	0.570	61.3	60.3
5.75	.136	.502	52.7	51.7
6.50	.223	.430	46.0	45.0
7.50	.314	.366	41.3	40.3
∞	—	—	25.8	24.8

*The reboiler is considered the equivalent of one theoretical step. With a partial instead of a total condenser, a second theoretical step also could have been deducted.

(5) Number of Actual Fractionating Plates

To predict the number of actual plates it is necessary to determine the average viscosity of the liquid on the plates. Since the temperature difference between the top and bottom of the tower is so small, the average viscosity may be taken as the viscosity at the average temperature. For this purpose the viscosity of butane at 150°F will be used.

Viscosity of C_4H_{10} @ 150°F = 0.216 cs → 0.216 × 0.523 = 0.113 cp
Fluidity = 1/0.113 = 8.9 cp^{-1}; Plate efficiency = 99%

Using a plate efficiency of 99% the number of actual plates is computed for various reflux ratios:

O/D	S	Theoretical Steps	Actual Plates
4.83	∞	∞	∞
5.25	61.3	60.3	60.9
5.75	52.7	51.7	52.2
6.50	46.0	45.0	45.5
7.50	41.3	40.3	40.7
∞	25.8	24.8	25.0

The number of actual plates is plotted against reflux ratio in Figure 1.

A reflux ratio of 6.50 to 1, or 1.35 times the minimum is selected. The number of actual plates corresponding to this reflux ratio is 45.5 so that a 50-plate tower would be required.

(6) Location of the Feed Plate

The number of plates above the feed is based on the proportion of theoretical steps at total reflux which would be required to effect the change in concentration of the key components between the feed and distillate. This proportion is applied to the actual number of plates (including the reboiler) to determine the number above the feed plate.

In order to take into account any appreciable difference in relative volatility above and below the feed, the relative volatility used for calculating the steps at total reflux between feed and distillate is the geometric mean of α_D and α_A or,

$$\alpha_n = \left(\frac{1.155}{0.97} \times \frac{1.15}{0.97}\right)^{1/2} = 1.19$$

The number of total reflux steps which would be required between the feed and distillate is calculated by the following equation:

$$\left(\frac{18.78}{20}\right)\left(\frac{10}{1.62}\right) = 1.19^n = 5.79;\ n = 10.1$$

Number of actual plates above the feed would then be:

$$\frac{10.1}{25.8}(50 + 1) = 20$$

The vaporization of the feed can be taken into account by adding the fraction vaporized to n since 100% vaporization would be equivalent to a theoretical step at total reflux. This would change the proportion of plates above the feed as follows:

$$\left(\frac{10.1+0.75}{25.8}\right)(50+1) = 21.4 \text{ plates above the feed}$$

Feed lines would probably be installed above the 24th, the 28th and 32nd plates from the bottom of the tower.

Nomenclature

X	moles of any component in distillate or bottoms per 100 moles of feed
x	mole fraction of any component in liquid
y	mole fraction of any component in vapor
D	moles of distillate per 100 moles of feed
O	moles of reflux per 100 moles of fee
O/D	reflux ratio
$(O/D)_M$	minimum reflux ratio corresponding to $S = \infty$
S	number of steps from still to distillate
S_M	minimum number of steps corresponding to $O/D = \infty$
P	number of theoretical plates; with a partial reboiler and partial condenser, $P = S - 2$, and with a partial reboiler and total condenser, $P = S - 1$
Z_L	ratio of mole fraction of any light component to heavy key component in the feed
Z_H	ratio of mole fraction of light key component to any heavy component in feed
α_D	relative volatility of any component to heavy key at the dew point of the distillate
α_W	relative volatility of any component at the bubble point of the bottoms
α_A	relative volatility of any component at the arithmetic average temperature of the dew point of the distillate and the bubble point of the bottoms
α_{av}	mean relative volatility of any component, $(\alpha_D \cdot \alpha_W \cdot \alpha_A)^{1/3}$
LK	used as a subscript to refer to the light key component
HK	used as a subscript to refer to the heavy key component
L	used as a subscript to refer to any light component
H	used as a subscript to refer to any heavy component
D	used as a subscript to refer to the distillate
W	used as a subscript to refer to the bottoms
n	used as a subscript to refer to the plates above the feed
m	used as a subscript to refer to the plates below the feed

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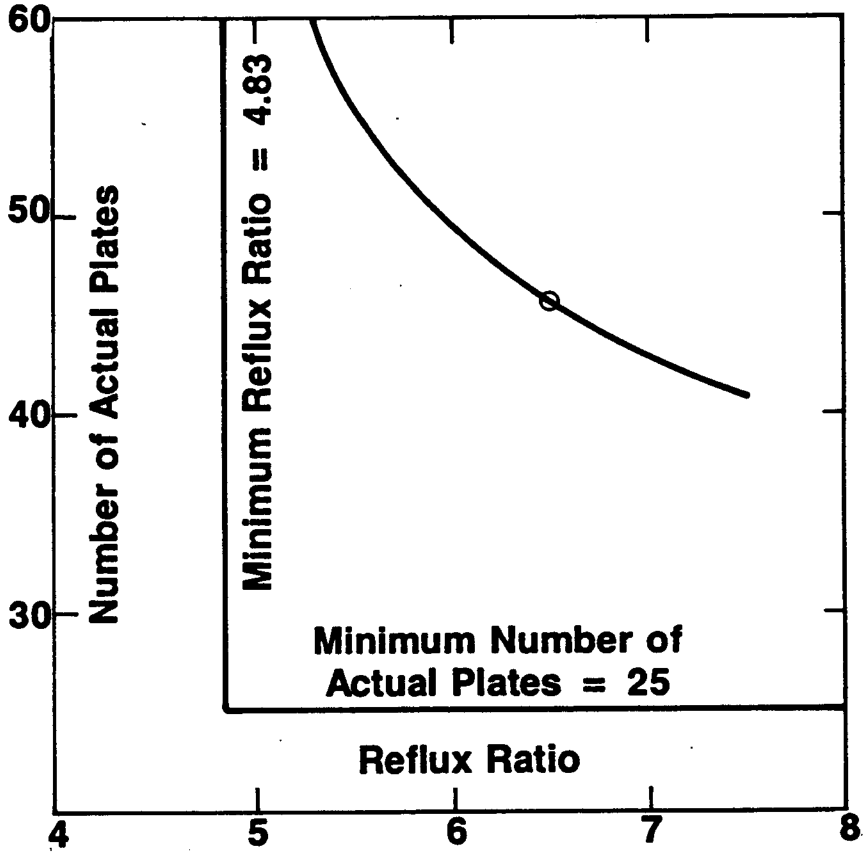
GRAPHIC DESIGN
0890-82-021

FIGURE 1



60
50
40
30
Number of Actual Plates
Minimum Reflux Ratio = 4.83
Minimum Number of
Actual Plates = 25
Reflux Ratio
4
5
6
7
8

FIGURE 2
OVERALL PLATE EFFICIENCY VS FLUIDITY OF LIQUID ON PLATES

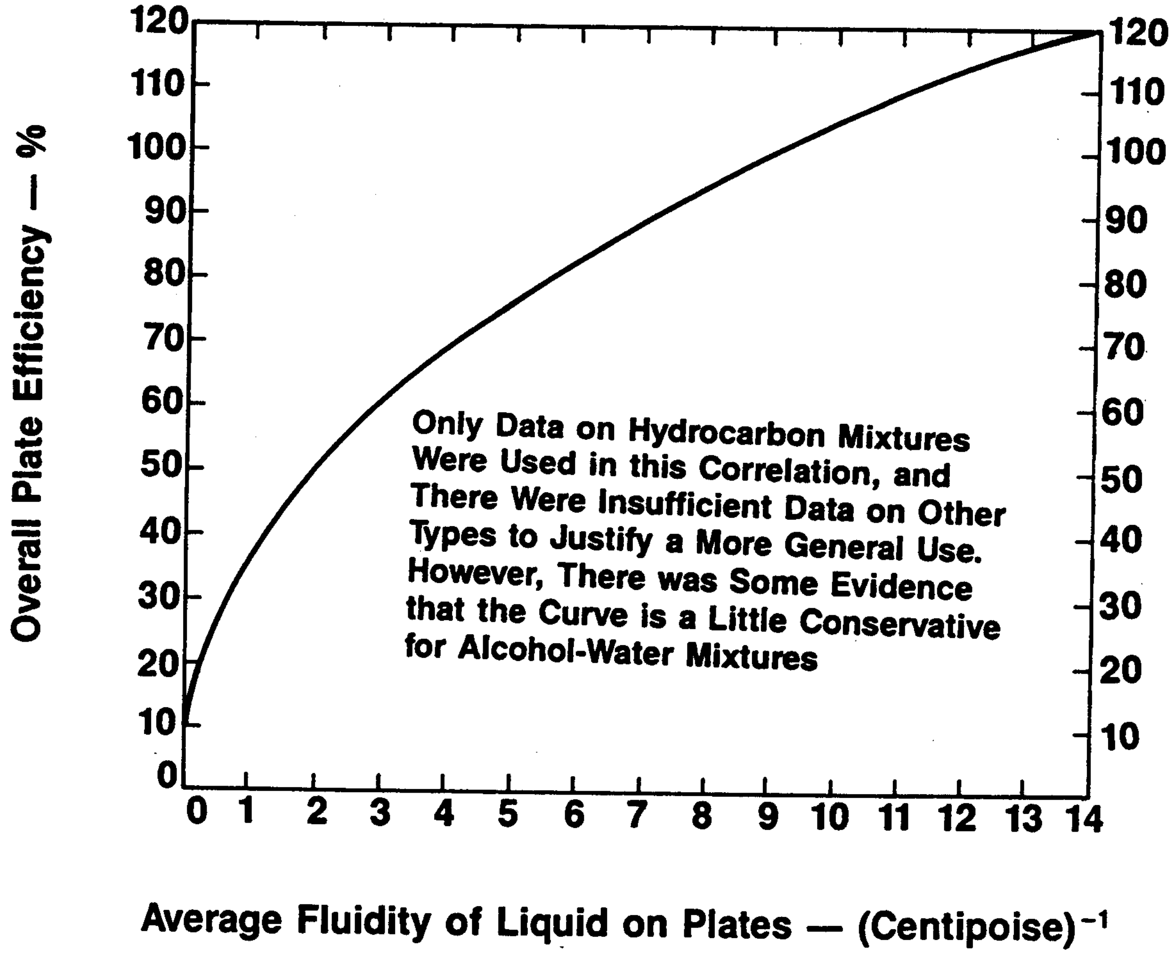
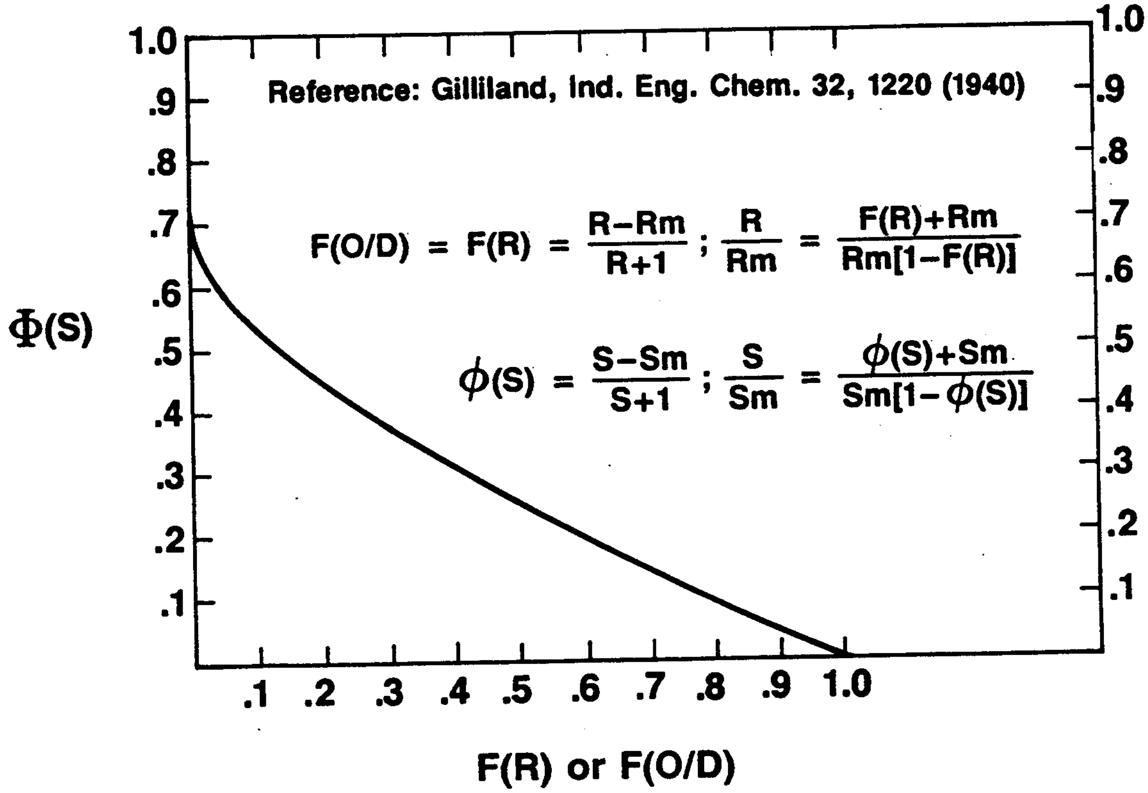


FIGURE 3
GILLILAND CORRELATION
CORRELATION OF THEORETICAL STEPS WITH REFLUX RATIO MULTICOMPONENT AND BINARY MIXTURES



FRACTIONATING TOWERS DEVICE SELECTION AND BASIC CONCEPTS

TABLE 1
TRAYS - A SUMMARY OF CHARACTERISTICS

Tray Type	Capacity	Efficiency	Cost Per Unit Area	Flexibility*	Remarks
Sieve	Medium to high.	High. Equal to or better than other tray types.	Lowest of all trays with downcomers.	Medium. 3/1 can usually be achieved.	First choice for most applications; extensive design data available.
Valve	Medium to high; as good as sieve trays.	High. As good as sieve trays.	Medium. About 10% greater than sieve trays.	High. Possibly up to 5/1.	Not recommended for moderate to severe fouling services.
Nutter V-Grid	Medium to high; as good as sieve trays.	High. As good as sieve trays.	About the same as sieve trays.	Medium. Slightly higher than sieve trays.	Good alternative to sieve trays. Increases run lengths in fouling services.
Jet	Highest at low pressure and high liquid rates.	Low to medium.	Low to medium. About 5% higher than sieve trays.	Low. 1.5 or 2/1.	Consider only when liquid rate exceeds 4.0 gpm/in. of diameter per pass (10.0 $dm^3/s/m$ of diameter per pass).
Bubble Cap	Medium to high, except low to medium at high liquid rate.	Medium to high.	High. At least twice the cost of sieve trays.	Medium to high. 5/1 or slightly higher.	Use for high flexibility where fouling of valve trays may be a problem.
Union Carbide MD	Very high.	Low to medium.	High. Paying for proprietary know-how.	Low. (< 2/1)	Can be installed on very low tray spacings. Consider for revamps where no other device is acceptable. Not recommended for fouling services.

*Ratio of maximum to minimum vapor loads at which tray efficiency remains above about 90% of its design value.

TABLE 2
COUNTER-CURRENT DEVICES - A SUMMARY OF CHARACTERISTICS

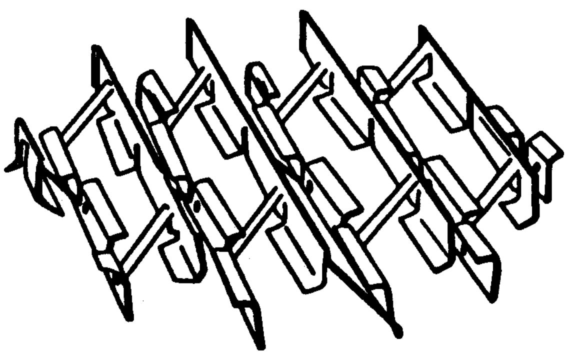
Device	Capacity	Efficiency	Cost Per Unit Area	Flexibility*	Remarks
Packing (Pall Rings, Metal Intallox, Nutter Rings, etc.)	Medium	Medium to high.	Medium to low, depending on material of construction.	> 3/1	Good for low ΔP service. Mainly used in vacuum pipestills and in various high liquid rate absorbers.
Structured Packing Flexipac; Montz Gempak; Mellapak Intalox-Structured	Medium to very high depending on size used.	Medium to very high depending on size used.	High - at least 2 times dumped packing cost.	> 3/1	Best efficiency per unit of ΔP.
Glitsch Grid Flexigrid Snapgrid	Very high.	Poor as fractionation device. Good for entrainment removal and heat transfer.	Medium to high.	Low; less than 2/1.	Good for high vapor-low liquid service to minimize effect of entrainment. Used in wash zones of heavy hydrocarbon fractionators where moderate coking occurs.
Sheds and Disc-and-Donuts	Very high.	Poor as fractionation device.	Medium.	Low. < 1.5/1	Used in severe fouling service; e.g., slurry pumparound in cat fractionator.
Downcomerless (Dual Flow or Ripple)	Highest in some instances.	Medium to good, at design liquid and vapor rates.	Lowest to medium; royalty on Ripple Tray, none on FRI types.	Low.	Of interest for revamps if poor flexibility is tolerable.

*Ratio of maximum to minimum vapor loads at which efficiency remains above approximately 90% of its maximum value. This may be limited by the turndown of the liquid distributor used.

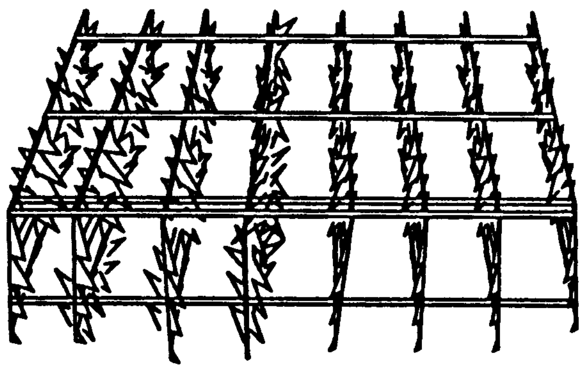
(8)

Grids

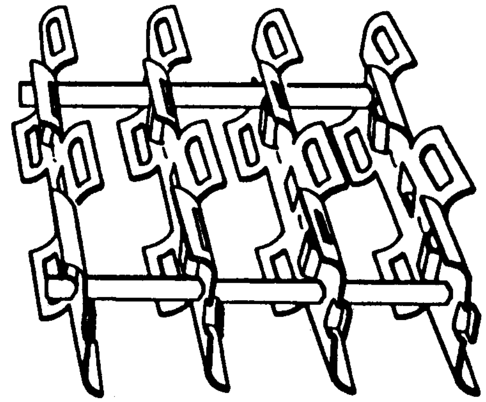
Glitsch Grid



Koch Flexigrid
Style 2

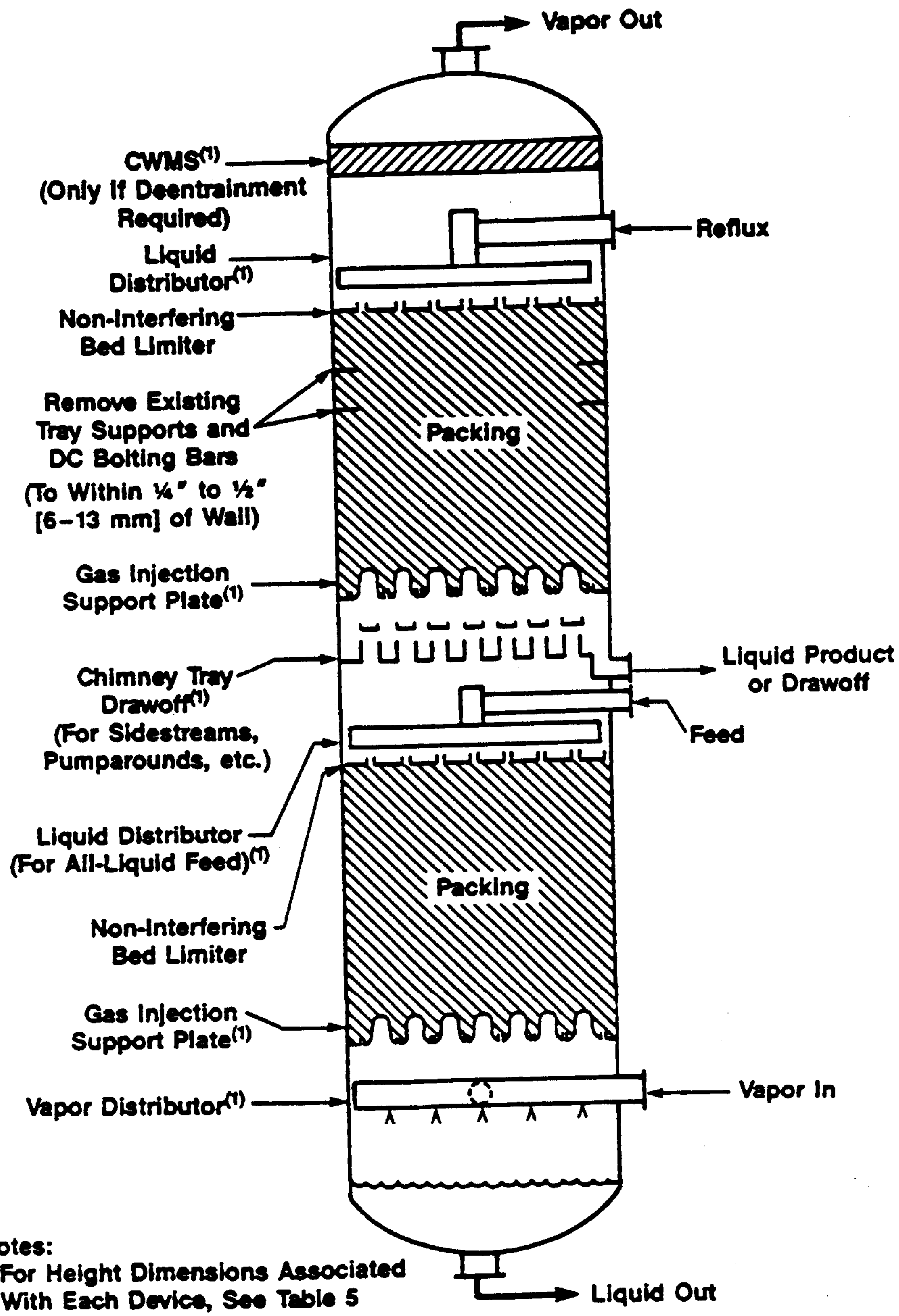


Nutter Snapgrid



- **High Capacity, Low Efficiency**
- **Fouling, Coking, Heat Transfer Services**

Figure 4
ANCILLIARY TOWER INTERNALS
NEEDED IN PACKING INSTALLATIONS



Notes:
(1)For Height Dimensions Associated With Each Device, See Table 5

XDP-IIIG-F4

DISTRIBUTOR CAN HAVE SIGNIFICANT IMPACT ON EFFICIENCY

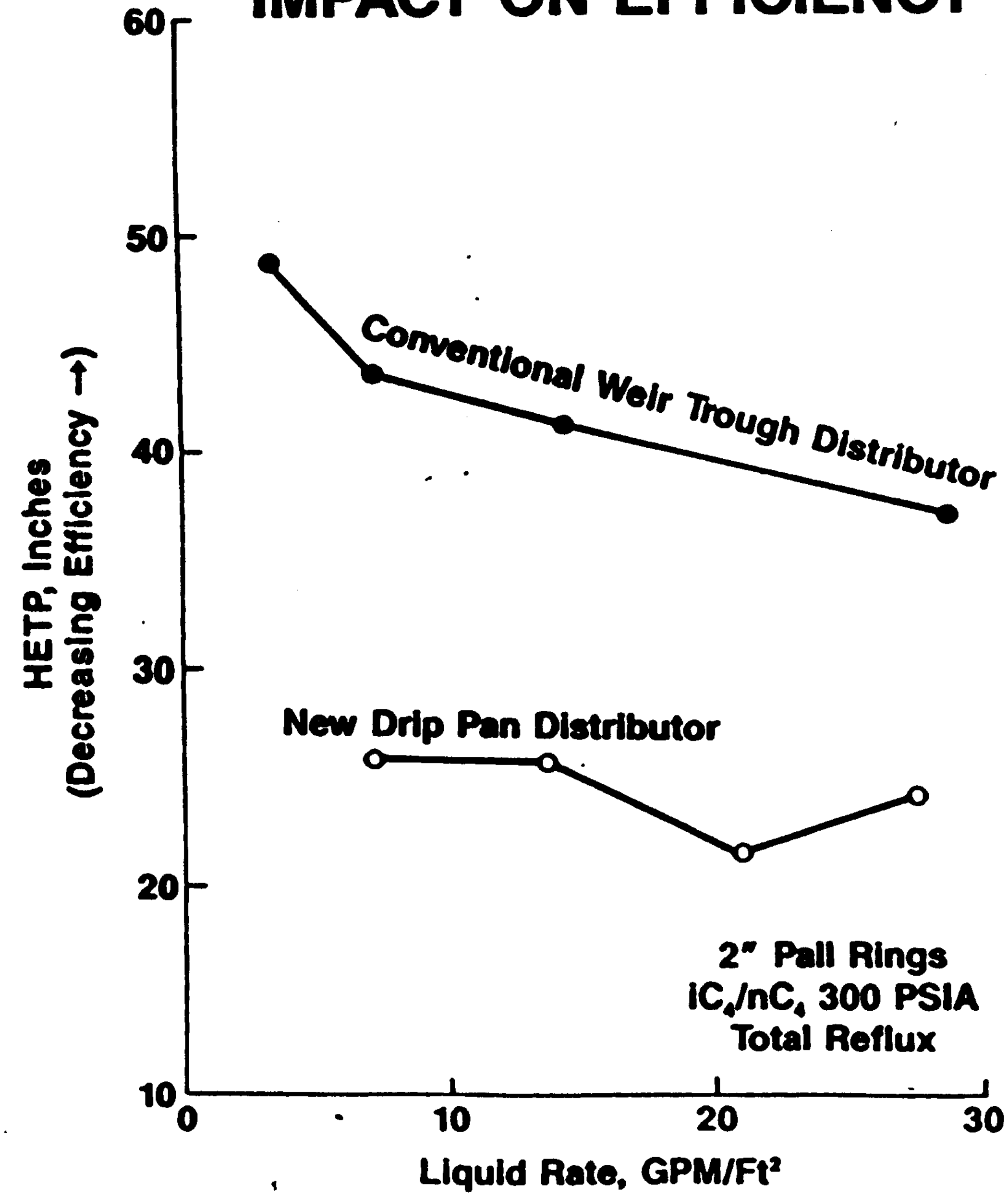
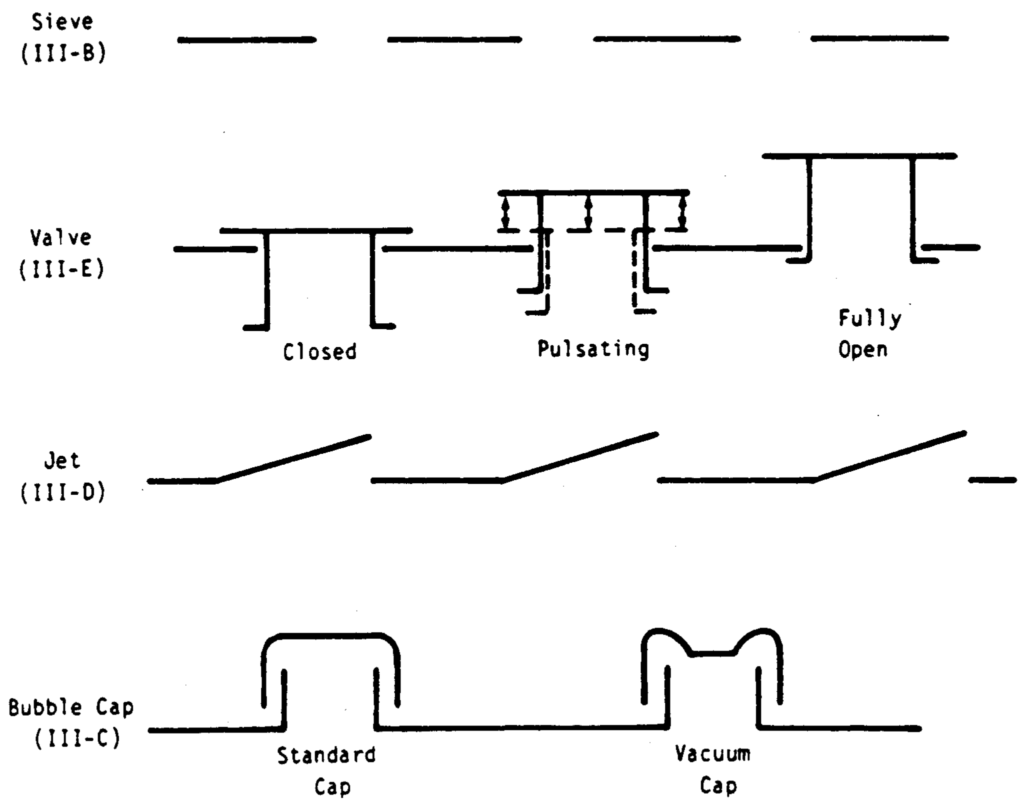


Figure 2
TYPES OF TRAYS
(SCHEMATIC)

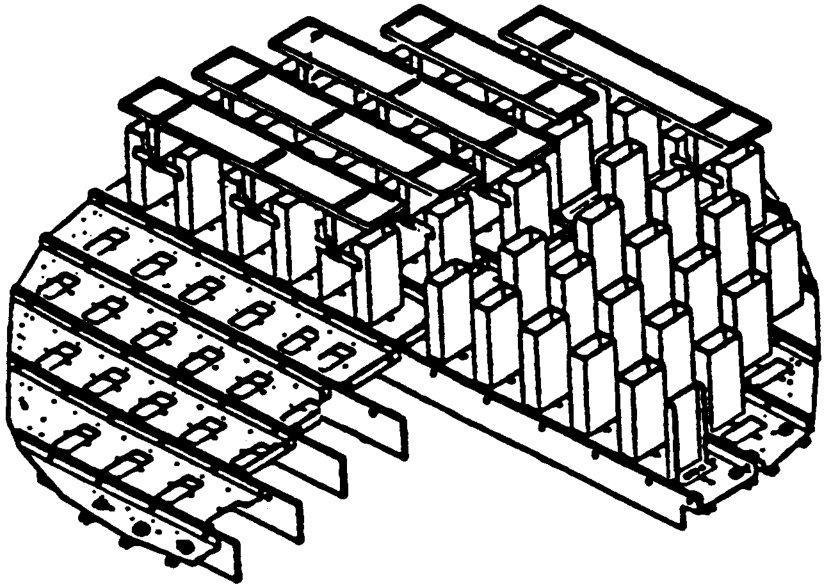


XDP-IIIA-F2

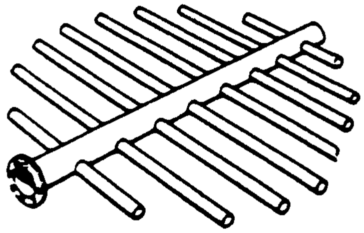
EXXON RESEARCH AND ENGINEERING COMPANY — FLORHAM PARK, N.J.

Figure 5
TYPICAL LIQUID DISTRIBUTORS

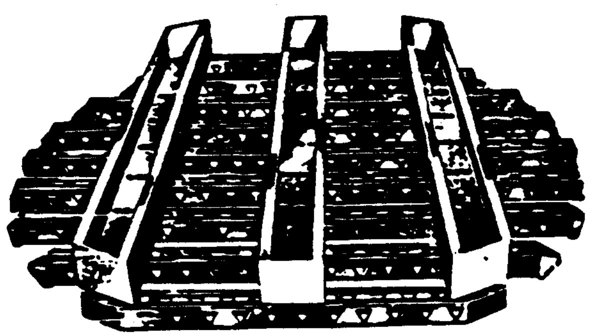
Orifice Pan



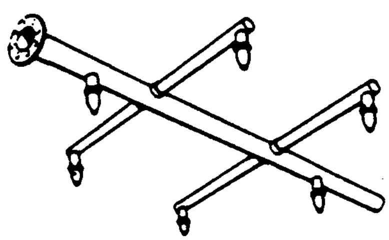
Ladder



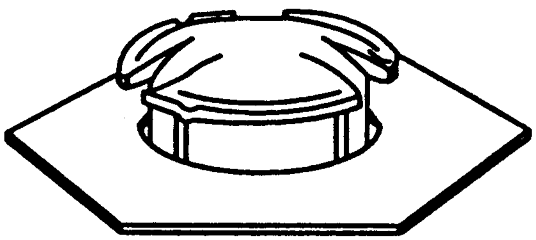
Narrow Channel



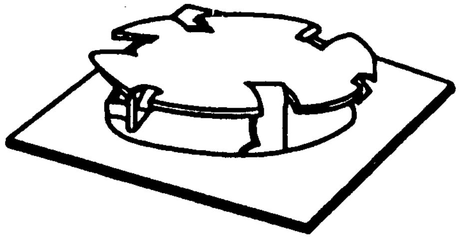
Spray Nozzle



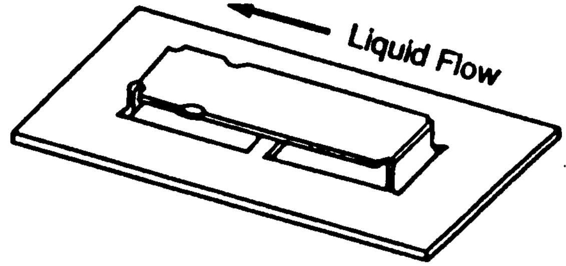
Valve Trays



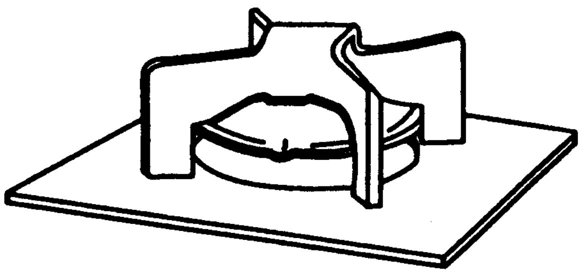
Koch Type A



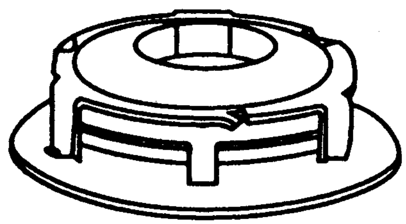
Glitsch V-1



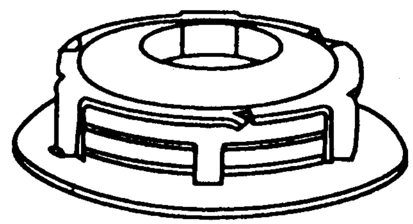
Nutter BDH



Koch Type T



Glitsch A-2



Glitsch Double-Seated A-1

ADDENDUM 2.4
SECTION 2
DISTILLATION

SELECTION AND DESIGN OF COMMERCIAL FRACTIONATION EQUIPMENT

BY P. W. BECKER AND R. K. NEELD
EXXON RESEARCH AND ENGINEERING COMPANY

Although Chemical Engineering technology has grown tremendously in the past ten years, unit operations still remain at the heart of the chemical and petroleum industries. And, within the unit operations area, distillation remains the chief means of the various physical separation techniques. Distillation as an art is old; but distillation as a science is much newer, and a proportionate share of the R&D effort involved in developing new technology has been and will continue to be expended in this area.

The purpose of this handout is to discuss what some of this technology is, especially from the standpoint of the design engineer. First, why is it important. Second, what changes have taken place in recent years. Third, what tools do the designers have to work with and how do they go about selecting and designing commercial distillation equipment.

Importance of Distillation

The large annual investment in distillation equipment is a major reason why work in the fractionation area is important to us, and why we undertake research and development work; spending both time and money trying to perfect our technology in this area. For example, during the past several years, Exxon Corporation alone has spent over ten million dollars annually for towers and trays. Obviously, then, the possibility of even small improvements in the selection, design, and performance of our distillation units can be readily justified.

A second reason for a continuing effort in this area is that distillation is usually the final processing step that determines whether on-spec products are being produced. And, thirdly, the purity specifications for both new and existing products are constantly being raised, thereby requiring more accurate tower design techniques. Furthermore, we could cite a variety of system physical properties, yield-purity relationships, liquid and vapor rates, temperature sensitive materials, and fouling requirements that create special problems affecting hardware design.

Since the oil embargo of 1974, and the subsequent quadrupling of oil prices, another important consideration has arisen--that of conserving energy. Distillation is a large energy consumer in refineries and chemical plants. Heat must be put in at a high temperature level at the bottom, and removed at a lower level at the top. This introduces thermal inefficiency, and necessitates the use of lower reflux and lower pressure drop internals.

COUNTER-CURRENT DEVICES
TYPES AVAILABLE (PACKING, GRIDS, BAFFLE SECTIONS, DUALFLOW TRAYS) (Cont)

Figure 20
PHOTOS OF DUMPED PACKINGS

PALL RING

XDP-IIIA-F20A

NORTON METAL INTALOX (IMTP)

XDP-IIIA-F20B

NUTTER RING

XDP-IIIA-F20C

There are a number of other packing types available but they are not widely used in Exxon towers and therefore have not been included here.

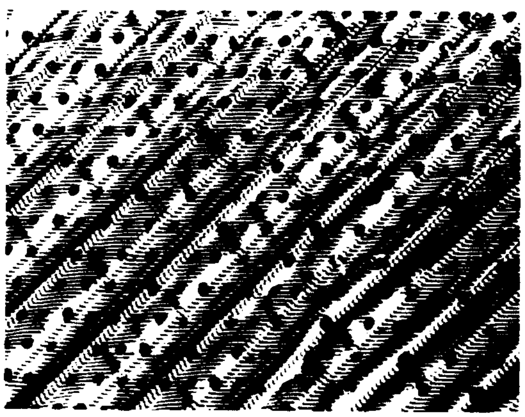
b) **Structured packings** (also called ordered packings). These devices are fabricated in bundles from crimped sheet metal and installed in the tower in layers having a fixed orientation. Since they provide more surface area per unit volume they are more efficient than random packings. However, they cost 2-4 times as much. Since the crimp height can be changed, the capacity, efficiency, pressure drop and cost can also be varied. Thus, the optimum choice must be determined by an economic study.

Of all the contacting devices available, they provide the lowest pressure drop per theoretical stage of contacting as well as the best capacity/efficiency combination. This feature makes them especially attractive in vacuum towers.

There are several suppliers including: Flexipac by Koch Engineering, Gempak by Glitsch, Intalox Structured by Norton, Montz by Nutter Engineering, and Mellapak by Sulzer. One of these devices, by Koch Engineering, is shown in **Figure 21.**

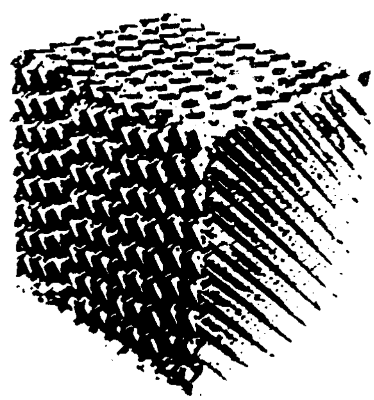
Figure 21
PHOTOS OF STRUCTURED PACKING
(BY KOCH ENGINEERING)

FRONT VIEW



XDP-IIIA-F21A

ISOMETRIC VIEW



XDP-IIIA-F21B

Why New Devices Outperform The Bubble Cap Tray

We must also remember that the variety of equipment (or hardware) available to handle these process-oriented problems is also constantly growing and being steadily improved. For instance, in the early 50's and for many years before, the bubble cap tray was the workhorse of the distillation field. This is no longer true, and we now have:

- devices with greater capacity and lower pressure drop than the bubble cap,
- devices with efficiencies that are equal to or better than the bubble cap,
- devices that are lower cost,
- devices that possess almost equal flexibility to handle varying liquid and vapor loads without loss in effectiveness, and finally,
- devices that are easier to maintain.

Tray Performance Diagram And Design Limitations

A number of physical boundary conditions limit our final tray design. The final tray design should eliminate or, at the very least, minimize the impact of each of these limitations. The correlations available for calculation a "performance diagram" for each contacting device vary from company to company. Contractors have correlations obtained from various sources, tray vendors generally have developed their own, as some companies have also, while other companies may use Fractionation Research Inc. or the literature sources. Regardless of the source, these design limitations are calculable, and the various correlations themselves will not be discussed.

The physical boundary conditions include:

Weeping

The weep point is that vapor rate at which liquid starts to leak through the tray openings. It is not necessarily the lower operation limit for good tray efficiency. For systems with high liquid-to-vapor ratios, a small amount of liquid bypassing will not seriously reduce tray efficiency, providing it is only a small fraction of the total liquid on the tray.

Dumping

Dumping is excessive leakage of liquid through the tray openings. It is characterized by a sudden and significant drop in tray efficiency. The minimum vapor rate for acceptable tray performance is equal to or greater than that at which dumping occurs.

Blowing

Blowing is a fine dispersion of fog of tray liquid which becomes entrained to the tray above. It is caused by excessive vapor velocity through the tray openings at relatively low liquid loadings. The high hole velocity is caused by a high vapor loading or low percent hole area, or both. Blowing results in poor contacting since it lifts the liquid or froth phase off the tray and the vapor phase thus becomes the continuous phase. This limitation does not necessarily result in flooding.

Flooding

Flooding is an unstable condition in which the liquid height on the tray and in the downcomer builds up until the tower is essentially full of dense foam. The two principle causes of flooding are, first, excessive entrainment and, second, excessive downcomer filling.

Entrainment Flooding or Jet Flooding

Entrainment or "jet" flood is a condition where a vapor handling limitation is caused by the carryover or "jetting" of liquid droplets from one tray to the tray above because of excessive vapor velocity through the tower free area. This represents the maximum capacity of the tray at a given tray spacing.

Excessive Downcomer Filling

High tray pressure drop or insufficient disengaging of vapor in the downcomer results in a buildup of froth in the downcomer and eventual tray flooding. It can occur at any liquid at any liquid rate if inadequate downcomer clearance, inadequate downcomer area, or inadequate tray spacing is provided.

Liquid Gradient

Liquid gradient may also cause problems. The change is depth of liquid on a tray, from the inlet toward the outlet, is the liquid gradient. Depending on the type of tray (that is, the resistance to flow) ad the motion of the vapor, the static head represented by the liquid gradient may furnish anywhere from a negligible to a major part of the driving force that moves the liquid across the tray. Gradient problems are worst on bubble cap trays and least on jet trays.

Downcomer Velocity and Disengaging

The liquid velocity into the downcomer must be low enough to allow vapor to disengage and travel up and out of the downcomer against the flow of incoming froth. If this velocity is exceeded, the increased downcomer level due to excessive aeration may cause flooding of the tray. In addition, the vapor in the froth mixture may be of such magnitude that when it disengages on the tray and result in premature jet flooding.

Ultimate Capacity

Ultimate capacity is another limitation that is occasionally encountered. The ultimate capacity is the upper limit of vapor loading which the tower can handle. It depends mainly on the physical properties of the system. This vapor load cannot be increased by changing the tray design or increasing the tray flooding occurs first; but, in certain cases, an ultimate capacity limitation may be reached before jet flooding. Thus, each tray design should always be checked for ultimate capacity.

Flexibility or Turndown Ratio

Flexibility is the ratio of maximum to minimum vapor loads which bound the range of operation conditions over which the tray will perform satisfactorily; i.e., tray efficiency remains above, roughly, 90 percent of its maximum value.

The flexibility of a typical tray is depicted in Figure 1. We see that tray efficiency remains relatively constant in the range of loadings from 40 to 90 percent of the flood point. Below 40 percent of flood, efficiency decreases due to weeping, dumping, and poor contacting due to low interfacial area. Above 90 percent of flood, efficiency rapidly drops off due to high entrainment. If better turndown that this is required, a special device might be required. Therefore, it is important that you, as a designer, carefully determine the minimum loading conditions that the tower will actually see. Quite often, arbitrary or unrealistic turndowns are specified, and the cost of the tower increases appreciably; for example, shifting to a costlier device.

Now that we briefly reviewed the various tray limitations, let's look at the various types of trays available to us. The contacting devices can be broken into three broad categories. These include: downcomer Type Trays, Downcomerless Type Trays, and Packing. Category 1 - The Downcomer Type Trays which include bubble cap, sieve, and valve trays. Some of the major characteristics of each device include:

Category 1 - The Downcomer Type Trays

- Bubble caps - There are many, many types of bubble cap tray designs available to the industry. And, surprisingly enough, there is probably very little difference in the performance of those designs that might be called the best. The better designs are characterized by being relatively high in efficiency, in capacity, and in flexibility. In general, however, they do possess a higher pressure drop than the other devices that we'll talk about. But, their biggest deficiency is that of cost. Other types of trays are as much as 50 percent cheaper than bubble cap trays; and this is, perhaps, the main reason why bubble caps have lost their popularity.

- Sieve trays - We find that they are currently the most widely-used of all the trays available. A typical, modern sieve tray is shown on the next Slide 6. This tray exhibits good capacity, excellent efficiency, and good flexibility--that is, it will operate quite efficiently at loadings which are one-half to one-third of design values. And, perhaps most importantly, it is the cheapest device currently available. One reason for the increasing popularity of sieve trays can be traced to the work of an organization called Fractionation Research Inc. (FRI). This organization has been in operation for over 20 years as a non-profit, industry-sponsored research cooperative. It has published performance data on almost all of the important contacting devices currently available. These data have allow not only pretty good comparative evaluations to be made, but have also allowed design procedures to be established.

- Valve trays - These trays contain proprietary devices manufactured by three concerns: Koch Engineering, F. W. Glitsch & Sons, and Nutter Engineering. These trays consist of a number of discs (or valves) suspended above the holes of a perforated tray at a height which depends upon the vapor flow rate to the hole. The trays themselves are characterized by a high turndown ratio. Capacity and efficiency are both high, and are similar to sieve trays, while their cost averages about ten percent more than sieve trays. They are generally used where wider flexibility than that obtained from sieve trays is required. By the way, this is a "two-pass" valve tray. That is, the liquid flows from left to center, or from right to center. On the trays directly above or below this one, liquid flows from the center to either side.

- Jet trays are similar in construction to a standard sieve tray except the punched holes are replaced by inclined tabs. The tabs are punched from the tray deck and inclined to the plate at a typical angle of 20° from the horizontal. This tray is characterized by having a high liquid and vapor handling capacity. Liquid handling capacity is high because the vapor assists or "pumps" the liquid across the tray. The tray is used in heavily-liquid loaded services where multi-pass convential trays are normally required. For most distillation towers (where liquid rate is normally low), the tray is not generally used since its capacity advantage becomes marginal, and its efficiency is less than a sieve or valve tray.

- V-Grid tray - A typical V-Grid tray is a convential crossflow tray. Distributed throughout the tray deck are V-Grid units, which can best be described as rectangular valves which are always in the fully-open position. The main difference between the operation of the V-Grid tray and a typical sieve tray is in its vapor flow patterns. While vapor always flows in a vertical direction through the sieve tray, it must flow horizontally through the V-Grid units. Because V-Grid units are laid out on a rectangular (rather than triangular) pitch, vapor jets leaving adjacent units oppose each other, thereby promoting increased turbulence and mass transfer.

The operation of the V-Grid was compared to that of a sieve tray during a laboratory test program conducted by Nutter at their facilities in Tulsa, Oklahoma(1). These data indicated that this increased action on the V-Grid tray resulted in its having somewhat higher fractionation efficiency than a sieve tray. Also, liquid did not weep as readily through the V-Grid tray, which indicated that V-Grid has a better turndown ratio than the sieve tray although certainly not as good as a capacity advantage over a sieve tray under most conditions, it did exhibit higher entrainment than the sieve tray at very low liquid rates. Thus, at moderate and high liquid rates, the V-Grid tray's performance is at least equivalent to, and possibly somewhat better than, that of a sieve tray. At low liquid rates, however, V-Grid trays should be used cautiously since they may entrain excessively.

The V-Grid tray should also be particularly attractive in fouling services. The violent tray action (caused by the opposing vapor jets) should make the V-Grid tray a self-cleaning device. Nutter claims that their trays have been used successfully in services which were previously plagued with tray fouling problems.

Category 2 - The Downcomerless Trays

- Dual flow is a generic term and describes, basically, a downcomerless sieve tray in which both liquid and vapor flow takes place through the same holes. This could well be the sieve tray you just looked at with the downcomers completely blocked off and the area filled with more of the perforated sheet that comprises the active area of the sieve tray.

- Ripple tray - This is a proprietary device that's been patented by the Stone and Webster Engineering Company. They fabricated this device by crimping a standard flat dualflow tray into sine waves of a desired frequency and amplitude.

  We have some data on the performance of both the dualflow and ripple tray and have found they possess very high capacity and moderate efficiency. However, the flexibility or turndown ratio over which good efficiency can be obtained is generally poor. As a matter of fact, unless the tray is properly designed for the load at which it is to operate, it won't do the job. It is normally considered only when higher capacity is desired and some sacrifice in efficiency and turndown ratio can be tolerated. This is the case in some debottlenecking studies.

(1) The results of this program were discussed in detail in A.I.Ch.E. Preprint 49c (Ammonia Stripping Efficiency Studies), Sixty-Eighth National Meeting, March-April, 1971 by D. E. Nutter.

Category 3 - Packing

Four of the more familiar packings that are widely used today are the Intalox Saddle, the Pall ring, the Raschig Ring, and the Berl Saddle. Both the Raschig Ring and the Berl Saddle, while still used in some limited services, have been largely replaced by the Pall ring and the Intalox Saddle. This is due to the higher capacity and efficiency of these newer packings. The Raschig Ring is nothing more than a short cylinder. When made from ceramic material, its walls are usually quite thick. The Pall ring, on the other hand, has thin, slotted sides with fingers that protrude inward. This permits much more of the ring's surface to be used in the mass transfer process by producing more turbulent contacting and reducing stagnant zones. Likewise, the new Intalox Saddle is more efficient than the older Berl Saddle it has largely replaced. On an overall basis, i.e., efficiency, capacity, turndown, and cost, we have found the two-inch metal Pall ring to be the most economic packing to use in most of our distillation services.

Packed towers are primarily used in the following applications:

- vacuum systems where low pressure drop is critical,
- debottlenecking existing trayed towers,
- light ends towers at high liquid loadings,
- absorption and stripping services (high liquid loadings).

In addition, packing is also used frequently in small diameter towers (under three feet) and in corrosive services where a ceramic material of construction is required.

Are These All of the Contacting Devices Available

The trays and packings discussed previously will usually fill more than 95 percent of the designer's needs. However, there are certain applications where some of the more non-conventional or high cost devices are justified. Although there are many, only a few more commonly-used ones are described here. These are:

- The MD Tray - This tray is a perforated fractionating device which features a multiplicity of narrow downcomers distributed over the entire tray surface. Hence, the name "multiple downcomer". The Linde[2] tray is a high capacity device which makes it especially attractive for debottlenecking a tower. While its efficiency is somewhat lower than a

(2) Delnicki, W.V., and Wagner, J. L., "Performance Characteristics of Linde Multiple Downcomer Distillation Trays", Preprint 24c A.I.Ch.E. Sixty-Second Annual Meeting, November 16-20, 1969.

standard cross-flow sieve tray, its high capacity may mean that extra trays can be added to a given tower or the reflux rate increased somewhat to offset any fractionation debit. For a specific case, you should contact Linde to determine whether the MD tray is attractive for your application.

- Sulzer Packing - This packing is manufactured by Koch Industries, Inc. The packing is a woven wire fabric material which is completely self-wetting. The packing is a high efficiency device (i.e., low HETP) with a very low pressure drop. It has special application in distilling heat-sensitive materials under vacuum conditions. Its main disadvantage for more general use is its cost, being about $\$450/ft^3$ (as compared to about $\$20/ft^3$ for stainless steel two-inch pall rings).

- Perform-Kontakt Tray - This device is made from pieces of expanded metal and contains a number of intermediate weirs. In addition, the metal is expanded in such a manner that the vapor helps "push" the liquid across the tray. Because of this feature, the tray is claimed to have a high capacity and is, therefore, recommended for debottlenecking as well as new designs. Unfortunately, due to tis newness in the U.S., relatively little data is publically available for comparison purposes. The vendor should, however, be contacted for more specific details and recommendations on the application of this tray(3).

What is Involved in Tower Design?

Having discussed some of the characteristics of the equipment that designers, might use, let's briefly go through the steps involved in designing a distillation tower. With the advent of computers, this has become more rigorous in recent years.

The steps involved are summarized as follows:

1. Define the key separations - Product specifications or degree of separation can be given to us in a number of ways; perhaps as a flash point or an amount of material boiling below or above a certain temperature. On simpler towers, where individual components can be identified, the amount of purity is given. We must, by some means, tie this into a yield and quality description. Although we don't need, at the start, to have a complete definition of the products (that is, we don't need a complete material balance on all components), we do need to know the key requirements of the separation, that is, the yield and the purity level of our products.

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(3) For further information, contact Ferrostaal Overseas Corporation, 17 Battery Place, New York, New York. (212) 422-6535, Attention Mr. Alfred B. Walter, Manager.

2. Obtain the appropriate vapor-liquid equilibrium and enthalpy data - A number of basic data correlations are available to use. At last count, Exxon had over a dozen different correlations or techniques in use for different types of problems. One of our big jobs as designers is to define the limits and proper application of each of these correlations.

3. Calculate the theoretical plates required at different reflux ratios - The calculation techniques for determining the right combination of theoretical trays and reflux ratio has improved tremendously in recent years, and these techniques are limited now only by the accuracy of the basic data. Today, we have plate-to-plate programs that can handle up to 150 components, five sidestreams, and five heat removal sections or pumparounds. A complete printout gives us tray and produce vapor and liquid rates, temperatures, compositions, and certain physical property information. If we run several cases with different numbers of trays and different reflux ratios, we can then define the shape of the theoretical trays vs. reflux ratio curve.

Before we can select the optimum combination, we need to:

4. Estimate the tray efficiency - Lacking specific data on a similar system, there are a number of correlations available to us to do this, the most fundamental being the A.I.Ch.E. method for bubble cap trays. Since this correlation was developed from bubble cap data on binary systems, most of them aqueous solutions, there is some question as to its applicability to the multicomponent hydrocarbon systems frequently encountered in the petroleum industry. Exxon has, therefore, derived our own correlations for predicting sieve tray efficiency.

   With our estimated plate efficiency, we can now go back and calculate the actual number of trays that corresponds to each reflux ratio. We now have the basis for an economic balance to determine which is the proper combination to use. In general, we have found that reflux ratios in the range of 1.2 to 1.5 times the minimum are usually optimum.

5. Define the minimum and maximum feed rates that our tower must handle - It is important that the designer be given the range of loadings over which his tower must operate, and this is a number that is frequently not easily obtained. In general, an assumption of a 2 to 1 rangeability fits most normal operations. When greater flexibility than this is required, the degree of flexibility becomes one of the key items in determining both the type of tray to use and how it should be designed.

The next step in designing towers should be:

6. Selecting the best tray for the given set of process conditions. Choosing the best tray type goes back to our earlier discussion on the merits of the various trays. For about 90 percent of new designs, sieve trays have proved adequate to cover both the range of loadings and the various services required. Of course, such items as coking tendency, allowable pressure drop, flexibility requirements, and, in the case of bottleneck removal, required capacity and/or efficiency will have some bearing on tray selection. By and large, however, sieve trays have

become the design engineer's first choice among the available contacting devices. If packing is used, it is hard to beat the two-inch Pall ring.

7. Tower sizing and tray hydraulic calculations complete our tower design. Most companies have their own design methods for this phase of tower design; and, in almost all cases, towers are designed to flood by entrainment; that is, liquid is carried from one tray to the tray above by the ascending vapor. But, it is also essential to check the overall tray pressure drop, downcomer liquid velocity, and downcomer filling to assure that the trays will be hydraulically stable. While the hydraulic calculations are pretty well standardized, tray capacity correlations are, by and large, of the generalized empirical variety. Principal variables governing the allowable vapor velocity are: liquid and vapor density, liquid surface tension, viscosity and rate, and certain tray hardware factors; namely, tray spacing and percent hole area.

8. Process control - In general, tower control schemes were pretty well standardized until a few years ago when so-called super fractionators, and by this we mean towers with large numbers of trays and high reflux ratios, appeared on the scene. This has given rise to a number of studies of control schemes involving lag time estimates, temperature-composition sensitivity, (which, incidentally, is usually poor), and chromatographic analyzer applications. In addition, small analog computers seem to offer promise in this area to optimize utilities and to standardize product output of these towers since large lag times make manual control difficult. Tower control issues may thus influence the final tower design.

SECTIONS 3 & 4
HEAT TRANSFER AND HEAT EXCHANGERS

What We'll Cover:

SECTION 3

- Heat Transfer Theory-Review
- Relation of Heat Transfer Theory to Shell and Tube Heat Exchangers
- Design of a S&T Exchanger—Procedure Outline

SECTION 4

- Design Features and Parameters of Shell and Tube Exchangers

3.00

BASIC HEAT TRANSFER CONCEPTS

- Flow of Heat Behaves Like Flow of Fluids and Flow of Electrons

$$\text{Rate} \propto K \cdot \frac{\text{Driving Force}}{\text{Resistance}} \text{ (General)}$$

$$Q \propto K \cdot \frac{\text{Pressure Drop}}{\text{Resistance}} \text{ (Fluids)}$$

$$I = 1.0 \cdot \frac{\text{Voltage}}{\text{Resistance}} \text{ (Electricity)}$$

$$Q \propto K \cdot \frac{\text{Temperature Difference}}{\text{Resistance}} \text{ (Heat)}$$

3.01

COMPARISON WITH FLUIDS

Fluids: $$\left(\frac{Q}{A}\right)^2 = K \cdot \frac{P_2 - P_1}{\left(\frac{fL}{D}\right)}$$

(Section 6 Will Treat Fluid Flow)

Heat: $$\frac{Q}{A} = 1 \cdot \frac{T_2 - T_1}{R_T}$$

FLUIDS	HEAT
Q = Volume/Second	Q = Btu/Hour
P_2, P_1 = Higher, Lower Pressures	T_2, T_1 = Higher, Lower Temperatures
A = Area Available for Flow	R_T = Total Specific Resistance
$\left(\frac{fL}{D}\right)$ = Number of Fluid Flow Resistance Units	A = Area Available for Flow of Heat

BASIC HEAT TRANSFER EQUATION

$$\frac{Q}{A} = 1 \cdot \frac{T_2 - T_1}{R_T} = 1 \cdot \frac{\Delta T}{R_T}$$

R_T = Total Resistance, $Hr \cdot FT^2 \cdot {}^\circ F/Btu$

tancia

$\frac{1}{R_T}$ = Total Conductivity = U_o $Btu/Hr \cdot Ft^2 \cdot {}^\circ F$

$$\frac{Q}{A} = 1 \cdot U_o \, \Delta T$$

$$Q = U_o \cdot A \cdot \Delta T \text{ Btu/Hr}$$

U_o is Referred to as the "Overall Heat Transfer Coefficient"

TOTAL RESISTANCE TO HEAT FLOW—HEAT EXCHANGERS

- There Are Two Areas Through Which Heat Must Flow: the Inside Tube Area and the Outside Tube Area. Resistance Occurs at Both Areas.
- The Industry Standard Reference Area is the *Outside* Tube Area.

3.04

INDIVIDUAL COMPONENTS OF THE TOTAL RESISTANCE

Inside Film Resistance = $R_{io} = R_i\left(\frac{A_o}{A_i}\right)$

Inside Fouling Resistance = $r_{io} = r_i\left(\frac{A_o}{A_i}\right)$

Tube Wall Resistance = $r_w = \frac{\ell_w}{k_w}$

Outside Fouling Resistance = r_o

Outside Film Resistance = R_o

$$R_{io} + r_{io} + r_w + r_o + R_o = R_T = \frac{1}{U_o}$$

ℓ_w = Wall Thickness, Feet

K_w = Thermal Conductivity, Btu/Hr·Ft^2·$\frac{°F}{Ft}$

r = Resistances, Hr·Ft^2·°F/Btu

3.05

TYPICAL RESISTANCE VALUES

	Very Low	Typical	Very High
Film Resistances (Each)	0.00050	0.004	0.04
(Inverse = h)	(2000)	(250)	(25)
Fouling Resistance (Each)	0.001	0.002	0.01
Inverse	(1000)	(500)	(100)
Wall Resistance	0.000030	0.00027	0.00049
Inverse	(32,000)	(3760)	(2030)
Total Resistance	0.00303	0.01227	0.10050
Inverse	(330)	(81)	(10)

THE CONTROLLING COEFFICIENT

- Frequently one of the Two Film Coefficients determines the Value of the Overall Coefficient:

Outside Coefficient, h_o =	75	75	150
Inside Coefficient, h_{io} =	1000	3000	1000
R_o =	0.01333	0.01333	0.00667
R_{io} =	0.00100	0.00033	0.00100
$r_w + r_{io} + r_o$ =	0.00070	0.00070	0.00070
R_T =	0.01503	0.01436	0.00837
U_o =	66.5	69.6	119.5
Improvement =	Base	+4.6%	+80%

- Hence h_o is the "Controlling Coefficient", and Efforts to Improve Exchanger Performance Should Concentrate on This Side of the Exchanger.

A. TEMPERATURE DROPS ACROSS THE RESISTANCES

- **Temperature Drop Across Each of the Resistances is Directly Proportional to Each Resistance.**
- **For Example, If $T_2 = 200$ and $T_1 = 80$, Then Total Temperature Drop = 120°F, and:**

		Temperature Drop	
R_o	= 0.01333	77.6	$= \frac{0.01333}{0.02063} \cdot 120$
R_{io}	= 0.00500	29.1	
r_w	= 0.00030	1.7	
$r_{io} + r_o$	= 0.00200	11.6	
R_T	= 0.02063	120°F	

B. TEMPERATURE DROPS ACROSS THE RESISTANCES

A Useful Concept is Heat Flux $= \left(\frac{Q}{A}\right) = \frac{\text{Btu}}{\text{Hr}\cdot\text{Ft}^2}$

$Q = U \cdot A \cdot (T_2 - T_1) = U \cdot A \cdot \Delta T$

Then $\underline{\underline{\Delta T}} = \frac{Q}{U \cdot A} = \left(\frac{Q}{A}\right) R = \underline{\underline{\text{Flux} \cdot \text{Resistance}}}$

In Previous Slide, If A = 4000 Ft^2 Then $\frac{Q}{A} = \frac{\Delta T}{R_T} = \frac{120}{0.02063}$

$= 5817 \frac{\text{Btu}}{\text{Hr}\cdot\text{Ft}^2}$, and ΔT Across $R_o = 5817 \cdot 0.01333 = 77.6$°F

as Shown on That Slide.

BACK TO BASICS

- We've Looked at Basic Theory, and Discussed $Q = U_o * A * \Delta T$. In Refinery Work We Usually Know Either Q or A, and Need to Calculate the Other Value.

  How Do We Do It?

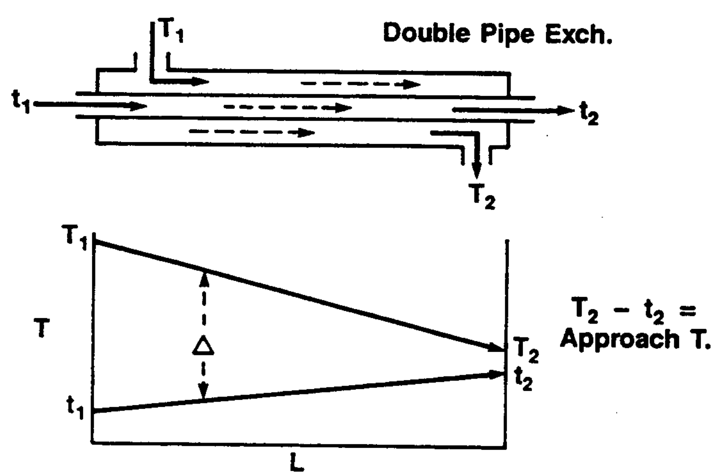
- Either Question Requires Calculating U_o and ΔT.
- We'll Talk About U_o Later, First Let's Discuss ΔT, The Temperature Driving Force.
- Note that Capital Letter T Denotes the Hot Stream, While Lower Case t Denotes the Cold Stream:

  T_1 = Hot In $\quad$ T_2 = Hot Out
  t_1 = Cold In $\quad$ t_2 = Cold Out

3.10

FLOW PATTERNS AND TEMPERATURE DRIVING FORCE

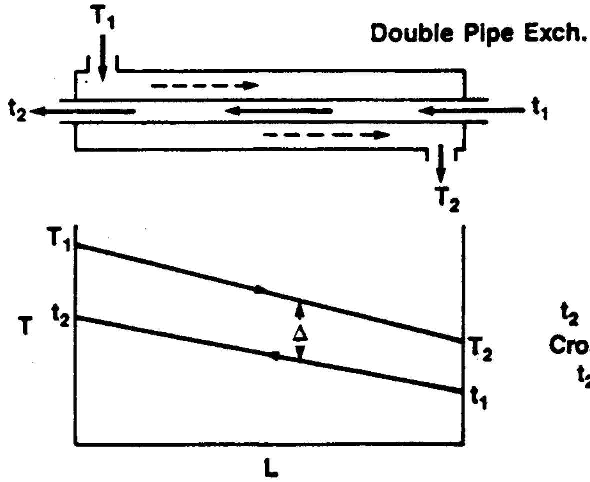
A. Cocurrent (Parallel) Flow



3.11

FLOW PATTERNS AND TEMPERATURE DRIVING FORCE

B. Countercurrent Flow

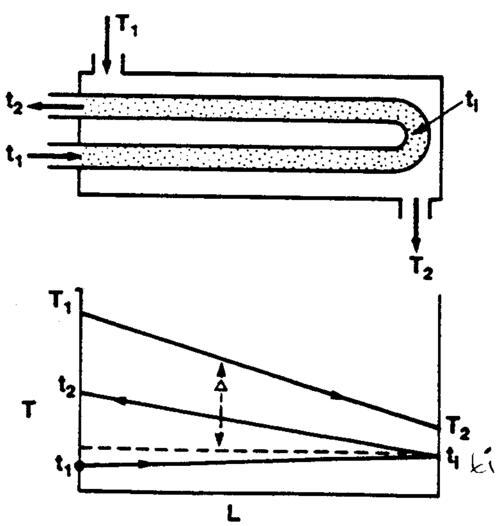


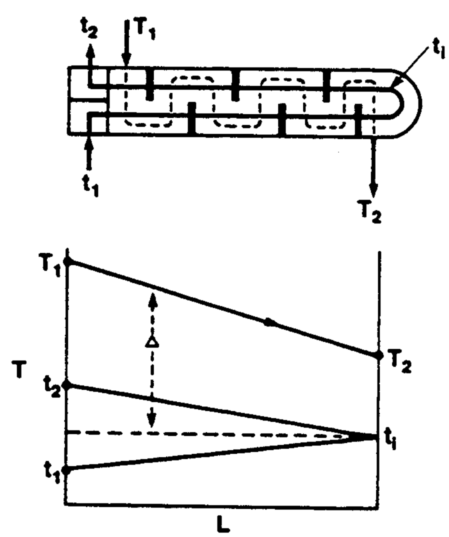
$t_2 - T_2$ =
Cross When
$t_2 > T_2$

3.12

FLOW PATTERNS AND TEMPERATURE DRIVING FORCE

C. Mixed Flow

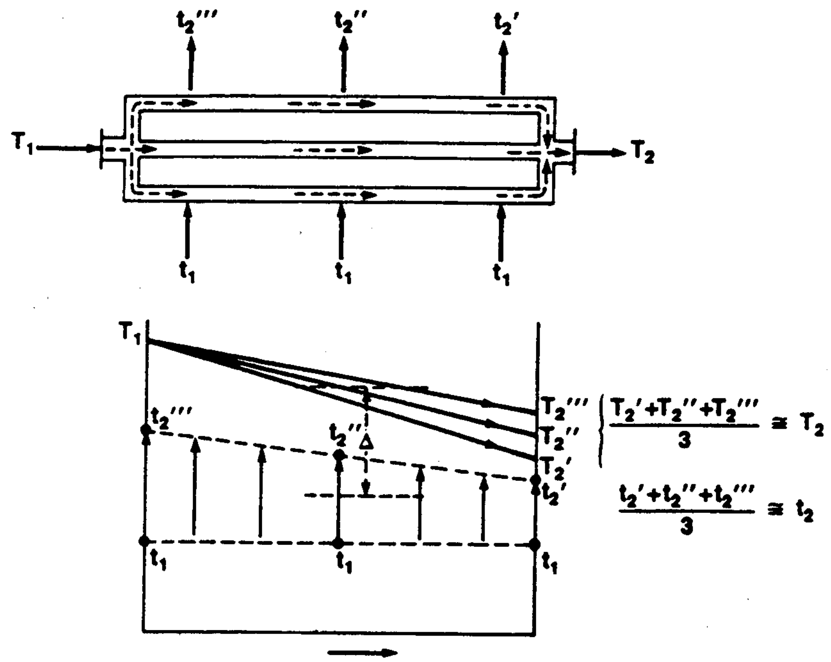




3.13

FLOW PATTERNS AND TEMPERATURE DRIVING FORCE

D. Transverse Flow



3.14

TEMPERATURE DRIVING FORCE

- From the Preceeding Slides, It is Clear That Some Sort of Average Driving Force Must Be Used in Design Calculations. What is This Average?
- The Average is Called "The Effective Mean Temperature Difference", or MTD_e.
- For True Countercurrent and True Cocurrent Flow, the Effective Driving Force Equals the Log Mean Average of the Two Extreme (Largest and Smallest) Deltas.

$$\Delta T_e = LMTD = \frac{\Delta T_L - \Delta T_S}{LN\left(\frac{\Delta T_L}{\Delta T_S}\right)}$$

This is Precisely True Only When the Heat Release Curves are Straight Lines. Otherwise it is an Approximation.

3.15

TEMPERATURE DRIVING FORCE

- What About Mixed Flow: Shell and Tube Exchangers?
- The Complex Flow in These Units was Analyzed Mathematically Many Years Ago, Resulting in Rigorous Equations for a Correction Factor, F_n. This is Multiplied by the LMTD to Give the Correct MTD_e.

  $MTD_e = F_n * LMTD$
- Equations are Valid Only When Heat Release Curves are Linear.
- Similar Relations are Available for Transverse Flow (Air Fin Coolers, For Example).

3.16

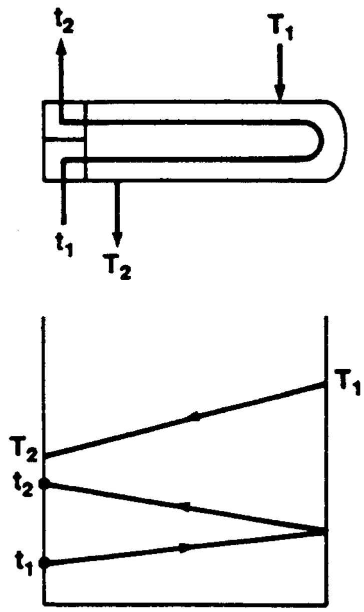
CALCULATION OF F_n

- Depends on the Number of Shells in Series ("Shell Passes")
- The More Shells One Has in Series, the Closer F_n Approaches 1.0
- Typically the Minimum Acceptable Value of F_n is 0.8
- What Exactly Do We Mean by "Shells in Series" or "Shell Passes"?

3.17

CALCULATION OF F_n—SHELL PASSES

ONE PASS

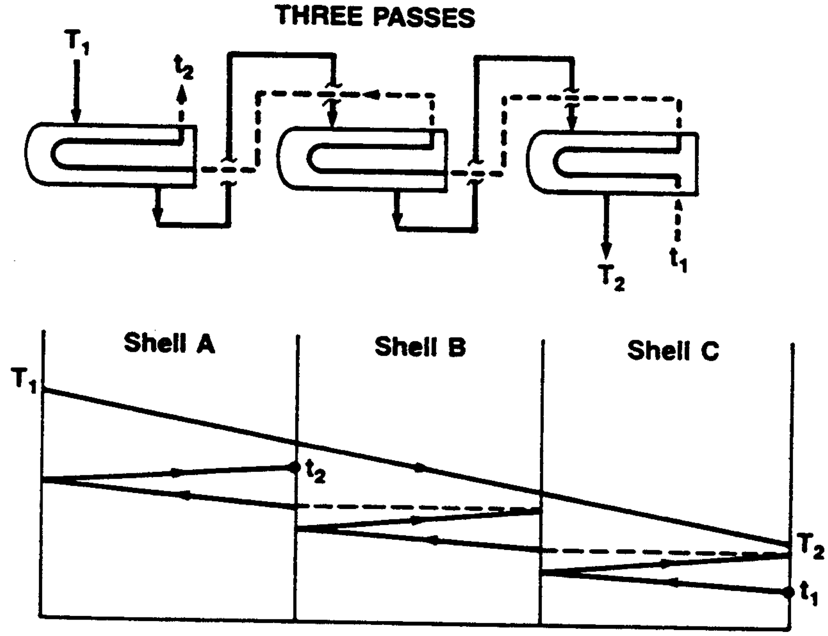


Max $t_2 \cong T_2$

3.18

CALCULATION OF F_n—SHELL PASSES

THREE PASSES

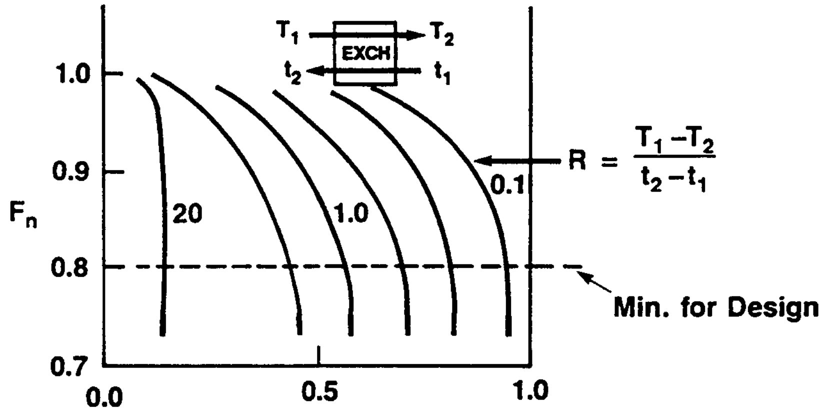


Max $t_2 >> T_2$

3.19

CALCULATION OF F_n

- Complex Equations Simplified to Charts
- See TEMA Section 7, Pages 112-127
  or Exxon DP IX-D, pp53-pp58
- Applicable Only to Linear Heat Curves



$$P = j = \frac{t_2 - t_1}{T_1 - t_1} = \text{“Temperature Efficiency”}$$

CALCULATION OF F_n

Example

$T_1 = 300$ $\quad$ $t_1 = 85$

$T_2 = 105$ $\quad$ $t_2 = 115$

$$P = j = \frac{115-85}{300-85} = 0.14$$

$$R = \frac{300-105}{115-85} = 6.5$$

F_n (1 Shell) = <0.8 (Unreadable on Chart)–Unacceptable

F_n (2 Shells) = 0.95 Use Two Shells

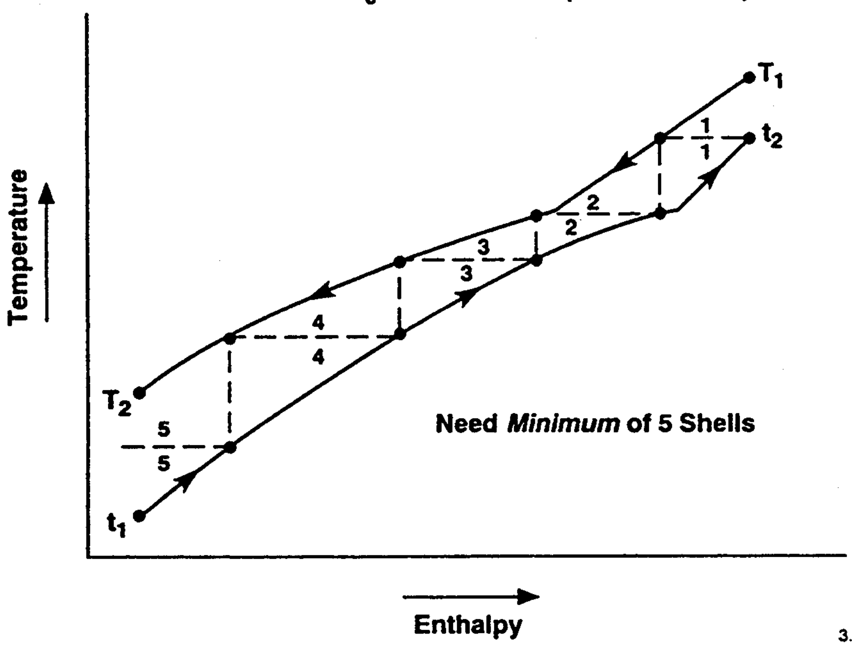
CALCULATION OF F_n

- Since This Technique is Applicable Only to the Case of Straight-Line Heat Release, How Do We Estimate Number of Shells and MTD_e for Other Cases?

3.22

NON-LINEAR HEAT RELEASE—MTD_e SUGGESTION FOR COMPLEX CASES SUCH AS REFORMER FEED/EFFLUENT

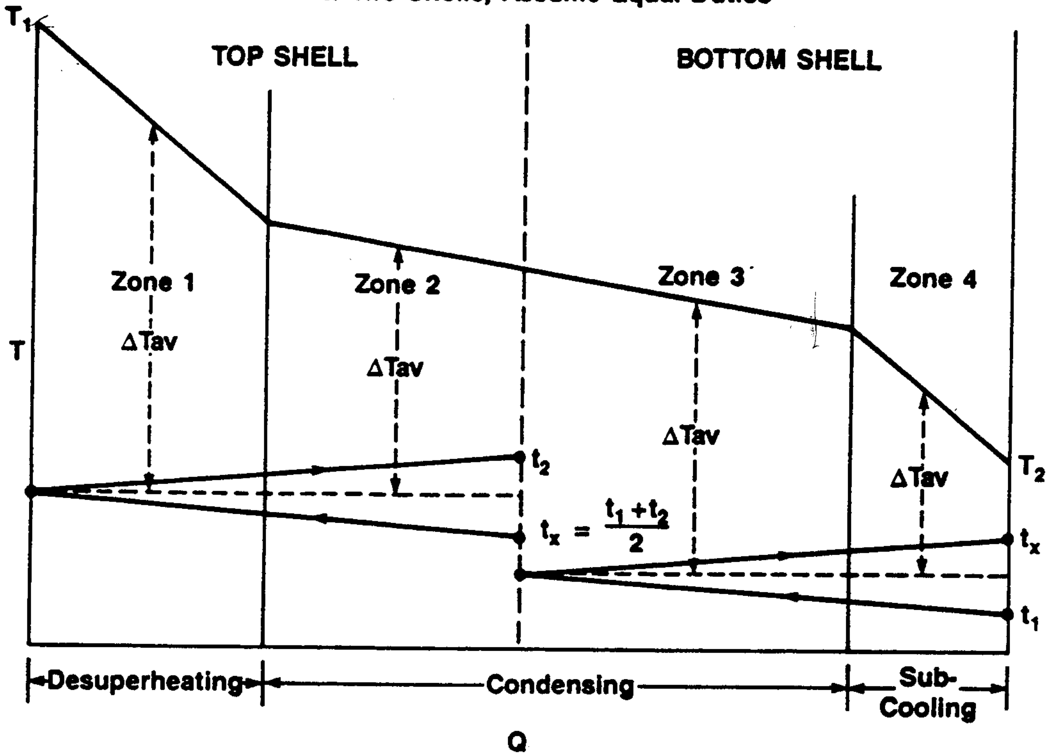
— Plot T vs. Enthalpy
— Step Off to Get *Minimum* Number of Shells
— Calculate MTD_e for Each Shell (Discuss Later)



3.23

NON-LINEAR HEAT RELEASE—MTD_e SUGGESTION FOR CONDENSERS

—Plot the Condensing Curve
—Assume Cold Side is Linear and Draw in Cold Side Flow Pattern
—If Two Shells, Assume Equal Duties



3.24

MTD_e FOR CONDENSERS (Cont'd.)

- Calculate the LMTD for Each Zone, Assuming That the Cold Temperature in Each Zone is the Average of the Inlet/Outlet Cold Temperatures of the Shell in Which the Zone Occurs (See Graph)
- Then Weight the Overall MTD_e as Follows:

$$MTD_e \text{ (Weighted)} = \frac{\text{Qtotal}}{\frac{\text{Qzone1}}{LMTD_1} + \frac{\text{Qzone2}}{LMTD_2} + \frac{\text{Qzone3}}{LMTD_3} + \frac{\text{Qzone4}}{LMTD_4}}$$

HEAT TRANSFER COEFFICIENTS

- **Film Coefficients are Relatively Easy to Estimate: They are a Function of**
  - **Reynolds Number** $\frac{DV\rho}{\mu}$
  - **Prandtl Number** $\frac{(C_p)(\mu)}{K}$

  See Addendum 4.01

- **Similarly, Pressure Drop is a Function of Reynolds Number and Length of Flow Path.**

3.26

HEAT TRANSFER COEFFICIENTS (Cont'd.)

- **The Handouts Just Examined are Suitable *ONLY* for *Estimates* of Coefficients.**
- **For Detailed Coefficients on Which to Base the Purchase of an Exchanger, Detailed Computer Calculations are Necessary.**
- **Detailed Computer Calculations Examine the Effects of Many Other Parameters, Particularly Shell-Side Effects Such as Channelling and Baffle Leakage.**

3.27

EXCHANGER DESIGN PROCEDURE

First Need to Know:

- **Permissible Tube Sizes—Diameter, Gauge, Length. (Frequently Set by Refinery Maintenance Department)**
- **The Appropriate Tube Material for the Service**
- **The Allowable System Pressure Drops for Each Stream.**

EXCHANGER DESIGN PROCEDURE (Cont'd.)

(1) Assume an Overall Heat Transfer Coefficient (U_o) and Calculate Area.

(2) Using the Required Tube Size and Length, Calculate the Number of Tubes.

(3) Using a "Reasonable" Tube-Side Velocity (2-15 Ft/s), Calculate the Tube Side Flow Area Required:

$$A = \frac{ft^3/s}{ft/s}$$

(4) Determine the *EVEN* Number of Tube Passes Which Will Most Closely Approximate the Needed Flow Area

EXCHANGER DESIGN PROCEDURE (Cont'd.)

(5) Calculate the Bundle Diameter

(6) Using a "Reasonable" Value of Shell-Side Velocity, Calculate the Flow Area Required Between Shell-Side Baffles (Gives Baffle Spacing)

(7) Calculate Tube-Side and Shell-Side Pressure Drop. If Satisfactory, Continue to Step 8. If Not, Modify the Exchanger Geometry Until Pressure Drop Requirements are Met.

(8) Calculate the Overall Coefficient U_o

(9) Compare [U_o(Calculated) • A • MTD_e] With the Required Value of Q. If It Doesn't Agree Within About 10%, Then Change Exchanger Geometry and Repeat Calculations

HEAT EXCHANGER DESIGN

- There are Several Major Types of Heat Exchanger Used in Refineries/Chemical Plants:
  - + Shell-and-Tube
  - + Air-Fin Coolers
  - + Double-Pipe
  - + Plate and Frame
- The Vast Majority are S & T
- We Will Briefly Review Usage of the Minor Types and Then Concentrate on the Features of Shell-and-Tube Exchangers

4.01

AIR COOLED EXCHANGERS

- Used for Cooling High to Medium Temperature Streams Where Heat Recovery is Not Practical
- Consists of Tube Bundle and Motor Driven Fans
- Can Be Forced or Induced Draft
- Can Be Countercurrent or Cocurrent to Air Flow
- Tubes are Usually Equipped With Circumferential Fins
- Design Outlet Temperature is Limited by Ambient Air Temperature
- Detailed Design of Air-Fins is Left to the Individual Vendors. Process Designers Simply Provide Duty Specification

4.02

DOUBLE PIPE

- Consists of One or More Pipes Within a Larger Pipe
- Internal Pipes Can Be Bare Surface or Have Longitudinal Fins
- True Cocurrent or True Countercurrent Flow Can Be Achieved
- Available in Standard Off-The-Shelf Sizes
- Several Standard Units May Be Connected in Series or in Parallel
- Not Usually Economical Where Surface Requirements Exceed About 300 Square Feet
- Especially Suited for High-Pressure Applications

4.03

PLATE AND FRAME

- Consists of a Series of Alternating Corrugated Plates Pressed Together in a Compression Frame
- Process Fluids Flow on Alternate Sides of the Plates in Channels Formed by the Corrugations
- Units Achieve True Countercurrent Flow

4.04

PLATE AND FRAME (Cont'd)

ADVANTAGES

- True Countercurrent Flow
- Highly Compact—Take Up Much Less Space Than an Equivalent S & T
- Much Less Expensive Than S & T
- Very Small Holdup of Process Fluids
- No Cross Contamination of the Two Fluids

DISADVANTAGES

- Limited to Moderate Temperatures and Pressures (Up to About 300°F and 150 psig)
- Some Hydrocarbon Streams Attack the Interplate Gasketing
- Require Great Time in Assembly/Disassembly
- Best Suited to Aqueous Streams, e.g. Amines, Water

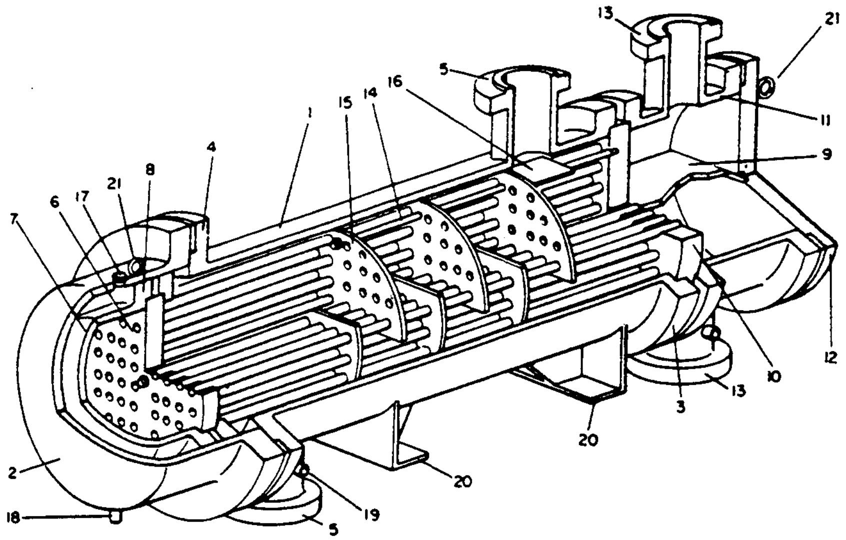
4.05

SHELL AND TUBE

- Most Common Type in Refinery Service
- Consists of Tube Bundle Within External Shell
- Not Truly Cocurrent or Countercurrent

4.06

NOMENCLATURE
Components of Shell and Tube Exchangers



1. SHELL
2. SHELL COVER
3. SHELL CHANNEL
4. SHELL COVER END FLANGE
5. SHELL NOZZLE
6. FLOATING TUBESHEET
7. FLOATING HEAD
8. FLOATING HEAD FLANGE
9. CHANNEL PARTITION
10. STATIONARY TUBESHEET
11. CHANNEL
12. CHANNEL COVER
13. CHANNEL NOZZLE
14. TIE RODS AND SPACERS
15. TRANSVERSE BAFFLES OR SUPPORT PLATES
16. IMPINGEMENT BAFFLE
17. VENT CONNECTION
18. DRAIN CONNECTION
19. TEST CONNECTION
20. SUPPORT SADDLES
21. LIFTING RING

4.07

MAJOR TYPES OF S & T UNITS

- Fixed Tube Sheet (Uncommon)
- Floating Tube Sheet
  + Pull-Through Floating Head
  + Split Ring Floating Head
- U-Tube

4.08

SHELL & TUBE EXCHANGERS

Fixed Tube Sheet—
Cleanest. Consider Only When Shell Side Fouling Factor $\leq$0.002 & Shell Side Can be Chemically Cleaned.

- Because of Thermal Stresses, This Type is Generally Unacceptable if the Average Shell Temperature and Average Tube Temperature Differ by More Than 50°F

U-Tube—
Least Expensive for High Tubeside Design Pressure. Normally Used When Tubeside Fouling $\leq$ 0.002. (Except for Water)

Split Ring Floating Head—
This Type is Normally Specified Unless Very Frequent Mechanical Cleaning is Required

Pull-Through Floating Head—
Most Expensive Type of S & T Unit; Thermally Inefficient Because of Shell Bypassing. Use When Both Sides Must be Mechanically Cleaned.

4.09

PRELIMINARY DECISIONS: DESIGN OF SHELL-AND-TUBE UNITS

- Which Fluid to Put in the Tubes
- Tube Nominal Diameter, Wall Thickness and Material
- Tube Length
- Tube Layout
- Baffle Orientation
- Baffle Pitch (Spacing)
- Maximum Bundle Diameter (Bundle Weight)

4.10

TUBE SIDE FLUID

- Between the Two Streams, the Stream With the Higher:
  - + Pressure
  - + Corrosion Rate
  - + Fouling Rate

  Is Usually Placed on the TUBE SIDE.

  When These Characteristics Apply to Both Streams, the Designer Uses His Judgement.
- In a Service Where One Stream is Changing Phase, That Stream is Assigned to the SHELL SIDE.
- In Steam-Heated Vaporizers/Reboilers, the Condensing Steam is Placed in THE TUBES.
- Streams With Very HIGH VISCOSITY are Placed on the SHELL SIDE (Better Coefficient).

4.11

TYPICAL TUBE DIAMETERS/WALL THICKNESSES

1. Oil Service—Ferrous Tubes

Severity of Service	OD, in.	Layout and Spacing, in.	BWG	Thickness, in.
Non-Fouling or Fouling (<0.003), Midly Corrosive	3/4	15/16 △	14	0.083
Non-Fouling or Fouling (<0.003), Corrosive	3/4	1.00△ or □	12	0.109
Extremely Fouling (≥0.003), Midly Corrosive	1	1.25△ or □	12	0.109
Extremely Fouling (≥0.003), Corrosive	1	1.25△ or □	10	0.134

2. General Service Alloy Tubes

Severity of Service	OD, in.	Layout and Spacing, in.	BWG	Thickness, in.
Water Service—Nonferrous Tubes	3/4	15/16△or 1.0□	16	0.065
Non-Fouling or Fouling (<0.003)	3/4	15/16△ or 1.0□	16	0.065
Extremely Fouling (≥0.003)	1	1.25□ or △	14	0.083

4.12

TUBE LENGTH

- Refinery Decision (Local Preference)
- Most Common Length is 20 Feet.
- Occasionally, 16′ Length is Used.
- For Special Situations, 8′ and 10′ Can be Considered.
- Longer Tube Bundles Require More Plot Area for Bundle Removal. Longer Bundles are Also More Difficult to Extract From the Shell and to Handle.

TUBE LAYOUT

3 Main Layouts—

□ Square— 1. Use When $r_o > 0.002$ & Shellside Must be Mechanically Cleaned.
2. Reboilers/Vaporizers

◇ Rotated Square—Use as Square, But Only When Flow is Laminar

Triangular 30°—
1. Use When $r_o \leq 0.002$
2. Cheapest so Use When Applicable

TYPE OF BAFFLE

- Segmental—Most Common
- Double Segmental (Modified Disk and Donut) is Used to Obtain Very Low Shell-Side Pressure Drop
- Tube Supports Only—No Real Baffles. Occasionally Used in Certain Reboiling or Condensing Services

4.15

BAFFLE ORIENTATION & CUT

Vertical Chord—Most Common
- Condensers, Vaporizers, & Fluids Containing Suspended Solids
- Flow is Side-to-Side

Horizontal Chord—
- Sediment-Free Fluids Being Cooled Through High Temperature Range (200 to 300°F) in One Shell
- Flow is Over-Under

Baffle Cut
- This is the Percent of the Baffle Which is Cut Away to Permit Flow
- Typical Cut is 25% (40% for Double Segmental Baffles)

4.16

BAFFLE PITCH

- Minimum Allowable Spacing (Pitch) is 20% of the Shell ID or Two Inches, Whichever is Greater

- Maximum Allowable Pitch:
  + For No Change of Phase, Equals Shell ID
  + For Change of Phase

Tube Size	Steel	Copper Alloys
¾″	30″	26″
1″	37″	32″
1½″	50″	43.5″

4.17

TEMA

"Tubular Exchanger Manufacturer's Association"

- This is the Basic Industrial Standard for Shell-and-Tube Exchangers

- Covers Heavy-Duty Type ("TEMA R") as Well as the Lighter Duty "TEMA C" Units

- Latest Edition is the Seventh, Dated 1988

4.18

TEMA TYPE

- "TEMA Type" Followed by Three Letters Refers to the Type of
  - + Front End (Channel) Arrangement
  - + Shell Nozzle/Baffle Arrangement
  - + Rear End (Floating Head End) Arrangement
- These Three Characteristics are Each Identified by a Single Letter of the Alphabet
- The Result, for Example, Would be the Entry "TEMA Type AES" in the Specification for the Heat Exchangers. The Type MUST be Specified

4.19

MOST COMMON TEMA TYPES

Front End (Channel) Arrangement

A
- Removable Channel With Removable Cover Plate
- May Be Used With Fixed or Removable Tube Bundles
- Tube Cleaning Easier Since No Piping Disassembly Required
- Flanged Channel End is Costly and Prone to Leakage
- Most Commonly Used

B
- Removable Channel With Integral Cover
- May be Used With Fixed or Removable Tube Bundles
- Used for Low Tube Side Fouling Services or Where Chemical Cleaning is Specified. Mechanical Cleaning Requires Piping Disassembly
- Less Costly and Less Prone to Leakage Than Type A

C
- Channel Integral With Tubesheet and With Removable Cover
- Two Types: Removable Bundle and Fixed Bundle

4.20

MOST COMMON TEMA TYPES (Cont'd)

Shell Types

E
- One Pass Shell
- Most Common Type Used

F
- Two Pass Shell With Longitudinal Baffle
- Used to Improve Cross Flow Correction Factor
- Equivalent to Two Shells in Series
- Max. Shellside Pressure Drop of 10 PSI
- Max. Shellside Temperature Range of 350 Deg. F

G/H
- Split Flow Arrangements
- Use Interanl Baffles to Split the Shellside Flow
- Used to Minimize Pressure Drop

J
- Divided Flow
- Also used to Minimize Pressure Drop
- No Internal Baffle

K
- Kettle Type
- Used for Vaporizing Services (Reboilers, Steam Generators and Refrigeration Services)

X
- Cross Flow
- No Baffles
- Low Pressure Drop

MOST COMMON TEMA TYPES (Cont'd)

Rear End Head

S
- Floating Tubesheet Sandwiched Between Split Ring and Tubesheet Cover
- Tubesheet Assembly Moves Within Shell Cover to Absorb Expansion of the Tubes
- Requires Removing Rear Shell Cover and Floating Tubesheet Cover for Bundle Removal, But Results in a Smaller Diameter Shell for the Same Heat Transfer Surface
- Usually First Choice for Removable Bundles if Mechanical Cleaning of Shell Side Will be Infrequent

T
- Pull Through Floating Head
- Floating Tubesheet Cover Bolted Directly to Floating Tubesheet
- Does Not Require Rear Head Disassembly for bundle Removal
- Results in Larger Diameter Shell for Same Heat Transfer Surface Than Type S
- Preferred Where Frequent Mechanical Cleaning of Shellside is Anticipated

U
- U-Tube Bundle
- No Floating Head. Tube Bundle Consists of U-Tubes
- Not Recommended Where Mechanical Cleaning of Tube Side is Anticipated
- Good for High Pressure, Clean Services or Where Chemical Cleaning of Tubeside is Specified

MOST COMMON TEMA TYPES (Cont'd)

- **Therefore a TEMA AES Exchanger Has**

  A = Removable Channel and Removable Channel Cover Plate

  E = One Pass Shell (One Inlet Nozzle and One Outlet Nozzle)

  S = Split Ring Type Floating Tube Sheet Construction

ADDENDUM 4.01
SECTION 4
PROCESS DESIGN COURSE
HEAT EXCHANGER DESIGN

A SHORTCUT PROCEDURE FOR APPROXIMATE EVALUATION OF SHELL AND TUBE EXCHANGERS WITH NO CHANGE OF PHASE

IMPORTANT NOTE AND WARNING:

- This procedure must not be used for the definitive design of heat exchangers. It is a shortcut technique which makes many simplifying assumptions, especially with regard to shell-side calculations.

- The Reynold's Number used in this Addendum is dimensional.

– INDEX –

SHORTCUT PROCEDURE

SCOPE

The following subsection presents an approximate procedure for evaluating shell and tube exchangers in which there is no change of phase, (i.e., vapor/vapor, vapor/liquid, or liquid/liquid exchangers). The actual calculations can be made on the calculation form, which starts on page 8. Each step of the procedure is explained in the following paragraphs.

DETAILED PROCEDURE

1. Terminal Conditions and Effective Log Mean Temperature Difference

   a. Determine the following temperatures:
      - Inlet temperature of fluid being cooled, T_1
      - Outlet temperature of fluid being cooled, T_2
      - Inlet temperature of fluid being heated, t_1
      - Outlet temperature of fluid being heated, t_2

   b. Determine the log mean temperature difference, Δt_m

   $$\Delta t_m = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln \frac{(T_1 - t_2)}{(T_2 - t_1)}}$$

   c. From Figure 1.01-1.03, determine the minimum number of shells required for a temperature correction factor (F_n) of at least 0.800.

   d. Determine the effective log mean temperature differences, Δt_e

   $$\Delta t_e = F_n \Delta t_m$$

2. Caloric Temperatures

   a. Decide which fluid to pass through the tubes and which through the shell.

   b. Calculate the caloric temperatures.

      For the fluid being heated, t_t or $t_s = 0.4(t_2 - t_1) + t_1$
      For the fluid being cooled, t_s or $t_t = 0.4(T_1 - T_2) + T_2$

3. Caloric Properties of Fluids

   a. Tube Side of Exchanger

      1. At the caloric temperature t_t, determine the following tube side fluid properties:
         - For water: density, m
         - For hydrocarbon liquids or vapors: density, m; viscosity, z.
         - For other fluids: density, m; viscosity, z; specific heat, c; and thermal conductivity, k.

   b. Shell Side of Exchanger

      1. At the caloric temperature, determine the density, m of the shell side fluid.

4. Shell Side and Tube Side Flow Rates

The values of the respective flow rates in lb/hr will normally be determined during the heat and material balance calculations.

5. Fouling Factors

a. Decide the tube side fouling factor r_i (See Table 1.04)
b. Decide the shell side fouling factor r_o (See Table 1.04)

6. Iteration, Tube Side

(1) The heat duty for the exchanger will normally be determined during the heat and material balance calculations.

(2) Assume U_o, the over-all coefficient (See Table 1.05)

(3) Calculate total area.

$$A = Q / U_o \Delta t_e$$

(4) Calculate the area per shell.

$$A_s = A / N_s$$

If necessary, the number of shells should be increased to meet the maximum shell size limitations (typically 48"). This will require recalculating $F_n, \Delta t_e, A, A_s$

(5) Decide the tube metal and determine tube thermal conductivity, k_w (See Table1.03).

(6) Choose the tube length, diameter, wall thickness, pitch, and layout (See Tables 1.01 and 1.02).

(7) Determine the number of tubes as follows:

$$N_T = \frac{3.82\, A_s}{(L - 0.5)d_o}$$

(8) Estimate N_p, the even number of tube passes per bundle which will give a reasonable tube-side velocity (3-20 fps).

(9) Calculate the linear velocity in the tubes and in the nozzles:

$$(d_N = \text{Nozzle ID}) \qquad V = \frac{N_p M}{19.6\, m\, N_T d_i^2}; \qquad V_N = \frac{M}{19.6\, m\, d_N^2}$$

(10) Tube side pressure drop and heat transfer coefficient (for water).

a. Tube side heat transfer coefficient, h_{io} for water from approximately 80°F to180°F.

$$\frac{1}{R_{io}} = h_{io} = \frac{368}{d_o}(Vd_i)^{0.7}\left(\frac{t_t}{100}\right)^{0.26}$$

b. Total tube side pressure drop, ΔP_t, for water at approximately 100°F.

$$\Delta P_t = 0.020 F_t N_s N_p \left[V^2 + 0.158 L \frac{V^{1.73}}{d_i^{1.27}} \right] + \Delta P_N$$

For ΔP_N, See Step 15 (nozzle pressure drop).

(11) For fluids other than water:

a. Calculate the tube side mass velocity, G

$$G = mV$$

b. Calculate tube side Reynold's Number, N_{Re} (dimensional)

$$N_{Re} = \frac{d_i G}{z}$$

<u>Note:</u> At this point, check for a transition problem by calculating N_{Re} using fluid properties at <u>inlet</u> (or outled) conditions. An Exchanger design is not valid if the type of flow conditions changes from viscous to turbulent (or vice-versa) within the unit..

(12) From Figure 2.01 determine the tube side pressure drop correlation factor, Y_{tp}.

(13) Calculate the tube side velocity head and the nozzle velocity head.

$$\frac{mV_N^2}{9270} \text{ in the nozzles } ; \quad \frac{mV^2}{9270} \text{ in the tubes}$$

(14) Calculate ΔP_{tf}, the frictional pressure drop per tube pass.

$$\Delta P_{tf} = Y_{tp} \frac{L}{d_i} \left[\frac{mV^2}{9270} \right] \left[\frac{z_w}{z} \right]^{0.14 \text{ or } 0.25}$$

The exponent 0.14 is for turbulent flow $(N_{Re} < 30)$; 0.25 is for streamline flow $(N_{Re} < 30)$.

(15) Calculate the pressure drop per tube pass due to turns, ΔP_{tr}, and the nozzle pressure drop, ΔP_N.

$$\Delta P_{tr} = 3\left(\frac{mV^2}{9270} \right); \quad \Delta P_N = 2\left(\frac{mV^2}{9270} \right) \text{ (two nozzles)}$$

(16) Calculate the total tube side pressure drop, ΔP_t

$$\Delta P_t = F_t N_s N_p \left(\Delta P_{tf} + \Delta P_{tr} \right) + \Delta P_N.$$

For: F_t, see Table 1.01.

If the pressure drop is reasonably close to the value desired, proceed to the next step. If it seems too high or low, change number of tube passes and repeat steps 9 through 16 until the pressure drop is satisfactory.

(17) From Figure 2.02, determine the heat transfer correlation factor, Y_{th}.

a. Calculate the thermal function:

$$k\left[\frac{cz}{k}\right]^{0.33}$$

For hydrocarbons, refer to Figures 4.01 and 4.02.

b. Calculate the tubeside heat transfer coefficient, h_{io}

$$\frac{1}{R_{io}} = h_{io} = \frac{Y_{th}}{d_o}k\left[\frac{cz}{k}\right]^{0.33}\left[\frac{z}{z_w}\right]^{0.14}$$

Initially assume $\left[\frac{z}{z_w}\right]^{0.14} = 1$, until tube wall temperature is calculated.

c. Estimate the average tube wall temperature, t_w

$$t_w = t_t + U_o(R_{io} + r_{io})(t_s - t_t)$$

d. At the average tube wall temperature, determine z_w and calculate:

$$\left[\frac{z}{z_w}\right]^{0.14}$$

(18) Recalculate h_{io} using this viscosity correction.

(19) Calculate the tube wall resistance, r_w

$$r_w = \frac{\ell}{12 \cdot k_w}$$

(See Tables 1.02 - 1.03)

7. <u>Iteration, Shell Side</u>

(1) Estimate t_f, the average shell side film temperature.

$$t_f = \frac{(t_s + t_t)}{2} + \frac{(U_o)}{2}(R_{io} + r_{io} + r_w + r_o)(T_s - t_t)$$

(2) At the average shell side <u>film</u> temperature, determine the following shell fluid properties:

a. For hydrocarbon liquids or vapors: Viscosity, z_f.

b. For other fluids: Viscosity, z_f; specific heat, c_f; and thermal conductivity, k_f.

(3) Determine the number of tubes across the centerline of the tube bundle, N_{TC}.

- For square tube layout:

$$N_{TC} = 1.19\,(N_T)^{0.5}$$

- For triangular layout:

$$N_{TC} = 1.10\,(N_T)^{0.5}$$

(4) Determine the outer tube limit, D_t.

$$D_t = (N_{TC} - 1)(P_t) + d_o$$

(5) Determine shell I.D. as follows:

$D = D_t / 0.9$; except for the following limitations:

1. Minimum $D = D_t + 1$
2. Maximum $D = D_t + 3$

(6) Determine the free width for fluid flow normal to and around the tubes.

One shell pass, $W = D - (d_o N_{TC})$; Two shell pass, $W = \dfrac{D - (d_o N_{TC})}{2}$

(7) Estimate the baffle pitch P_b which will give a reasonable shell-side velocity (3-15 fps). See Table 1.01 for maximum P_b.

(8) Calculate the number of shell side baffles, N_B (always a whole number).

$$N_B = 10L / P_b$$

(9) Determine the free area, S, for fluid flow across the tube bundle between each pair of baffles.

	For Calculating the Film Coefficient, h	For Calculating the Pressure Drop, ΔP
Segmental Baffles:	$S = W\,(P_b - 0.375)$	$S = W\,(P_b - 0.375)$
Modified Disc & Donut Baffles:	$S = W(P_b - 0.375)$	$S = 0.85\,W\,(P_b - 0.375)$

In each case, 0.375 in. represents the approximate baffle thickness.

(10) Calculate the shell side mass velocity, G.

Disc and donut baffles, G = M/50•S; Segmental baffles, G = M/25•S

(11) Calculate the shell side linear velocity, V and the shell side nozzle velocity, V_N

$$V = G / m$$

$$V_N = \frac{M}{19.6 m d_N^2} \quad (d_N = \text{Nozzle ID})$$

(12) Calculate the shell side Reynold's number, N_{Re}.

$$N_{Re} = d_o G / z_f$$

(13) Calculate the ratio of the tube diameter to the tube spacing:

$$\frac{d_o}{P_t - d_o}$$

From Figure 3.01 determine the shell side pressure drop correlation factor, Y_{SP}.

Total Shell Side Pressure Drop

(14) Calculate the shell side velocity head and the nozzle velocity head.

$$\frac{mV_N^2}{9270} \text{ in the nozzles } ; \frac{mV^2}{9270} \text{ in the shell.}$$

(15) Calculate ΔP_{sf}, the frictional pressure drop per shell. Table 1.01 gives values for B_2.

$$\Delta P_{sf} = B_2 Y_{sp} N_{TC} N_B \left(\frac{mV^2}{9270} \right)$$

(Note!: For Disc & Donut baffles, divide N_{TC} by 2.0)

(16) Calculate the pressure drop per shell due to turns, ΔP_{sr}, and the nozzle pressure drop, ΔP_N.

$$\Delta P_{sr} = (N_B + 1) \left(3.5 - \frac{2P_b}{D} \right) \left(\frac{mV^2}{9270} \right) ; \ \Delta P_N = 2 \left(\frac{mV_N^2}{9270} \right)$$

(17) Calculate the total shell side pressure drop, ΔP_s.

$$\Delta P_s = F_s N_s \left(\Delta P_{sr} + \Delta P_{sf} \right) + \Delta P_N$$

For F_s, see Table 1.01.

If the pressure drop is reasonably close to the desired value, proceed to the next step. If it seems too high or low, change the baffle pitch P_b and repeat steps 7 through 17 until the pressure drop is satisfactory.

Shell Side Heat Transfer Coefficient, h_o

(18) From Figure 3.02 determine the heat transfer correlation factor, Y_{sh}.

a. Calculate the thermal function:

$$k_f \left(\frac{c_f z_f}{k_f} \right)^{1/3}$$

(For hydrocarbon liquids or vapors, refer to Figures 4.01 and 4.02)

b. Calculate the correction factor for the deviation from ideal baffle pitch.

$$\left(\frac{4P_b}{D} \right)^{0.1}$$

$$\frac{1}{R_o} = h_o = B_1 \frac{Y_{sh}}{d_o} k_f \left(\frac{c_f z_f}{k_f} \right)^{1/3} \left(\frac{4P_b}{D} \right)^{0.1}$$

See Table 1.01 for B_1.

8. Duty Coefficient

Calculate U_o, the over-all duty heat transfer coefficient.

$$\frac{1}{U_o} = R_t = R_{io} + r_{io} + R_o + r_w + r_o$$

If U_o calculated does not agree with U_o assumed, repeat the calculations with a new U_o assumed until agreement is reached (±10%).

9. Clean Coefficient

Calculate U_c, the over-all clean coefficient.

$$\frac{1}{U_C} = R_C = R_{io} + r_w + R_o + 0.001$$

10. Design Temperatures

Determine the following mechanical design features:

1. The design temperature and pressure of the shell and tube sides.
2. The nozzle size and flange rating for the inlets and outlets on both the shell and tube sides.
3. The design temperature of the tube sheet, T_M.
   a. For coolers (water on tube side), specify the higher result of the following equations:

$$T_M = T_{DT} + \frac{R_{io}}{R_C}(T_{DS} - T_{DT})$$

or

$$T_M = T_{DT} + \frac{(R_{io} + r_{io})}{R_t}(T_{DS} - T_{DT})$$

   b. For other exchangers:

   (1) When the fluid being cooled is on the tube side

$$T_{M.} = T_{DT} - 0.1(T_{DT} - T_{DS})$$

   (2) When the fluid cooled is on the shell side

$$T_M = T_{DT} + 0.3(T_{DS} - T_{DT})$$

TABLE 1.01

DESIGN CONSTANTS FOR SHELL AND TUBE EXCHANGER CALCULATIONS

SHELL SIDE

Maximum Allowable Baffle Pitch

Tube O.D., Inches	Maximum P_b, Inches: Steel	Maximum P_b, Inches: Copper, Aluminum Alloys
0.75	30.0	26.0
1.00	37.0	32.0
1.50	50.0	43.5

(For no change of phase, P_b should not exceed the shell ID.

Heat Transfer & Pressure Drop Factor B_1 and B_2

Baffle Position	Tube Layout	Transfer B_1	Pressure Drop B_2
Vertical to tube rows	Square	0.50	0.30
On the bias (45°)	Square	0.55	0.40
Vertical to tube rows	Triangular	0.70	0.50

Pressure Drop Fouling Factors, F_s

Fluid	F_s
Liquids	1.15
Gases or condensing vapors	1.00

TUBE SIDE

Pressure Drop Fouling Factors

Tube O.D., Inches	F_t
0.75 Steel	1.50
1.00 Steel	1.40
1.50 Steel	1.20
0.75 Copper based	1.20
1.00 Copper based	1.15

Typical Tube Pitch

Tube O.D., In.	Pitch, In
0.75 Triangular	0.9375
0.75 Square	1.0
1.0 Square	1.25
1.5 Square	1.875

Design Cooling Water Velocity

Material	Type of Water	Most Favorable Velocity, ft/sec	Permissible Range, ft/sec (4)
Carbon steel	Fresh, non-inhibited	4	3 to 6
	Fresh, inhibited	6 to 8	3 to 10
Red brass	All types	3	3 to 4
Admiralty (inhibited)	Fresh (inhibited or not)	6 to 8	3 to 10
	Salt or brackish	3	3 to 5
Aluminum brass	Fresh (inhibited or not)	6 to 8	3 to 10
	Salt or brackish	5	4 to 8
Cupronickel (70-30)	All types	7 to 8	6 to 12
Cupronickel (90-10)	All types	7 to 8	6 to 12
Monel	All types	8	6 to 12
Type 316 alloy steel	All types	10	8 to 15

Table 1.02

EXCHANGER TUBE DATA

do = O.D. of Tubing, In.	BWG	ℓ = Wall Thickness, In. (3)	di = I.D. of Tubing, In.	Internal Cross Sectional Area Sq. In.	External Surface Per Foot Length Sq Feet
3/4	12	0.109	0.532	0.223	0.1963
3/4	14	0.083 (1)	0.584	0.268	0.1963
3/4	16	0.065 (2)	0.620	0.302	0.1963
3/4	18	0.049	0.652	0.334	0.1963
1	10	0.134	0.732	0.421	0.2618
1	12	0.109 (1)	0.782	0.479	0.2618
1	14	0.083 (2)	0.834	0.546	0.2618
1	16	0.065	0.870	0.594	0.2618
1 1/2	10	0.134	1.232	1.192	0.3927
1 1/2	12	0.109	1.282	1.291	0.3927
1 1/2	14	0.083	1.334	1.397	0.3927

GAGE EQUIVALENTS

Inches	BWG
0.220	5
0.165	8
0.148	9
0.134	10
0.120	11
0.109	12
0.095	13
0.083	14
0.072	15
0.065	16
0.058	17
0.049	18
0.035	20

MAXIMUM RECOMMENDED NUMBER OF TUBE PASSES

Shell ID Inches	Max. Passes
<10	4
10-19	6
20-29	8
30-39	10
40-49	12
51-59 (Rare)	14

Notes:

(1) Typical wall thickness for carbon steel tubes.

(2) Typical wall thickness for copper alloy tubes.

(3) Average wall thickness is typically 10% greater than the minimum wall thickness. Tubes may be specified (and purchased) on either an average wall or minimum wall basis. (Exxon normally specifies minimum wall.)

Table 1.03
THERMAL CONDUCTIVITIES OF METALS AT TYPICAL HEAT EXCHANGER TEMPERATURES

Material	Composition	Thermal Conductivity, k, Btu/hr sq ft °F/ft
Admiralty	(71 Cu - 28 Zn - 1 Sn)	64
Type 316 Stainless Steel	(17 Cr - 12 Ni - 2 Mo)	9
Type 304 Stainless Steel	(18 Cr - 8 Ni)	9
Brass	(70 Cu - 30 Zn)	57
Red Brass	(85 Cu 15 Zn)	92
Aluminum Brass	(76 Cu - 22 Zn - 2 Al)	58
Cupro-Nickel	(90 Cu - 10 Ni)	41
Cupro-Nickel	(70 Cu - 30 Ni)	17
Monel	(67 Ni - 30 Cu - 1.4 Fe)	15
Inconel		11
Aluminum		117
Carbon Steel		26
Carbon-Moly Steel	(0.5 Mo)	25
Copper		223
Lead		20
Nickel		36
Titanium		11
Chrome-Moly Steel	(1 Cr - 0.5 Mo)	24
	(2-1/4 Cr - 0.5 Mo)	22
	(5 Cr - 0.5 Mo)	20
	(12 Cr - 1 Mo)	16

Table 1.04
TYPICAL FOULING FACTORS

Stream Type	Typical r_i or r_o
Vapor Overheads	0.001
Virgin distillate liquids to tankage	0.001
Virgin distillate liquids from tankage	0.002
Cracked distillate liquids to tankage	0.002
Reduced Crudes	0.004
Tar, bitumen	0.005
Cracked Tar	0.010
Crudes	0.002 - 0.004
Steam	0.001
BFW	0.001
Cooling Water, Fresh	0.0015 - 0.0025
Cooling Water, Salt	0.0025 - 0.0035

Table 1.05

SOME TYPICAL OVERALL COEFFICIENTS

Type of Source	Typical U_o
Light Ends Liquid Coolers (Water)	120
Distillate Coolers (Water)	70-90
Light Ends Reboilers (Steam)	80
Light Ends Feed/Bottoms	100
Crudes/distillates	25-50
Condensers (Tower overheads)	90

NOMENCLATURE

A – Total exchanger area, ft^2
A_s – Area/shell, ft^2
B_1 – Bundle factor for shell side beat transfer, dimensionless.
B_2 – Bundle factor for shell side pressure drop, dimensionless.
c – Specific beat at caloric temperature, Btu/lb -°F.
c_f – Specific beat of the shell side fluid at average film temperature, Btu/lb - °F.
D – Shell I.D., inches
D_t – Diameter of tube bunde ("outer tube limit"), inches
d_i – Tube I.D., inches
d_o – Tube O.D., inches
F_n – Correction factor for log mean temperature difference (due to partially concurrent flow), dimensionless.
F_s – Shell side pressure drop correction factor, dimensionless.
F_t – Tube side pressure drop correction factor, dimensionless.
G – Mass velocity, lbs/sec - ft^2
h_{io} – Inside film coefficient corrected to outside area, Btu/hr-ft^2 -°F.
h_o – Outside film coefficient Btu/hr-ft^2 -°F.
k – Thermal conductivity at caloric temperature, Btu/hr-ft^2 -°F/ Ft.
k_f – Thermal conductivity of the shell side fluid at average film temperature, Btu/hr-ft^2 °F/ Ft.
k_w – Thermal conductivity of the tube metal at average tube temperature, Btu/hr-ft^2 -°F/ Ft.
ℓ – Tube wall thickness, in.
L – Tube length, ft.
M – Mass rate, lbs/hr.
m – Density, lbs/ft^3
N_B – Number of shell baffles.
N_P – Number of tube passes per shell.
N_{Re} – Reynolds number, inch-lbs/sec - ft^2 - centipoise
N_S – Number of shells in series.
N_T – Number of tubes in the bundle.
N_{TC} – Number of tubes across the center line of the bundle.
P_b – Baffle pitch, inches
P_t – Tube pitch, inches.
Q – Rate of heat transfer, Btu/hr.
R_C – Total resistance (clean) to heat transfer (Note 1)
R_{io} – Inside film resistance corrected to outside area, (Note 1)

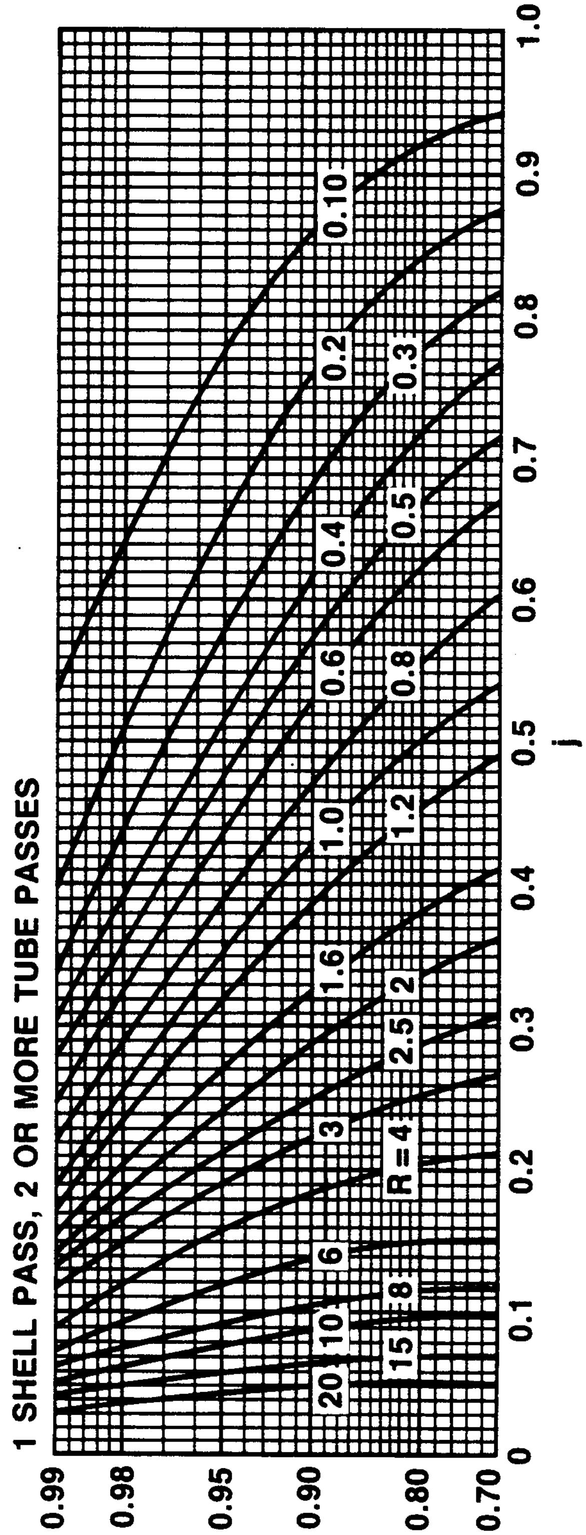
Notes: (1) All resistances are hr-ft^2-°F/Btu

R_o – Outside film resistance to heat transfer, (Note 1).
R_t – Total resistance (duty) to heat transfer (Note 1)
r_{io} – Inside fouling factor corrected to outside area, (Note 1)
r_o – Outside fouling factor (Note 1).
r_w – Resistance of tube wall metal at average wall temperature (Note 1).
S – Free flow area between shell baffles, $in.^2$
T_{DS} – Design temperature of the shell side, °F.
T_{DT} – Design temperature of the tube side, °F.
T_M – Tube sheet design temperature, °F.
T_1 – Inlet temperature of fluid being cooled, °F.
T_2 – Outlet temperature of fluid being cooled, °F.
t_1 – Inlet temperature of fluid being heated, °F.
t_2 – Outlet temperature of fluid being heated, °F.
t_f – Average shell side film temperature, °F.
t_s – Caloric temperaature of the shell fluid, °F.
t_t – Caloric temperature of the tube fluid, °F.
t_w – Average tube wall temperature, °F.
U_c – Over-all clean coefficient of heat transfer, Btu/hr-ft^2 -°F.
U_o – Over-all duty coefficient of heat transfer, Btu/hr-ft^2 -°F.
V – Velocity in the tubes or shell ft/sec.
V_N – Velocity in the nozzles, ft/sec.
W – Free width between baffles, in.
Y_{sh} – Shell side heat transfer correlation factor.
Y_{sp} – Shell side pressure drop correlation factor.
Y_{th} – Tube side heat transfer correlation factor.
Y_{tp} – Tube side pressure drop correlation factor.
z – Viscosity at caloric temperature, centipoises.
z_f – Viscosity of the shell side fluid at average film temperature, centipoises.
z_w – Viscosity of the tube side fluid at tube wall temperature, centipoises,
ΔP_{tf} – Tube pressure drop due to friction, psi/tube pass.
ΔP_{tr} – Tube pressure drop due to turns, psi/tube pass.
ΔP_t – Total tube side pressure drop, psi.
ΔP_{sf} – Shell side pressure drop due to friction, psi/shell
ΔP_{sr} – Shell side pressure drop due to turns, psi/shell.
ΔP_N – Nozzle Pressure drop, psi/shell.
ΔP_s – Total shell side pressure drop, psi.
Δt_e – Long mean temperature difference corrected for non-ideal countercurrent flow ("Effective temperature difference") °F.
Δt_{ew} – Weighted effective log mean difference, °F.

FIGURE 1.01

FIGURE 1.01

LMTD CORRECTION FACTORS



$$R = \frac{T_1 - T_2}{t_2 - t_1} \qquad j = \frac{t_2 - t_1}{T_1 - t_1}$$

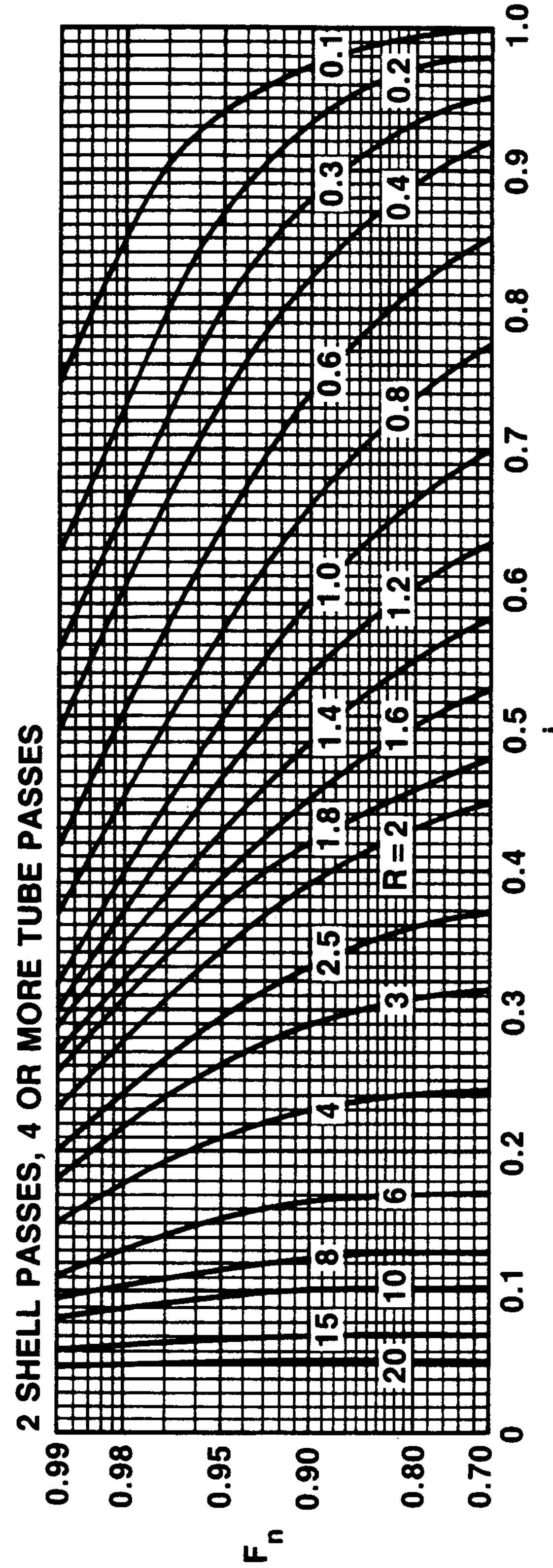
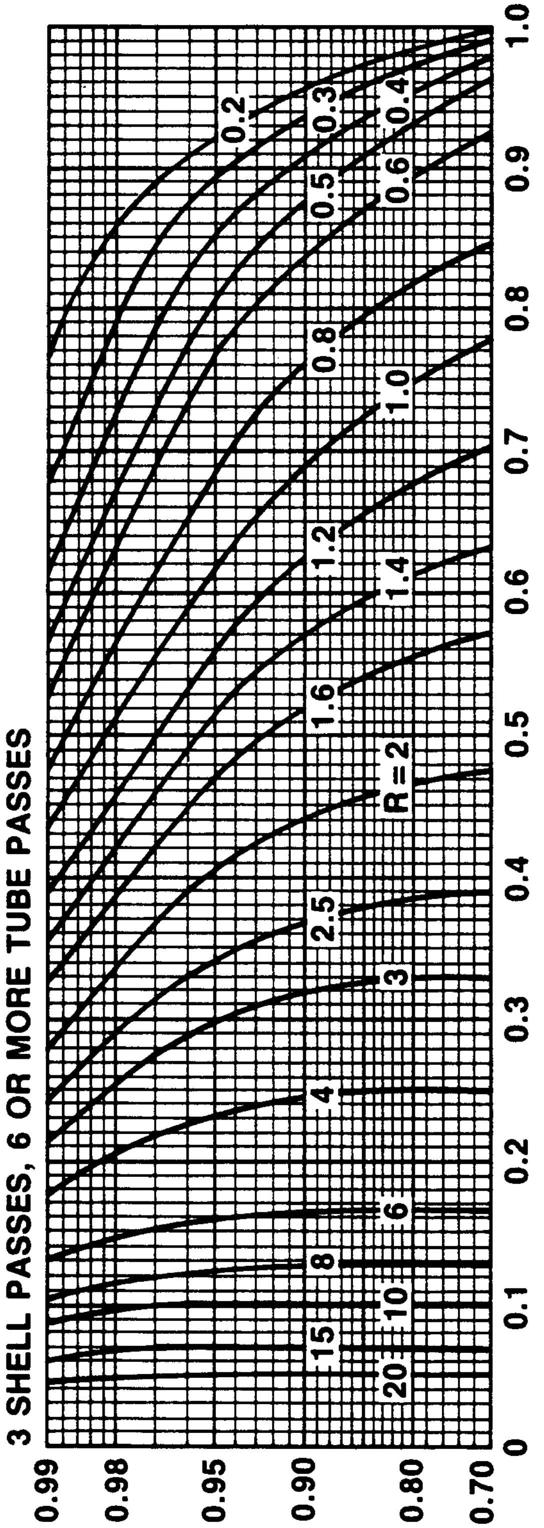


FIGURE 1.02

FIGURE 1.02

LMTD CORRECTION FACTORS



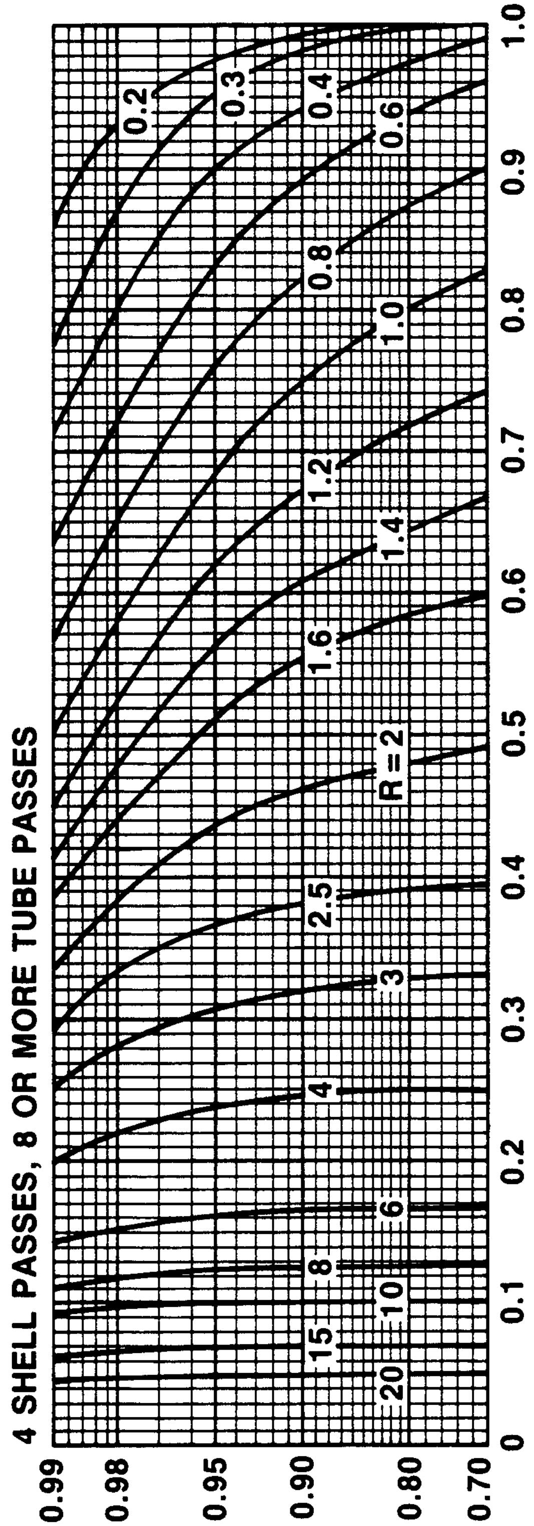


FIGURE1.03

FIGURE 1.03

LMTD CORRECTION FACTORS

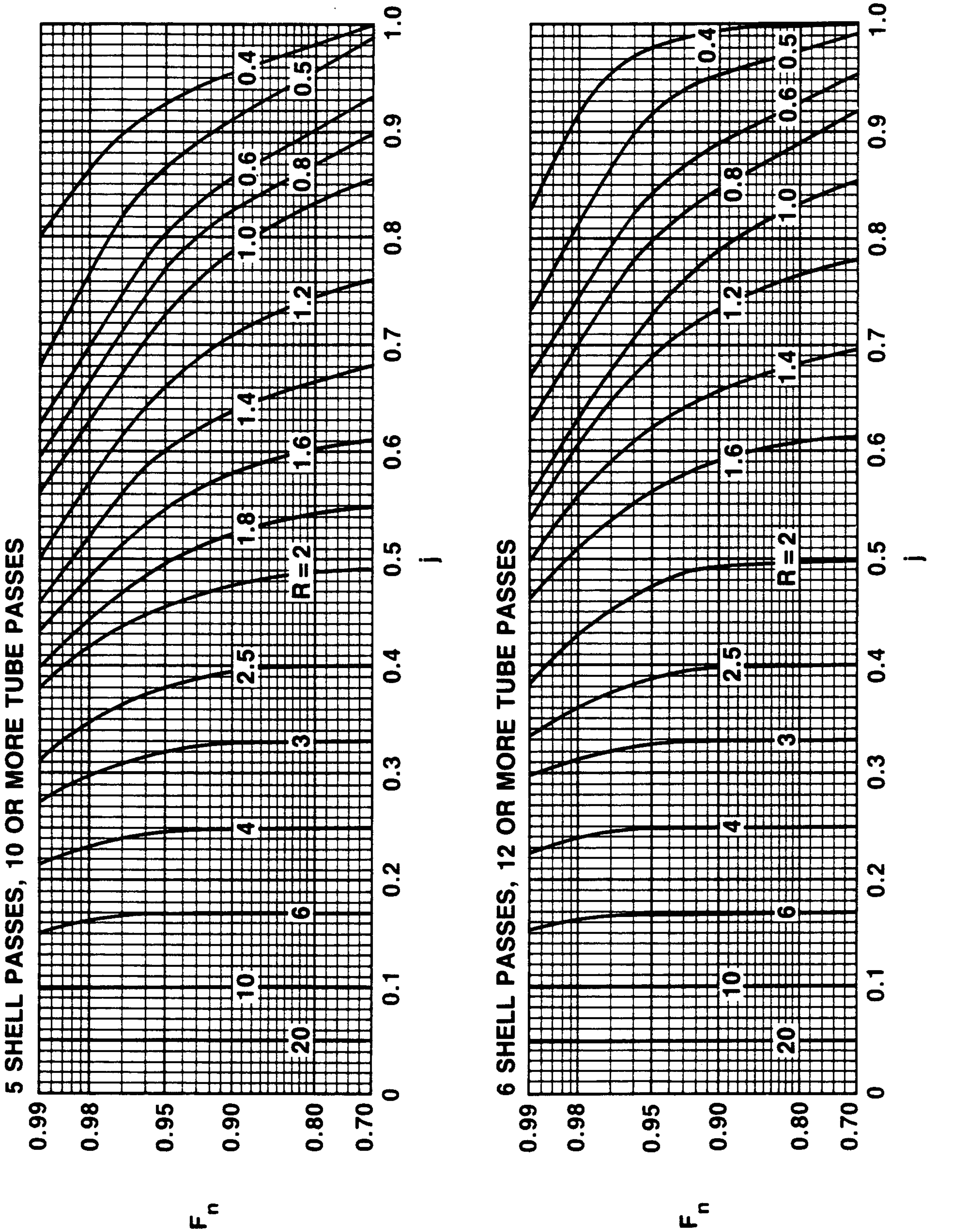


FIGURE 2.01

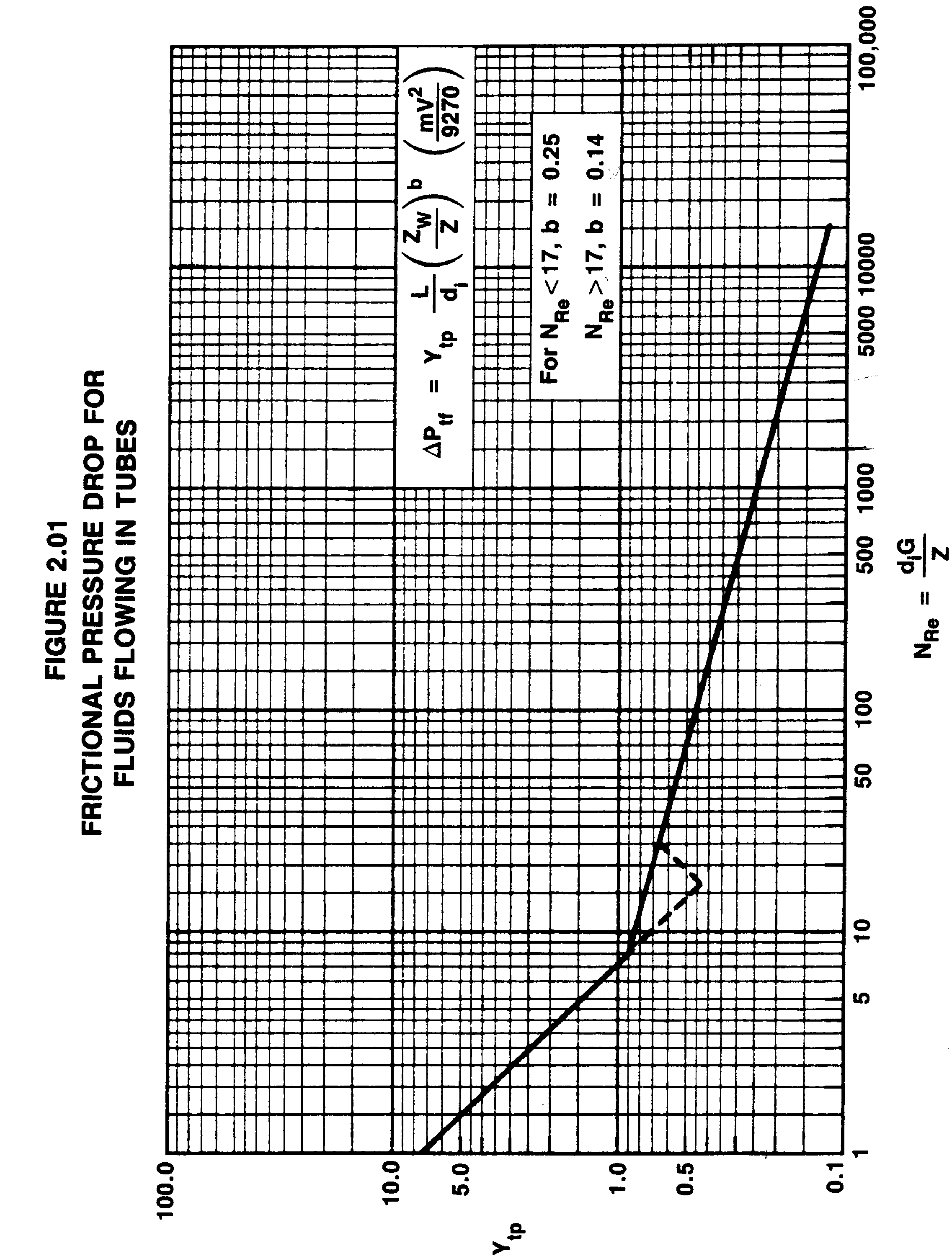


FIGURE 2.01
FRICTIONAL PRESSURE DROP FOR
FLUIDS FLOWING IN TUBES

FIGURE 2.02

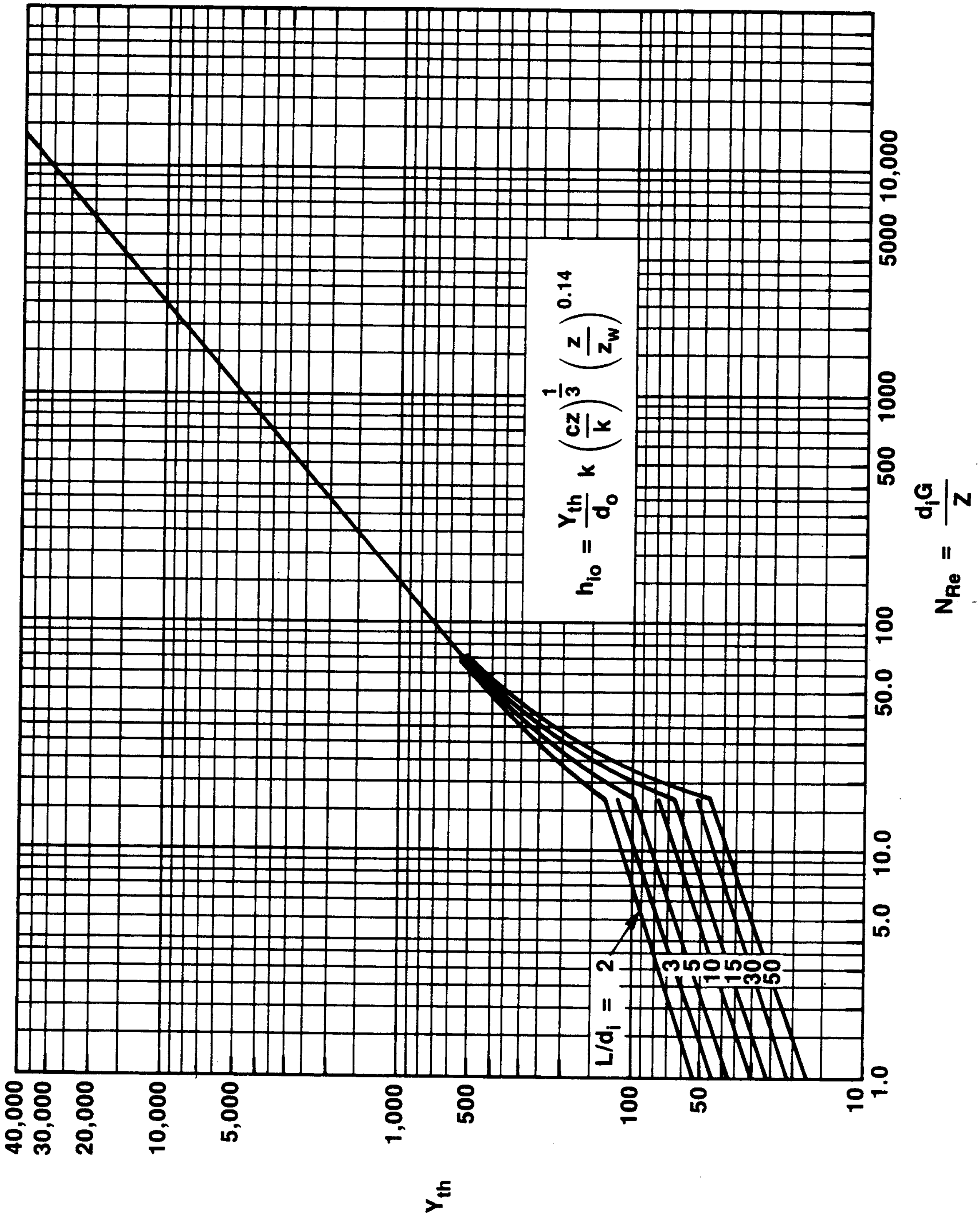


FIGURE 2.02
HEAT TRANSFER COEFFICIENT FOR FLUIDS IN TUBES

FIGURE 3.01
FRICTIONAL PRESSURE DROP
FLUIDS FLOWING ACROSS TUBE BANKS

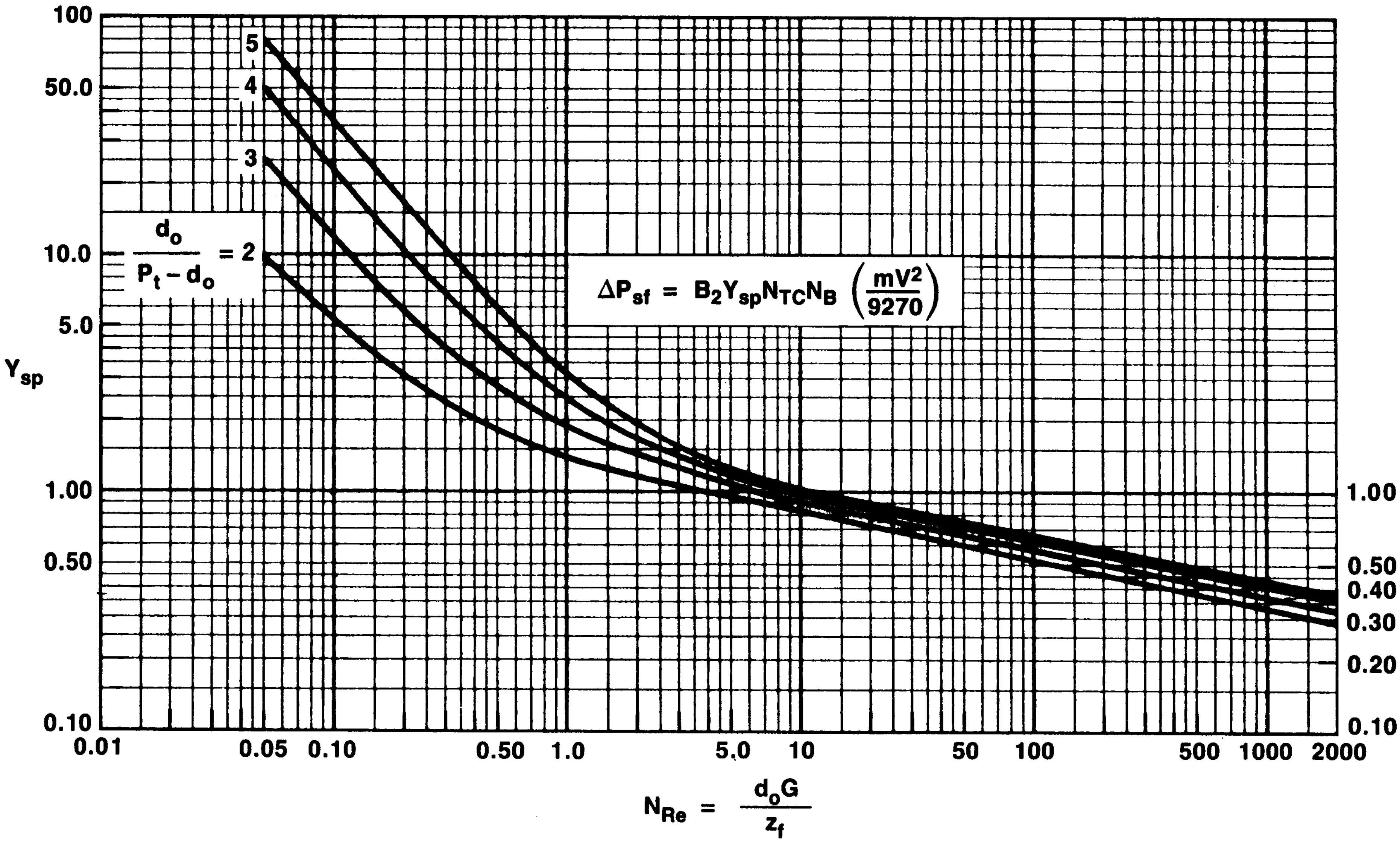


FIGURE 3.02

FIGURE 3.02
HEAT TRANSFER COEFFICIENT
FLUIDS FLOWING ACROSS TUBE BANKS

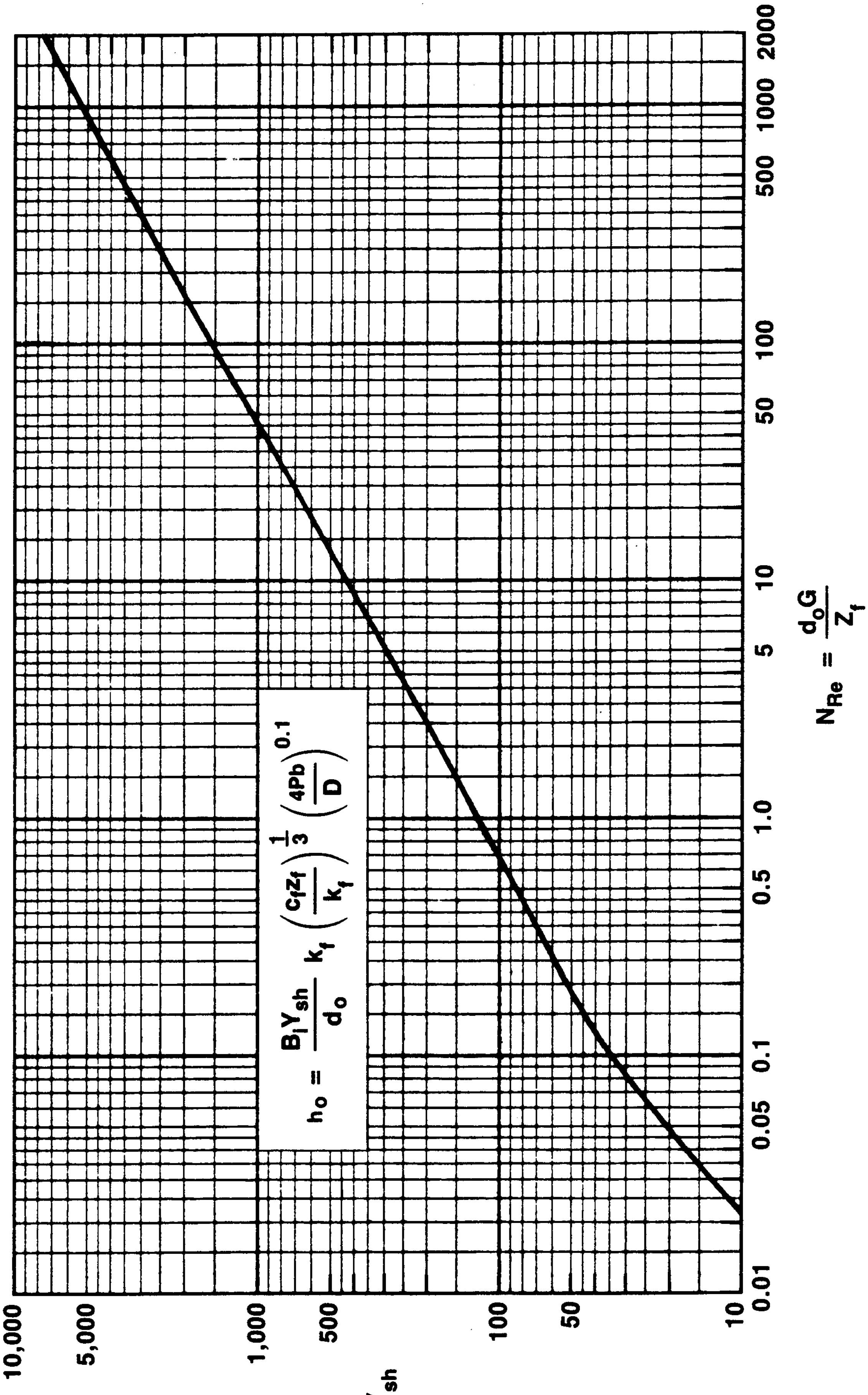


FIGURE 4.01

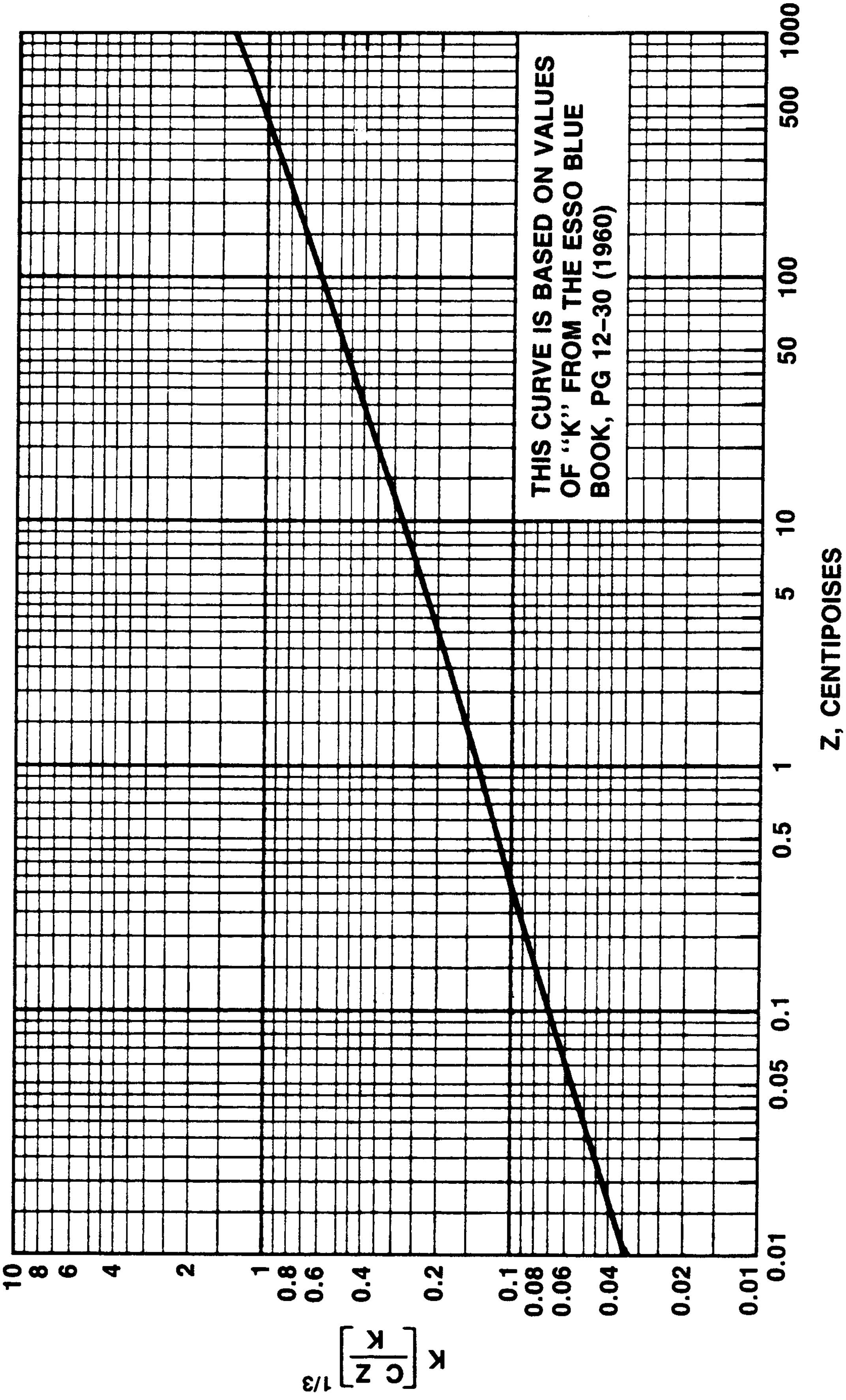
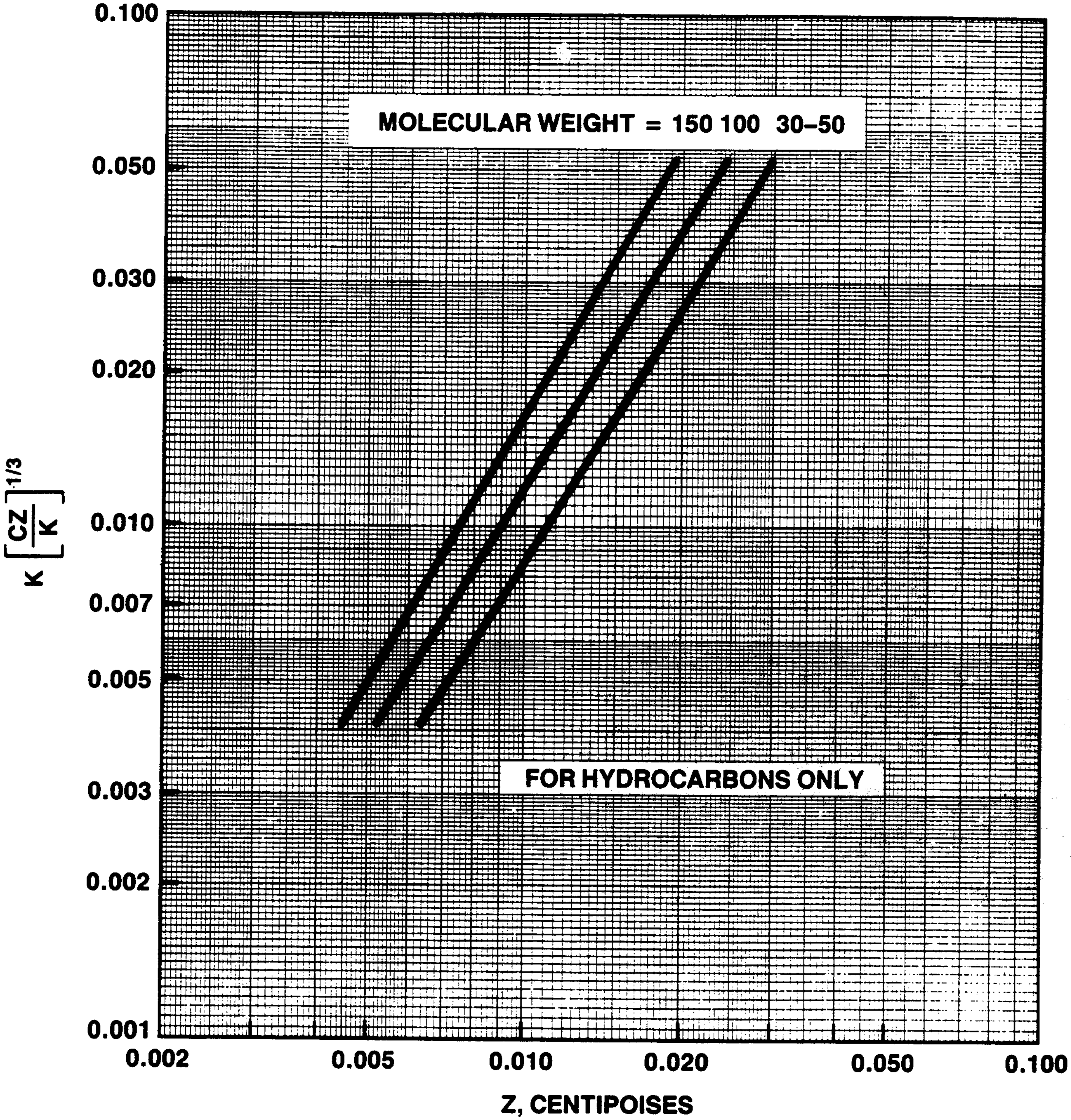


FIGURE 4.01
VALUES OF THE THERMAL FUNCTION
k(PRANDTL NO.)$^{1/3}$ FOR LIQUID HYDROCARBONS

FIGURE 4.02
VALUES OF THE THERMAL FUNCTION K (PRANDTL NO.)$^{1/3}$ FOR HYDROCARBON VAPORS



ADDENDUM 4.02

FOR FLOW INSIDE TUBES APPROXIMATE EFFECT OF VARIABLES IN THE TRANSFER OF MOMENTUM AND HEAT

Property Changed	To Find ΔP_2 Multiply ΔP_1 By:	To Find h_2 Multiply h_1 By:
$N_{Re} > 10,000$ (Note 1) Turbulent Flow		
Linear Velocity	$(V_2 / V_1)^{1.8}$	$(V_2 / V_1)^{0.8}$
Tube Diameter (at constant linear velocity)	$(D_1 / D_2)^{1.2}$	$(D_1 / D_2)^{0.2}$
Tube Diameter (at constant weight rate)	$(D_1 / D_2)^{4.8}$	$(D_1 / D_2)^{1.8}$
Viscosity	$(\mu_2 / \mu_1)^{0.2}$	$(\mu_2 / \mu_1)^{0.5}$
Density (at constant linear velocity)	$(\rho_2 / \rho_1)^{0.8}$	$(\rho_2 / \rho_1)^{0.8}$
$N_{Re} < 2,100$ (Note 1) Laminar Flow*		
Linear Velocity	V_2 / V_1	$(V_2 / V_1)^{0.33}$
Tube Diameter (at constant linear velocity)	$(D_1 / D_2)^{2}$	$(D_1 / D_2)^{0.33}$
Tube Diameter (at constant weight rate)	$(D_1 / D_2)^{4}$	D_1 / D_2
Density (at constant linear velocity)	No dependence	$(\rho_2 / \rho_1)^{0.33}$
Tube Length	L_2 / L_1	$(L_1 / L_2)^{0.33}$

Note 1: This is the dimensionless Reynolds Number.

FLUID FLOW

Determination of the Pressure Difference Between Two Points in a Flowing System Can Be Determined by Bernoulli's Theorem (Crane Page 1-5).

Simplified it Says,

Pressure Difference = Δ Elevation Head + Δ Velocity Head + Frictional Losses

For Incompressible Flow Applications, Velocity is Constant and There is No Change in Velocity Head. For Now, We Will Concentrate on Incompressible Flow and Come Back to Compressible Flow Later.

BASIC CONCEPTS

Let's Consider Frictional Losses,

Frictional Pressure Drop = No. of Velocity Heads Lost × ΔP Per Velocity Head

- Velocity Head $= \dfrac{V^2}{2g_c} = \dfrac{1}{2g_c}\left(\dfrac{G}{\rho}\right)^2$ = Feet of Flowing Fluid

  Where, V = Ft/Sec
  G = Lb/Sec-Ft^2
  ρ = Lb/Ft^3
  g_c = Dimensional Constant = $32.174 \dfrac{Lb_m}{Lb_f}\dfrac{Ft}{Sec^2}$

- In Typical Engineering Calculations, Velocity Head is Expressed as,

$$\frac{\rho}{144}\left(\frac{V^2}{2g_c}\right) = \frac{1}{144}\left(\frac{G^2}{2g_c\rho}\right) = \text{PSI} \qquad \text{(EQ. 1)}$$

BASIC CONCEPTS (Cont'd)

- Number of Velocity Heads Lost in a Line of Length L,

$$N = \frac{f_D L}{D} \quad \text{or} \quad N = \frac{4 f_f L}{D} \qquad \text{(Eq. 2)}$$

Where, f_D = Darcy Friction Factor (Dimensionless)
f_f = Fanning Friction Factor (Dimensionless)
L = Length, Ft.
D = Diameter, Ft.

- How Do We Determine Friction Factor?

BASIC CONCEPTS (Continued)

- Friction Factor Is Determined From Plots of Reynolds Number, Re, vs. Relative Roughness, ε/D, (See Crane Pages A-23 Thru 25 or D.P. Sect. XIV-B Pages 22 and 23).

$Re = \dfrac{DV\rho}{\mu}$ = Dimensionless (Crane Page 3-2 or D.P. Sect. XIV Page 6)

Where, μ = Centipoise x 0.000672 = Lb/Ft-Sec

- The Numerical Value of Darcy's Friction Factor Is Four Times That of Fanning,

$$f_D = 4 f_f$$

FRICTIONAL PRESSURE DROP IN LINES

- Assuming Fanning's f and Multiplying Eq. 1 by Eq. 2,

$$\Delta P_f = \frac{4f_f L}{D} \times \frac{\rho}{144}\left(\frac{V^2}{2g_C}\right), \text{ PSI} \qquad \text{(Eq. 3A)}$$

- This Equation Can Be Transformed Into the Useful Equation,

$$\Delta P_f = C \bullet L \frac{f}{D^5}\left(\frac{W^2}{\rho}\right), \text{ PSI} \qquad \text{(Eq. 3B)}$$

Where, W = Lb/Sec, L = Feet, D = Feet
C = 1.75×10^{-4} If Darcy's f Is Used
C = 7.0×10^{-4} If Fanning's f Is Used

This Exact Equation Can Be Approximately Represented In Charts Such as Addendum 6.04. These Approximate Charts Can Be Used for Line Size Selection. They Should Not Be Used For Rigorous Pressure Drop Calculations.

FRICTIONAL PRESSURE DROP IN LINES (Cont'd)

By Substitution, We Can Derive the Commonly Used Equations Found In D.P. Sect. XIV-B

$$\Delta P_f = 13.4\frac{f_f L W^2}{\rho d^5} = 8.63 \times 10^{-4}\frac{f_f L Q^2 \rho}{d^5} \qquad \text{(Eq. 4A)}$$

Or In Crane, Page 3-2 Eq. 3-5,

$$\Delta P_f = 3.36 \times 10^{-6}\frac{f_D L (W')^2 \bar{V}}{d^5} = 2.16 \times 10^{-4}\frac{f_D L \rho Q^2}{d^5} \qquad \text{(Eq. 4B)}$$

Where, f_f, f_D = Fanning/Darcy Friction Factors
W = kLb/H, W' = Lb/H
d = Diameter, In.
Q = Gallons/Min.
L = Feet
ρ = Density, Lb/Ft^3 = $\frac{1}{\bar{V}}$

FRICTIONAL PRESSURE DROP IN LINES

We Said We Would Come Back to Compressible Flow and This is How We Handle It,

- **For Compressible Flow Applications, Equations 3 and 4 Still Apply But We Must Use an Accurate Density Which Changes With Static Pressure.**
- **Good Rule of Thumb,**

  Eq. 3 and 4 are Accurate if $\Delta P_f \leq 10\%$ of Upstream Pressure and Average ρ is used. If Not, Break Line Length Into Smaller Segments and Calculate ΔP For Each, Starting at Point of Known Pressure.
- **Where Exxon Material is Available, See D.P. Sect. XIV-C for Detailed Compressible Flow Techniques and Nomagraphs. Otherwise, Consult a Basic Text on Fluid Flow for Similar Techniques.**
- **For Applications Within a Refinery Complex, It is Seldom Necessary to Resort to Detailed Compressible Flow Calculations. These are Only Needed in Long Transmission Lines.**

EQUIVALENT LENGTHS OF FITTINGS

For Piping Systems, We Must Include Resistances of Pipe Fittings (i.e. Valves, Elbows, Tees, etc.)

N Becomes $N + \Sigma K_{FITTINGS}$

and Total Length Becomes "Equivalent" Length ($L_{EQ.}$),

$$L_{EQ.} = L + \frac{D}{4f_f}(\Sigma K) = L + \frac{D}{f_D}(\Sigma K)$$

Where K is the "Resistance Coefficient" or Number of Velocity Heads Lost Per Fitting. See Addendum 6.05 for a Summary of "Resistance Coefficients" for Various Fittings.

EQUIVALENT LENGTHS OF FITTINGS (Cont'd)

For a 12" Pipe of 50 Ft. Linear Length With 2 Block Valves and 6 Elbows With $f_f = 0.005$,

$$L_{EQ.} = 50 + \frac{1.0}{4(0.005)}\left[(0.16)(2) + (0.4)(6)\right] = 190 \text{ Eq. Ft.}$$

(See D.P. Sect. XIV-B Pages 33-35, Crane Page A-26, or Addendum 6.05)

EQUIVALENT LENGTHS OF FITTINGS (Cont'd)

For Many Engineering Applications, Generalized Factors May Be Applied to the Linear Length to ESTIMATE Equivalent Length,

Onsite Lines $L_{EQ.} = (3.0 \text{ to } 6.0) \times \text{Linear Length}$

Offsite Lines $L_{EQ.} = (1.2 \text{ to } 1.8) \times \text{Linear Length}$

FRICTIONAL PRESSURE DROP SUMMARY

- ΔP_f = No. of Velocity Heads $\times \dfrac{\Delta P_f}{\text{Vel. head}}$
- For Engineering Applications, the Number of Veloci'y Heads Must be Adjusted to Include All System Resistances (e.g. Valves, Elbows, etc.)
- Compressible Flow Calculation Must Use Accurate Fluid Density (i.e. $\Delta P_f \leq 10\%$ of Upstream Pressure)

Many Useful Shortcut Techniques Allow Quick Calculation of Pressure Drop Such as,

- Figure 3 Charts in D.P. Sect. XIV-B Correlating GPM vs. $\Delta p/(S.G.)(100)$
- Figure 4 in D.P. Sect. XIV-B Correlating W^2/ρ vs. $\Delta p/100'$
- Figure 2 and Eq. 4 in D.P. Sect. XIV-C for Compressible Fluids
- Crane, Pages 3-22 and 3-23
- Addendum 6.04 Which Approximates $\Delta p/100'$ vs. W^2/ρ

STATIC HEAD LOSSES

Pressure Balances in Refinery Applications Must Account for Elevation Head, i.e. Static Head, Losses.

- What is Static Head?

  Static Head of a Fluid is the Weight of a Column of the Fluid. As the Elevation of a Fluid Relative to Some Datum Point (Typically Grade) Changes, the Fluid's Static Head Changes.

- In Engineering Calculations, Static Head Difference is Calculated as Follows,

$$\Delta P_{S.H.} = \frac{\rho(Z_2 - Z_1)}{144}$$

  Where,

  Z_1, Z_2 = Fluid Elevation Above Datum Point, Ft.

  For Most Engineering Pressure Balance Applications, Z_1 is Considered to be Grade.

- We Will See How We Make Use of this Term as Well as ΔP_f When We Cover Pressure Balances for Pump Applications.

TWO-PHASE FLOW

For Onsite Applications of Two Phase Flow, Frictional Pressure Drop or Static Head Can Be Satisfactorily Calculated Using the Previous Equations Presented if We Use the Average Density,

$$\rho_{2\ \mathrm{PHASE}} = \frac{(\mathrm{Lb/Hr})_V + (\mathrm{Lb/Hr})_L}{(\mathrm{Ft^3/H})_V + (\mathrm{Ft^3/H})_L}$$

Use of Average Density to Determine Two Phase Frictional Pressure Drop is Called the "No Slip" Method.

TWO-PHASE FLOW (Cont'd)

Offsite Applications of Two Phase Flow Where Acceleration Effects Can Be Significant are Beyond the Scope of this Course and are Best Referred to Fluid Flow Specialists.

FLOW REGIME

- Important to Know Flow Regime to Avoid Mechanical Problems Such as Vibration

Horizontal Pipe			Vertical Pipe (Upflow)	
Desired →	Bubble	Increasing Liquid Component ↑	Bubble	
	Plug		Slug	← Avoid
	Stratified		Froth	
	Wavy		Annular	
Avoid →	Slug	↓ Increasing vapor Component	Mist	
	Annular			
Desired →	Spray			

FLOW REGIME (Cont'd)

- Size Lines to Avoid Slug Flow. See D.P. Sect. XIV-D or Addendum 6-03 for Flowmap Coordinates Calculation.
- Specific Techniques,
  + For All Horizontal Lines Use D.P. Sect. XIV-D Fig. 1 (Page 26) or Addendum 6.03 Fig. 3
  + For Vertical Lines, d>12", Use D.P. Sect. XIV-D Fig. 2B (Page 29) or Addendum 6.03 Fig. 2
  + For Vertical Lines, d<6", Use D.P. Sect. XIV-D Fig. 2A (Page 28) or Addendum 6.03 Fig. 1
  + For Vertical Line, 6" ≤d≤12", Check Both Vertical Flow References

FLOW REGIME (Cont'd)

Important Point to Remember,

- All Flowmap Regime Predictions are for FULLY DEVELOPED Flow. It Takes 50 to 100 Pipe Diameters of Line Length for a Flow Regime to be Developed.

- Use Judgement in Employing Expensive Features to Avoid Slug Flow in Short Run Two Phase Applications

6.17

RESTRICTION ORIFICES

- Used to Absorb Some of the Excess Pressure Drop in a System

- Used to Limit Flow of Some Medium

  Purge Air or Nitrogen
  Steam to a Catalyst Bed

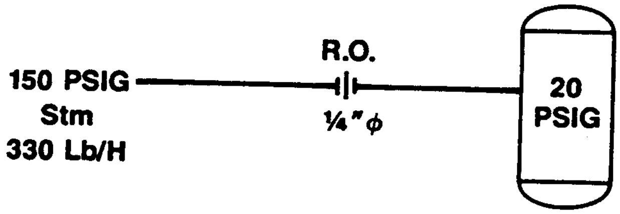
- References,

  D.P. Sect. XIV-B Page 10
  D.P. Sect. XIV-D Page 12
  Crane Pages 3-14 and 3-15

6.18

RESTRICTION ORIFICE EXAMPLES

1.



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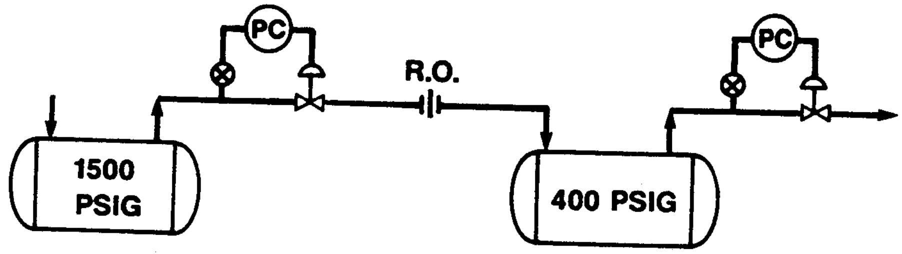


SONIC VELOCITY IN LINES

- Speed of Sound in a Fluid, V_s
- The Mass Flow at V_s is the Maximum That Can Occur
- When V_s is Attained in a Line, the Pressure at That Point Will Go No Lower Even if the Line Discharges Into a Vacuum
- The Critical Pressure Ratio is Given By

$$\frac{P_s}{P_1} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \qquad k = C_P/C_V$$

- Flow at V_s For a Given System is Dependent Only on Upstream Pressure

SONIC VELOCITY

- Sonic Velocity Can be Calculated From Fluid Properties,

$$V_s = 223\sqrt{\frac{kTz}{M}}$$

K = Specific Heat Ratio, C_P/C_V
T = Temperature, °R
Z = Compressibility
M = Molecular Weight

- See Sample Problem Addendum 6.01 at the End of this Section

and

- See Addendum 6.02 for Quick Calculation of Pressure (P_s) at V_s

PUMP SPECIFICATION

Process Design Engineer's Function—

Specify the Pumping Service

Flow Rates
Operating Conditions
Liquid Properties of Material Pumped
NPSH Availability (i.e. Suction Vessel Elevation)
Pump Type & Driver Type
Sparing Requirements
Pump Sealing and Flushing Requirements
Material Requirements (i.e. Carbon Steel, Alloy etc.)
Pump Selection

PUMP TYPES

Centrifugal—Most Common

- **High Capacity and High Head** → la altura es independ. del [illegible] el ΔP no
- **Continuous Service**
- **Often Used for Slurry Service— Need Machinery Specialist Input**

Reciprocating—

- **Low Capacity (3-20 GPM) at High Head**
- **High Viscosity Fluids**
- **Intermittent Services**
- **Additive Injection**

Rotary—

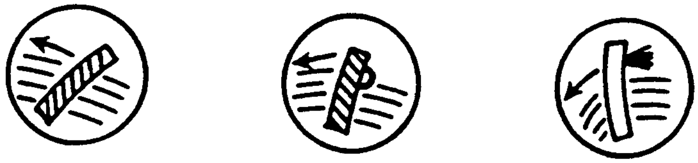
- **Limited to Services Too Viscous for Other Types**

6.23

NPSH

Cavitation—Almost Synonomous With Boiling.

- **Occurs When Static Pressure of the Liquid Falls to or Below the Vapor Pressure.**
- **Bubbles Form at this Point and as the Bubbles are Carried into Higher Pressure Zone of Pump they Implode.**

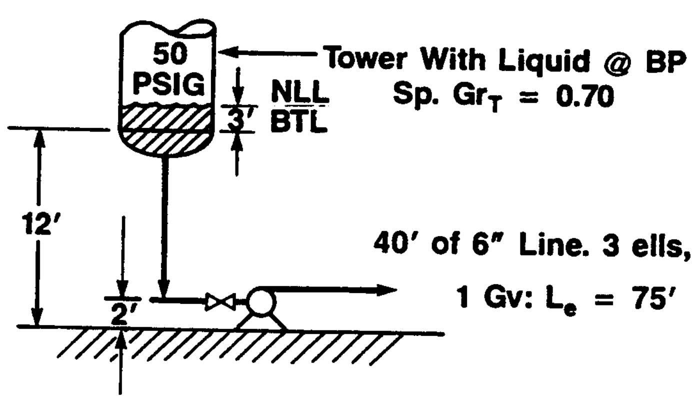


SUPPRESS CAVITATION

- The Force Suppressing Cavitation is the Margin by Which the Local Static Pressure of the Liquid Exceeds the Liquid Vapor Pressure at That Temperature.
- When Converted to Terms of Head of the Liquid, This Pressure Margin is Termed NPSH, Net Positive Suction Head.

6.25

NPSH EXAMPLE



ΔP/100′ of Suction Pipe = 0.43 PSI/100 Ft = 1.42 Ft/100 Ft

$$\Delta P_L = \frac{1.42}{100}(75) = 1.07 \text{ Ft}$$

6.26

NPSH CALCULATION

$$\text{NPSH Available} = \left[\frac{\text{Press on Liquid} + \text{Atm. Press} - \text{VP}}{\text{Sp. Gr.}_T}\right] 2.31$$

$\pm$ Level of Liquid Above Pump C.L.

$- \Delta P_{LINE}$

$$= \left(\frac{50 \text{ PSIG} + 14.7 - 64.7}{0.70}\right)(2.31)$$

$+ 12 - 2 - 1.07 = 8.93$ Ft. Available

For Safety Factor, Divide this by 1.10;

then $NPSH_A = \frac{8.93}{1.10} = 8.12'$

NPSH CALCULATION

The Previous Slide Illustrates That for Pump Applications In Which Suction Is Taken From a Vessel at Equilibrium,

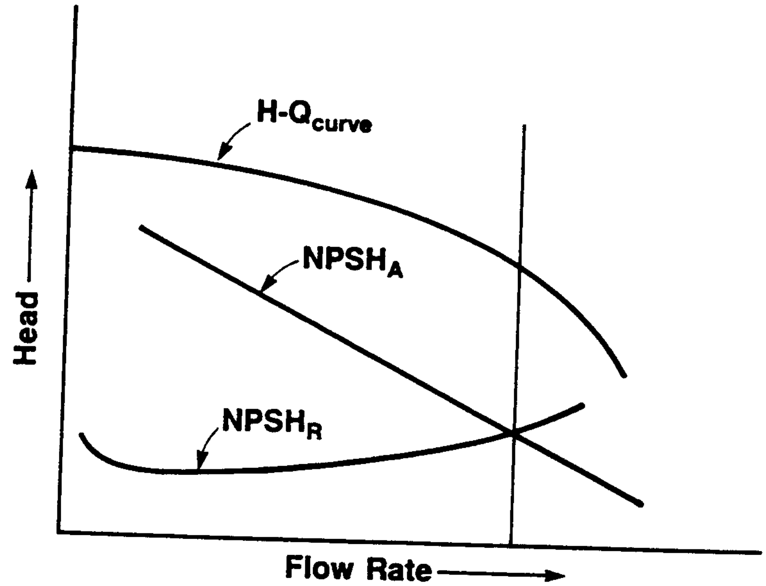
$NPSH_{AVAIL.}$ = Static Head of Liquid to Pump Centerline – Frictional Loss in the Line

This Simple Equation Allows Calculation of NPSH Available for the Vast Majority of Refinery Pump Applications. That is,

Vessel Pressure, PSIA = Vapor Pressure

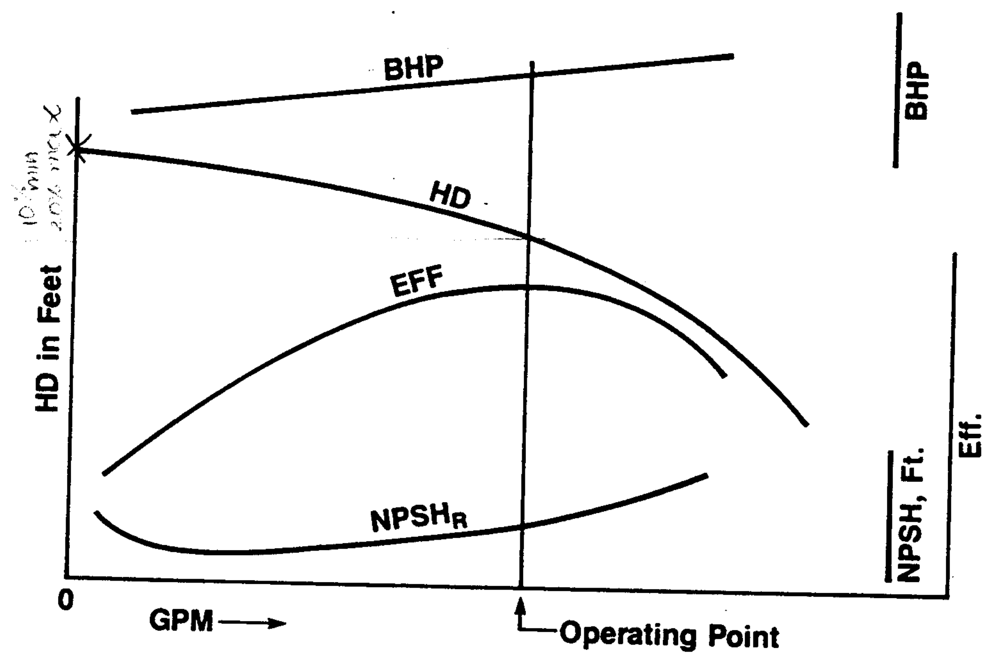
VAPOR PRESSURE OF FLUID IS THE VESSEL PRESSURE EVEN IF VESSEL ATMOSPHERE IS MAINLY STEAM OR INERTS. (This Does Not Apply to Atmospheric Pressure Tank Blanketing.) Steam and/or Inerts Under Pressure are Soluble in Hydrocarbon Liquids. NPSH Available at Pump Suction Must be Adequate to Ensure this Material Stays in Solution.

NPSH VS. FLOW RATE



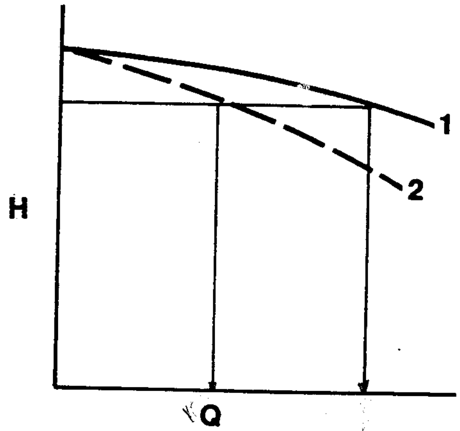
6.29

CENTRIFUGAL PUMP CHARACTERISTICS

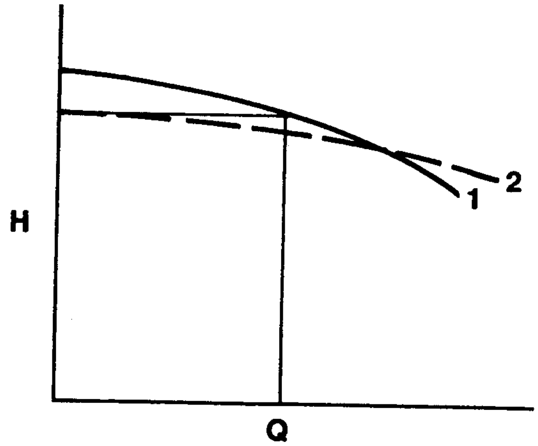


6.30

PARALLEL PERFORMANCE



Cent. Pumps
Properly Matched



Cent. Pumps
Poorly Matched

6.31

HORSEPOWER CALCULATION

- Hydraulic Horsepower, HP $= \dfrac{(\Delta P_{pump})\ (GPM)}{1715}$
- Brake Horsepower, BHP $= \dfrac{\text{Hydraulic Horsepower}}{E_o}$

  Where,

  E_o = Pump Efficiency
- E_o May Be Obtained From Fig. 3 or 4 D.P. Sect. X-A or From Addendum 6.06
- Minimum Driver Horsepower = BHP × Load Factor

  Where,

  Load Factor = 1.1 for Electric Motors

6.32

HORSEPOWER CALCULATION (Cont'd)

- Power Consumed $= \dfrac{BHP\ (0.746)}{E_M} = \dfrac{\text{kw-hr}}{\text{hr}}$

  Where E_M = Motor Efficiency
- E_M May Be Obtained From D.P. Sect. XI-L Table 1 or Addendum 6.0
- Example: $\Delta P_{PUMP} = P_2 - P_1 = 150$ PSI 1000 GPM $S.G._C = 0.65$

  + BHP $= \dfrac{(150)(1000)}{(1715)\ E_o}$

  + See Handout 6-6: E_o = 71% for Head = 533 Ft. $= \dfrac{(2.31)(150\ \text{PSI})}{0.65}$

  + BHP = 123.2

  + Min. Driver HP = 123.2 x 1.1 = 135.5 Hp → Select 150 Hp Motor

  + Op. Load $= \dfrac{(123.2)(0.746)}{0.89}$ Where 0.89 is 150 Hp Motor Efficiency @ Approx. 75% Load

  + Op. Load = 103.3 kw

CHANGING IMPELLER DIAMETER OR SPEED

Q ~ Peripheral Speed

H ~ (Peripheral Speed)^2

HP ~ (Peripheral Speed)^3

These Relationships are Called the "Pump Affinity Laws"

IMPELLER DIAMETER CHANGE EXAMPLE

10″			11″		
Rate	Hd	HP	Rate	Hd	HP
100	335	200	110	405	266
200	335	220	220	405	293
600	325	290	660	393	386
1000	295	328	1100	357	437
1400	245	412	1540	296	548

$$\text{Rate (11)} = (100)\left(\frac{11}{10}\right) = 110$$

$$\text{Hd (11)} = (335)\left(\frac{11}{10}\right)^2 = 405$$

$$\text{HP (11)} = (200)\left(\frac{11}{10}\right)^3 = 266$$

6.35

COMPRESSOR TYPES

A. Dynamic

- Centrifugal
- Axial

B. Positive Displacement

- Reciprocating
- Rotary (Screw)

6.36

TYPICAL TYPE/USAGE

Centrigugal—First Choice, General Process

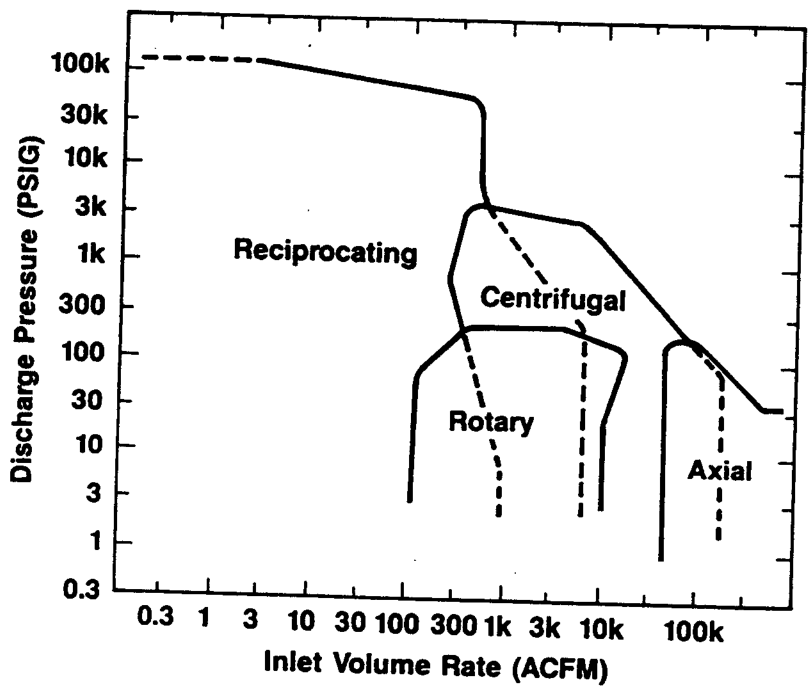
Axial—For Very High Suction Volume, Typically > 50,000 ACFM

Reciprocating—Small Inlet Suction Volume (<500 ACFM) and/or Very High Discharge Pressure

Rotary—Small Inlet Suction Volume and Moderate Discharge Pressure

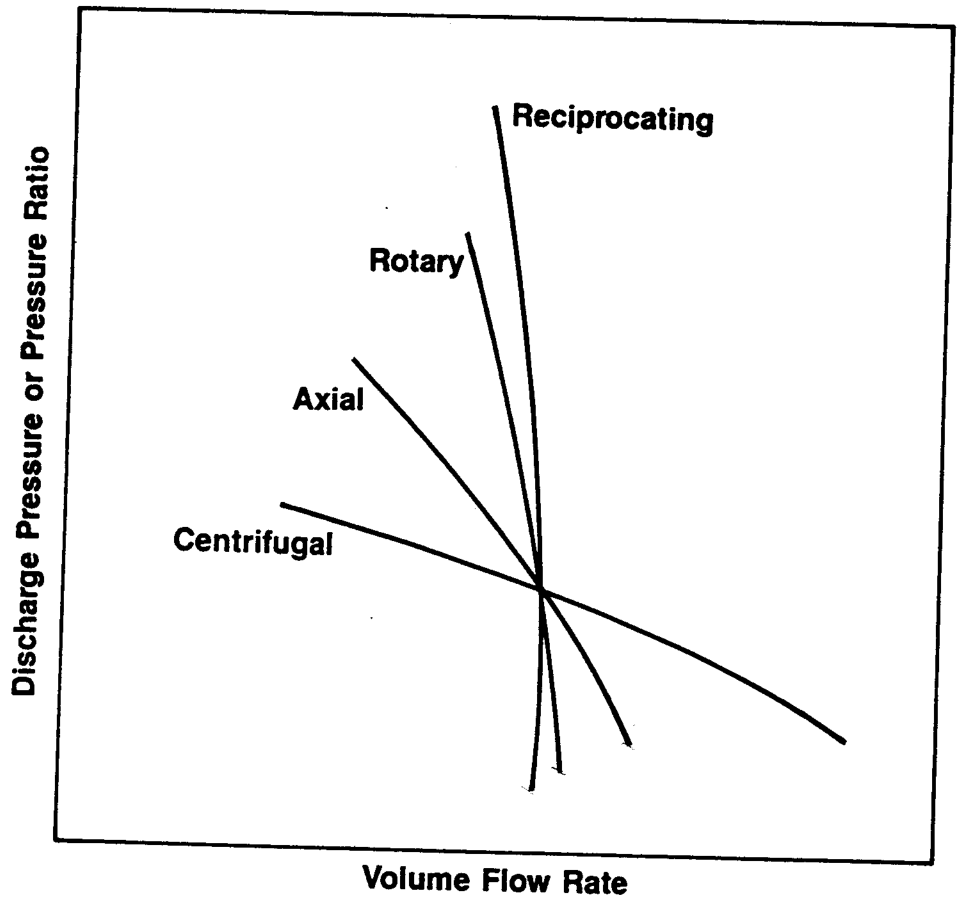
6.37

COMPRESSOR APPLICATION RANGES



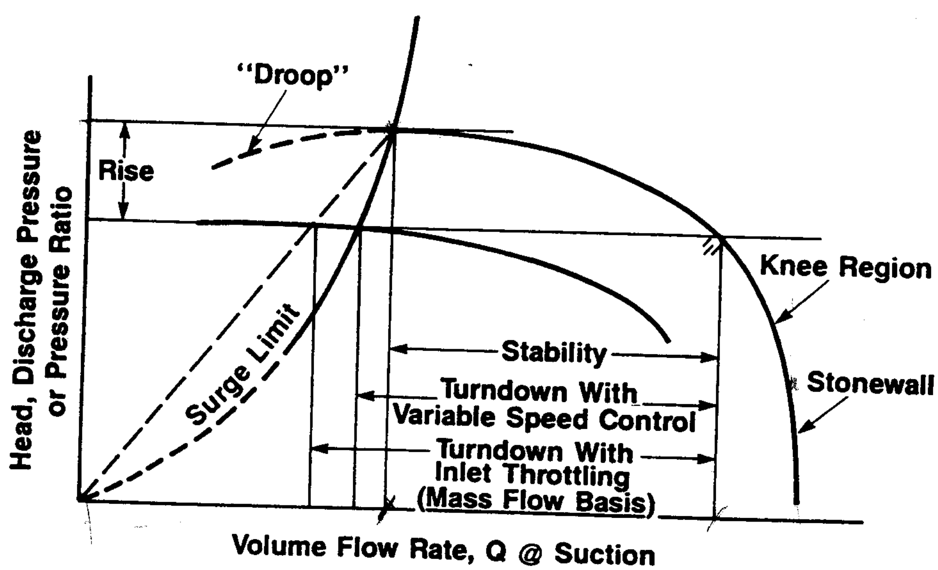
6.38

CHARACTERISTIC CURVES FOR MAJOR COMPRESSOR TYPES



6.39

CENTRIFUGAL COMPRESSOR PERFORMANCE CURVE TERMINOLOGY



6.40

CENTRIFUGAL COMPRESSOR PERFORMANCE CURVE TERMINOLOGY (Cont'd)

Surge Limit — that Point on the Curve Where Compressor Suddenly Loses Its Ability to Develop the Pressure Existing in the Discharge Piping. Gas Spills Backwards Through the Machine Lowering the Pressure Differential and Allowing the Machine to Deliver Positive Flow Forward. The Pressure Differential Again Rises and Flowrate Falls Until the Surge Inception Occurs Again and the Cycle Repeats Itself.

Stonewall —that Point Defining the Maximum Flowrate Which the Compressor Can Pass. The Maximum Flow Limitation is Created by the Gross Turbulence, Shock Waves and Flow Separation Which Occur as Gas Velocity (Relative to the Impeller Surface) Approaches Sonic Velocity (Mach 1).

6.41

CENTRIFUGAL COMPRESSOR PERFORMANCE CURVE TERMINOLOGY (Cont'd)

Stability — the Stable Flow Range Between Surge and Design Flow. This is Expressed as a Percentage and Equals 100% Minus the Percent Ratio of Volume Flowrate at Surge to Design Volume Flowrate. API 617 Defines "Stability" in Terms of "Rated" Rather than Design Flow.

Knee Region — Portion of the Curve in Which the Slope is Changing Rapidly, Immediately Before "Stonewall" is Reached. To Prevent Design Point in the "Knee Region", BP10-3-1 Requies Head Capability at 115% of Design Flow to be Not Less Than "Approximately" 85% of the Head Developed at Design Flow.

6.42

CENTRIFUGAL COMPRESSOR PERFORMANCE CURVE TERMINOLOGY (Cont'd)

Turndown — Is the Mass Flow Reduction Which is Possible Before Encountering Surge, Recognizing the Effects of the Control Method Employed. "Turndown" is Expressed as a Percentage and Equals 100% Minus the Percent Ratio of the Surge Point Mass Flow at Design Head to the Design Mass Flow. API 617 Defines "Turndown" in Terms of "Rated" Rather than Design Flow.

Rise — to Insure Stable Operation Within the Required Flow Range, the Head-Capacity Curve of the Compressor Must Rise Slightly and Continously Between the Design Flowrate and the Flowrate Where Surge Begins. Design Specifications Typically Specify a Minimum Curve Rise that a Well Designed Compressor Can Reasonably be Expected to Provide. (See D.P. Sect. XI-E Fig. 5 or Addendum 6.08).

6.43

FAN LAWS CENTRIFUGAL COMPRESSOR AFFINITY LAWS

These Laws Describe the Relationships Among:

- Compressor Speed (N = RPM)
- Volumetric Throughput (Q = ACFM)
- Discharge Head (H = Feet)
- Horsepower Requirement (HP = Gas Horsepower)

As Speed Goes From N_1 to N_2:

$$\frac{Q_2}{Q_1} = \frac{N_2}{N_1} \qquad \frac{H_2}{H_1} = \left(\frac{N_2}{N_1}\right)^2$$ (While Q Rises Linearly)

$$\frac{HP_2}{HP_1} = \left(\frac{N_2}{N_1}\right)^3$$ (If Head and Flow are Allowed to Rise)

6.44

HORSEPOWER CALCULATIONS

Two Methods:

- **Isentropic**

  PV^k = Constant

  $k = C_p/C_v$

  Can Be Applied to Reciprocating Compressors.

- **Polytropic**

  PV^n = Constant

  $n \neq k$

  n = Polytropic Compression Exponent
  Basic Method for Centrifugal Compressors

CENTRIFUGAL COMPRESSOR CALCS
POLYTROPIC METHOD

1. **Calculate Temperature Rise**

   $T_2 = T_1 (P_2/P_1)^m$

   T in °R; P in PSIA

   m = Temperature Rise Exponent

   - When $Z \sim 1.0$, $m = \frac{n-1}{n} = \frac{k-1}{k(\text{Poly. Eff.})}$

     $k = C_P/C_V = C_P/(C_P - R) = C_P/(C_P - 1.9872)$

   Polytropic Eff. ~ $0.0109 \cdot \ell n\,(\text{ACFM}) + 0.643$

   - When $Z \neq 1.0$ or Precise Calculations are Necessary – Use a Computer

CENTRIFUGAL COMPRESSOR CALCS
POLYTROPIC METHOD (Cont'd)

2. Calculate Polytropic Head: $$H_p = \left(\frac{ZRT_1}{MW}\right)\left(\frac{1}{\frac{n-1}{n}}\right)\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

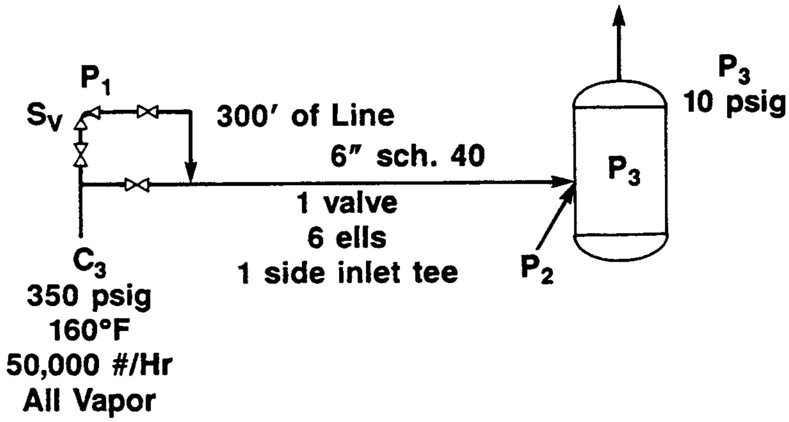
H_p = Polytropic Head
Z_1 = Compressibility @ Suction
R = Gas Constant = 1545 (Ft^3 x PSF)/(Lb Mol x °R)
T_1 = Inlet Temperature °R
N = Compression Exponent
P_1 = Suction Pressure, PSIA
P_2 = Discharge Pressure, PSIA
MW = Molecular Weight

3. Gas Horsepower = $$\frac{(Lb / Min)}{33000}\left(\frac{H_p}{Poly\ \ Eff}\right)$$

4. BHP = GHP + Mechanical Losses

Addendum 6.01

CALCULATION OF SONIC VELOCITY AND PRESSURE DROP



Problem: 1. Determine if V_S is Reached at End of Line

2. Calculate Pressure at Safety Valve Outlet Using Any Approximate Method.

Solution: When the C_3 is Expanded From 350 PSIG and 160°F to 10 psig, t_2 = 110°F (Exxon Bluebook p. 7-91 or Maxwell p. 133)

We Can Determine if Sonic Velocity is Reached by Calculating the Minimum Pressure that Will Produce Sonic Velocity. If this Pressure is Less Than the Drum Pressure, Sonic Velocity is Not Reached.

$$P_s = 0.048\ G \sqrt{\frac{ZT}{kM}}$$ (See Addendum 6.02 for Derivation)

C_3: t_c = 206.3°F $\quad p_c$ = 42 ATM (Exxon Bluebook p. 1-18 or Maxwell p. 2)

$$T_r = \frac{570}{666.3} = 0.86 \quad p_r = \frac{1.68}{42} = 0.04$$

$\therefore Z = 0.97$ (Exxon Bluebook p. 8-40 or Maxwell p.148)

$$k = {}^{C_p}/_{C_v} = {}^{C_p}/_{C_p - R}$$

C_p = 0.39 BTU/lb−°F (Exxon Bluebook p. 7-23 or Maxwell p. 89)

$$k = \frac{(0.39)\ (44.1)}{(0.39)(44.1)-1.9872}$$

$$k = 1.13$$

$$P_s = 0.048\ (69.2\ \text{lb/s-Ft}^2)\ \sqrt{\frac{(0.97)(570)}{(1.13)(44.1)}}$$

$$= 11.06\ \text{PSIA}$$

Since the Drum Runs at 24.7 PSIA, Sonic Velocity is Not Reached.

2. Use the Method in Crane on page 3-22/23

$$\Delta P/_{100} = \frac{C_1\ C_2}{\rho}$$

From Figure on p. 3-22, C_1 = 2.5

From Table on p. 3-23, C_2 = 0.61

Assume ΔP = 27 PSI

$$P_{AVG} = 24.7 + \frac{27}{2} = 38.2\ \text{PSIA}$$

$$\rho_{AVG} = \frac{pM}{ZRT} = \frac{(38.2)(44.1)}{(0.97)(10.73)(570)} = 0.283\ \text{lb/Ft}^3$$

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$$\Delta P/_{100} = \frac{(2.5)(0.61)}{0.283} = 5.39/100 \text{ Ft.}$$

Determine Equivalent Length,

$$L_{eq} = L + \frac{D}{f_D}(\Sigma K)$$

Fully Developed Turbulent Flow is a Good Assumption

$f_D = 0.015$ for 6″ Line

Refer to Addendum 6.05 for Resistances of Valves/Fittings,

$$L = 300 + \frac{0.5}{0.015}(0.16 + 6(0.4) + 1.2) = 425 \text{ Eq. Ft.}$$

$$\Delta P = 5.39 \frac{\text{PSI}}{100'}(425) = 22.9 \text{ PSI}$$

This is Close Enough to Assumed ΔP

$$\boxed{\therefore P_1 = 10 \text{ PSIG} + 22.9 = 32.9 \text{ PSIG}}$$

Since ΔP is Greater Than 10% of Upstream Pressure, We May Consider Dividing the Live Segment Into Smaller Segments to Improve Calculation Accuracy. For This Calculation that Degree of Accuracy is Unnecessary.

Addendum 6.02

CALCULATION OF MINIMUM PRESSURE THAT WILL GIVE SONIC VELOCITY

$$V_s = 223\sqrt{\frac{kTz}{M}}$$

ρ = lb/Ft^3
k = Cp/Cv
T = °R
Z = Compressibility
M = Mol. Wt.
G = lb/s-Ft^2

If We Have Sonic Velocity,

$$V_{ACTUAL} = V_S$$

$$223\sqrt{\frac{kTz}{M}} = \frac{G}{\rho}$$

$$223\sqrt{\frac{kTz}{M}} = \frac{G}{\frac{PM}{Z(10.73)T}}$$

$$(223)^2\left(\frac{kTz}{M}\right) = \frac{G^2}{\frac{P^2M^2}{Z^2(10.73)^2T^2}}$$

$$\frac{(223)^2}{(10.73)^2}\,\frac{kM}{TZ} = \frac{G_2}{P^2}$$

$$P^2 = \frac{(10.73)^2}{(223)^2}\,\frac{TZ\,G^2}{kM}$$

$$\boxed{P = 0.048\,G\sqrt{\frac{TZ}{kM}}\text{, PSIA}}$$

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Addendum 6.03
TWO-PHASE FLOW REGIMES

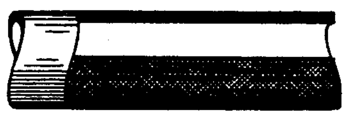
Flow Regimes in Horizontal or Slightly Inclined Pipe

In two-phase flow, interactions between liquid and vapor phases, as influenced by their physical properties and flow rates and by the size, roughness and orientation of the pipe, cause the fluids to flow in various types of patterns. These patterns are called *flow regimes*. Only one type of flow exists at a given point in a line at any given time. However, as flow conditions change, the flow regime may change from one type to another.

Seven principal flow regimes have been defined to describe flow found in horizontal or slightly inclined pipes. These flow regimes are described below, *in order of increasing vapor velocity*. In the accompanying sketches, the direction of flow is from left to right.

Bubble Flow - Liquid occupies the bulk of the cross-section and vapor flows in the form of bubbles along the top of the pipe. Vapor and liquid velocities are approximately equal. If the bubbles become dispersed throughout the liquid, then this is sometimes called froth flow. In uphill flow bubbles retain their identity over a wider range of conditions. In downhill flow the behavior is displaced in the direction of plug flow.

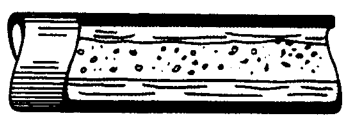
Plug Flow - As the vapor rate increases, the bubbles coalesce, and alternating plugs of vapor and liquid flow along the top of the pipe with liquid remaining the continuous phase along the bottom. In an uphill orientation, the behavior is displaced in the direction of bubble flow; downhill, stratified flow is favored.



Stratified Flow - As the vapor rate continues to increase, the plugs become a continuous phase. Vapor flows along the top of the pipe and liquid flows along the bottom. The interface between phases is relatively smooth and the fraction occupied by each phase remains constant. In uphill flow, stratified flow rarely occurs with wavy flow being favored. Downhill, stratified flow is somewhat enhanced, as long as the inclination is not too steep.

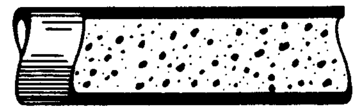
Wavy Flow - As the vapor rate increases still further, the vapor moves appreciably faster than the liquid, and the resulting friction at the interface forms liquid waves. The wave amplitude increases with increasing vapor rate. Wavy flow can occur uphill, but over a narrower range of conditions than in horizontal pipe. Downhill, the waves are milder for a given vapor rate and the transition to slug flow, if it occurs at all, takes place at higher vapor rates than in horizontal pipe.

Slug Flow - When the vapor rate reaches a certain critical value, the crests of the liquid waves touch the top of the pipe and form frothy slugs. The velocity of these slugs, and that of the alternating vapor slugs, is greater than the average liquid velocity. In the body of a vapor slug the liquid level is depressed so that vapor occupies a large part of the flow area at that point. Uphill, slug flow is initiated at lower vapor rates than in horizontal pipe. Downhill, it takes higher vapor rates to establish slug flow than in horizontal pipe, and the behavior is displaced in the direction of annular flow. Since slug flow may lead to pulsation and vibration in bends, valves and other flow restrictions, it should be avoided where possible.



Annular Flow - The liquid flows as an annular film of varying thickness along the wall, while the vapor flows as a high-speed core down the middle. There is a great deal of slip between phases. Part of the liquid is sheared off from the film by the vapor and is carried along in the core as entrained droplets. The annular film on the wall is thicker at the bottom of the pipe than at the top, the difference decreasing with distance from slug flow conditions. Downstream of bends, most of the liquid will be at the outer wall.

In annular flow, the effects of friction pressure drop and momentum outweigh the effect of gravity, so that pipe orientation and direction of flow have less influence than in the previous flow regimes. Annular flow is a very stable flow regime. For this reason and because vapor-liquid mass transfer is favored, this flow regime is advantageous for some chemical reactions.



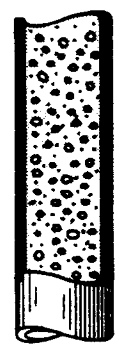
Spray Flow (Also known as mist flow or dispersed flow) - When the vapor velocity in annular flow becomes high enough, all of the liquid film is torn away from the wall and is carried by the vapor as entrained droplets. This flow regime is almost completely independent of pipe orientation or direction of flow.

Flow Regimes in Vertical Pipe

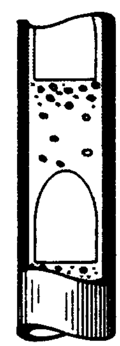
Flow behavior in vertical pipes, where gravity plays an important role, has been less extensively investigated than has flow in horizontal pipes. Most of the available information on vertical flow pertains to upflow.

Conditions under which certain flow regimes exist depend largely on the orientation of the pipe and the direction of flow. In a situation where stratified or wavy flow would exist in a horizontal pipe, tilting the pipe downward increases the relative velocity of the liquid, making a larger part of the flow area available for the vapor. On the other hand, tilting the pipe upward causes the liquid to drain back downhill until enough has accumulated to block off the entire cross-section. The vapor can then no longer get past the liquid, and therefore pushes a slug of liquid through the inclined section of the line.

Five principal flow regimes have been defined to describe vertical flow. These flow regimes are described below, *in order of increasing vapor velocity*. In the accompanying sketches, the direction of flow is upward.



Bubble Flow - Upward flowing liquid is the continuous phase, with dispersed bubbles of vapor rising through it. The velocity of the bubbles exceeds that of the liquid, because of buoyancy. As vapor flow rate is increased, the sizes, number and velocity of the bubbles increase. The bubbles retain their identity, without coalescing into slugs, at higher vapor rates than in horizontal pipe.

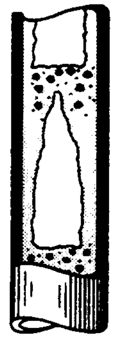


Slug Flow - As the vapor rate increases, bubbles coalesce into slugs which occupy the bulk of the cross-sectional area. Alternating slugs of vapor and liquid move up the pipe with some bubbles of vapor entrained in the liquid slugs. Surrounding each vapor slug is a laminar film of liquid which flows toward the bottom of the slug. As the vapor rate is increased, the lengths and velocity of the vapor slugs increase.

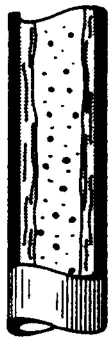
Slug flow can occur in the downward direction, but is usually not initiated in that position. However, if slug flow is well established in an upward leg of a coil, it will persist in a following downward leg, provided that other conditions remain the same.

In designing for two-phase flow it is normal practice to try to avoid slug flow, since this regime can lend to serious pressure fluctuations and vibration, especially at vessel inlets and in bends, valves and other flow restrictions. This could lead to serious equipment deterioration or operating prblems. When slug flow cannot be avoided (for instance, in thermosyphon reboilers), one should avoid flow restrictions and use long-radius bends to make turns as smooth as possible.

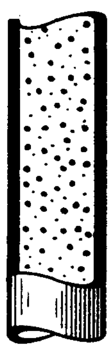
Froth Flow - As the vapor rate increases further, the laminar liquid film is destroyed by vapor turbulence and the vapor slugs become more irregular. Mixing of vapor bubbles with the liquid increases and a turbulent, disordered pattern is formed with ever shortening liquid slugs separating successive vapor slugs. The transition to annular flow is the point at which liquid separation between vapor slugs disappears and the vapor slugs coalesce into a continuous, central core of vapor. Since froth flow has much in common with slug flow, the two regimes are often lumped together and called slug flow. In the downward direction, froth flow behaves much the same as slug flow does, except that the former is more easily initiated in this position, particularly if conditions are bordering on those for annular flow.



Annular Flow - This flow regime is similar to annular flow in horizontal pipe, except that the slip between phases is affected by gravity. In upflow, the annular liquid film is slowed down by gravity, which increases the difference in velocities between vapor and liquid. In downflow, the reverse is true, with gravity speeding up the liquid and reducing the difference in velocities between vapor and liquid. On the other hand, the liquid film thickness is more uniform around the circumference of the pipe than in horizontal flow.



Mist Flow - This flow regime is essentially the same as spray flow in horizontal pipe. The very high vapor rates required to completely disperse the liquid essentially eliminate the effects of orientation and direction of flow. In identification of vertical two-phase flow regimes, annular and mist flow are often considered together (and called annular-mist).



Effect of Fittings on Flow Regimes

Fittings may strongly affect the flow of vapor-liquid mixtures. Bends will tend to separate the flow, causing the liquid to follow the outer wall, while valves and other flow restrictions will disperse the two phases in each other. Downstream of a fitting, it may take more than 100 pipe diameters before the flow has reached an equilibrium pattern again. Separation in bends can be minimized by use of blanked-off tees instead of elbows. The fluid should enter through one straight-through end and exit through the branch.

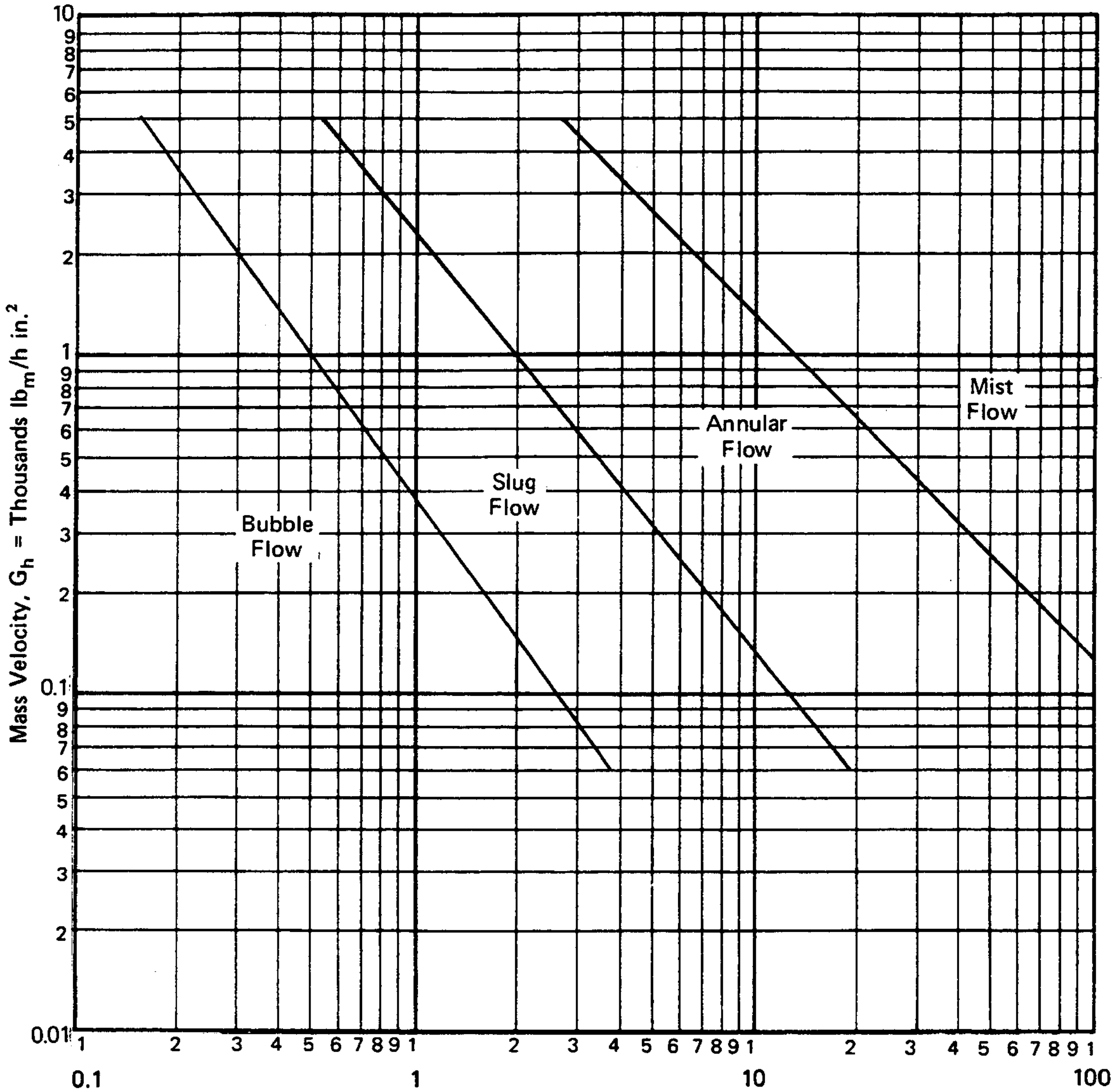
Distribution of two-phase flow to parallel equipment should be done in a symmetrical fashion. For example, uniform distribution over four heat exchangers requires that the flow first be divided symmetrically into two substreams, and then each substream again be split into two streams. Elbows immediately upstream of a distributing tee should be mounted perpendicular to the plane of the tee. If this is not possible, a blanked-off tee should be used. In cases where gravity would seriously affect distribution, parallel equipment should be kept at the same level.

Figure 1
TWO-PHASE FLOW REGIMES IN VERTICAL PIPE

NOMENCLATURE

Y = Weight Fraction Gas or Vapor in Mixture
ρ_L = Liquid Density, lb/ft^3
ρ_G = Gas or Vapor Density, lb/ft^3
μ_L = Liquid Viscosity, cP
μ_G = Gas or Vapor Viscosity, cP

(Line Diameter < 6″)



$$Z = \left(\frac{y}{1-y}\right)^{0.9}\left(\frac{\rho_L}{\rho_G}\right)^{0.5}\left(\frac{\mu_G}{\mu_L}\right)^{0.1}$$

NOMENCLATURE

ρ_L = Liquid Density, lb/ft^3
ρ_V = Vapor Density, lb/ft^3
d = Inside Pipe Diameter, in.
Y = Weight Fraction Gas or Vapor in Mixture
W = Total Mass Flowrate, klb/hr

Figure 2
FLOW REGIMES IN VERTICAL PIPES, UPFLOW
(Line Diameter > 12")

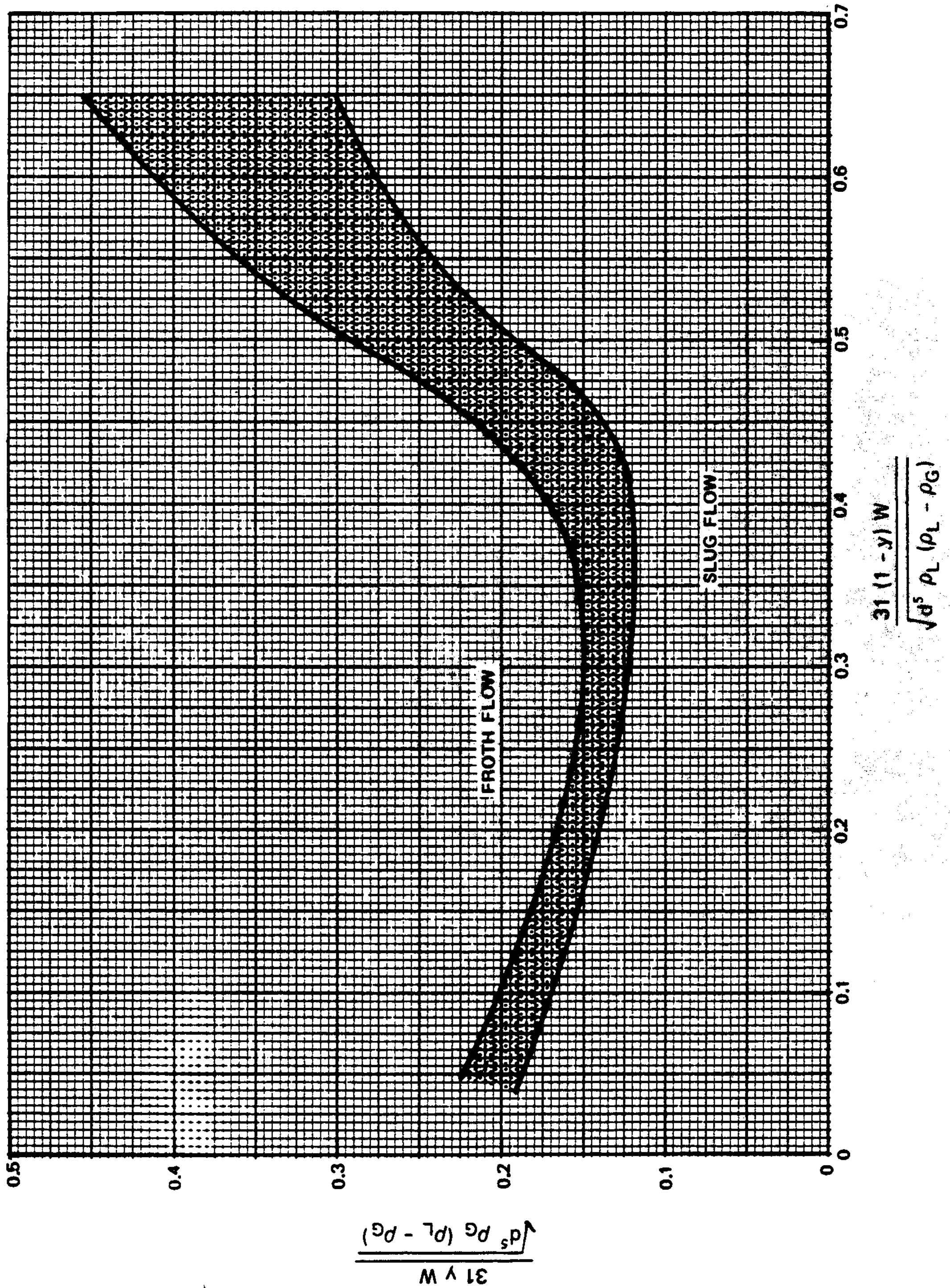
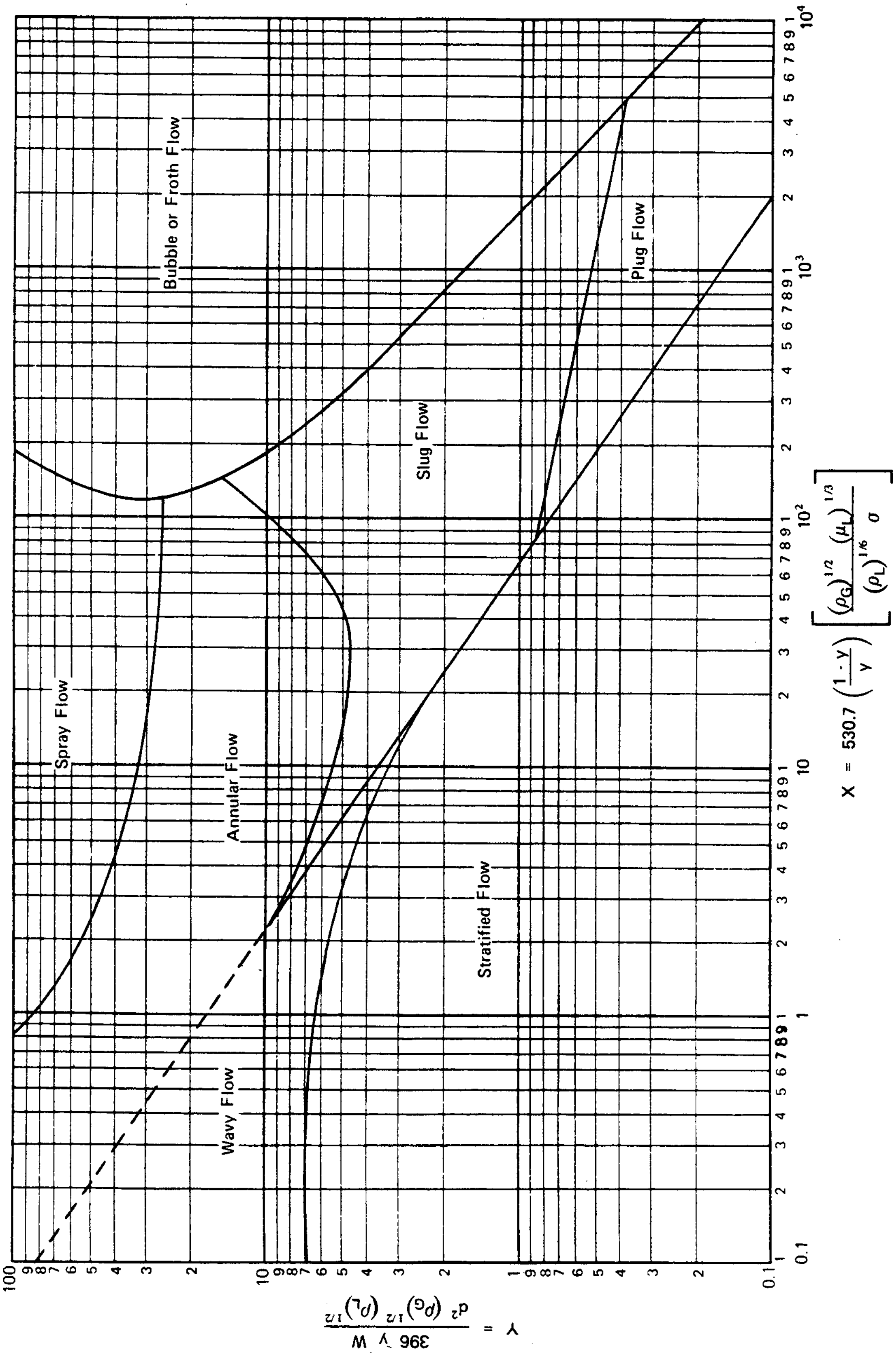


Figure 3
TWO PHASE FLOW REGIMES IN HORIZONTAL PIPE

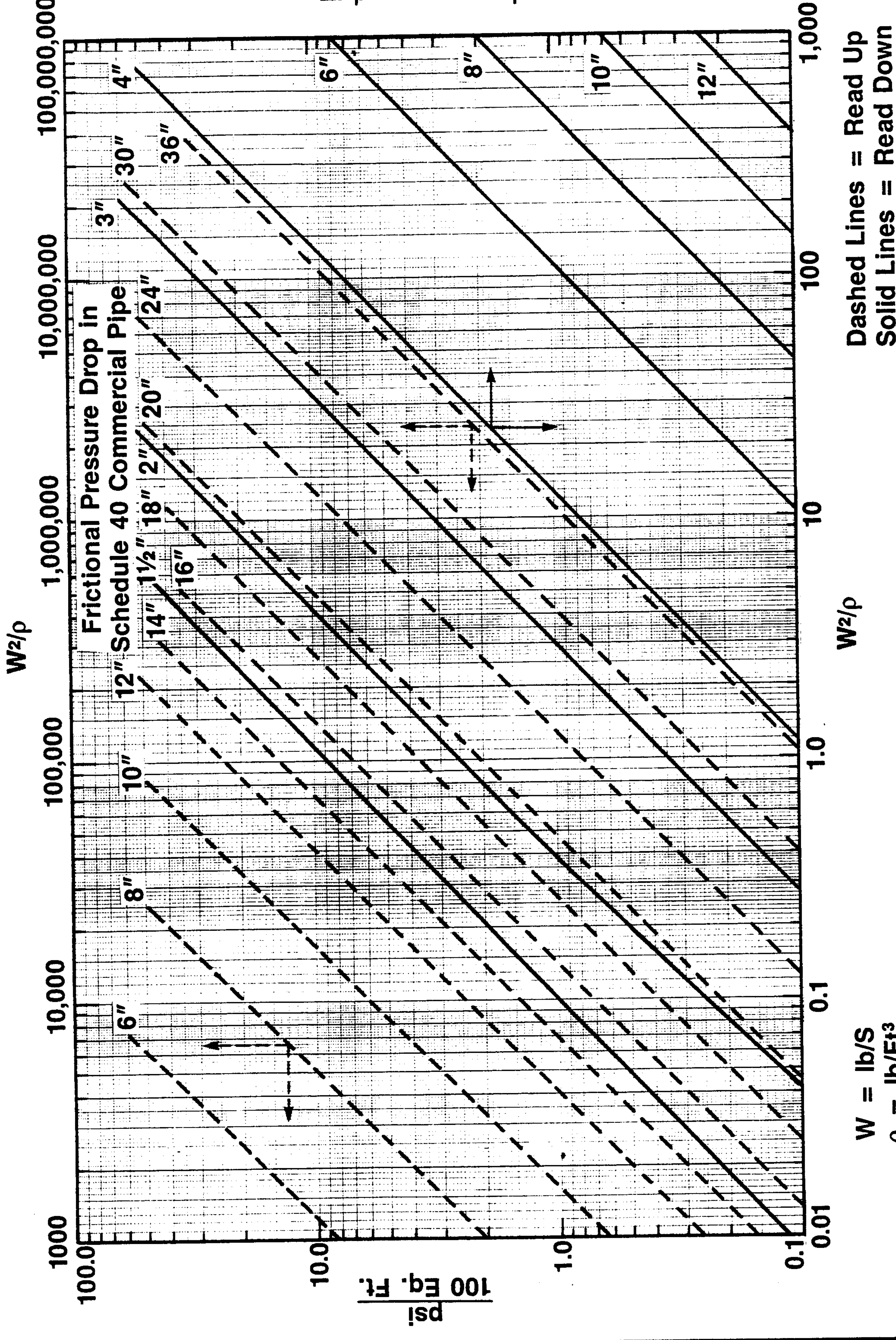
NOMENCLATURE

Y = Weight Fraction Gas or Vapor in Mixture
W = Mass Flow Rate, klb/h
d = Inside Pipe Diameter, in.
ρ_L = Liquid Density, lb/ft^3
ρ_G = Gas or Vapor Density, lb/ft^3
μ_L = Liquid Viscosity, cP
σ = Liquid Surface Tension, mN/m (dyne/cm)



Addendum 6.04

$\Delta P_f/100$ Vs. W^2/ρ



Addendum 6.05

QUICK REFERENCE
PRESSURE LOSS THROUGH FITTINGS

Fitting	No. of Velocity Heads Lost, K (1)
Check Valve (Swing)	2.0
Check Valve (Stop)	8.0
Globe Valve	6.8
Gate Valve	0.16
Angle Valve	3.0
Elbow, Std. (r/d=1)	0.4
Elbow, L.R. (r/d=1.5)	0.28
Bend, 90° (r/d=6)	0.34
Bend, 180° (r/d=1.5)	0.43
Tees, Sideout	1.2
Sidein	1.2
Straight Thru	0.4

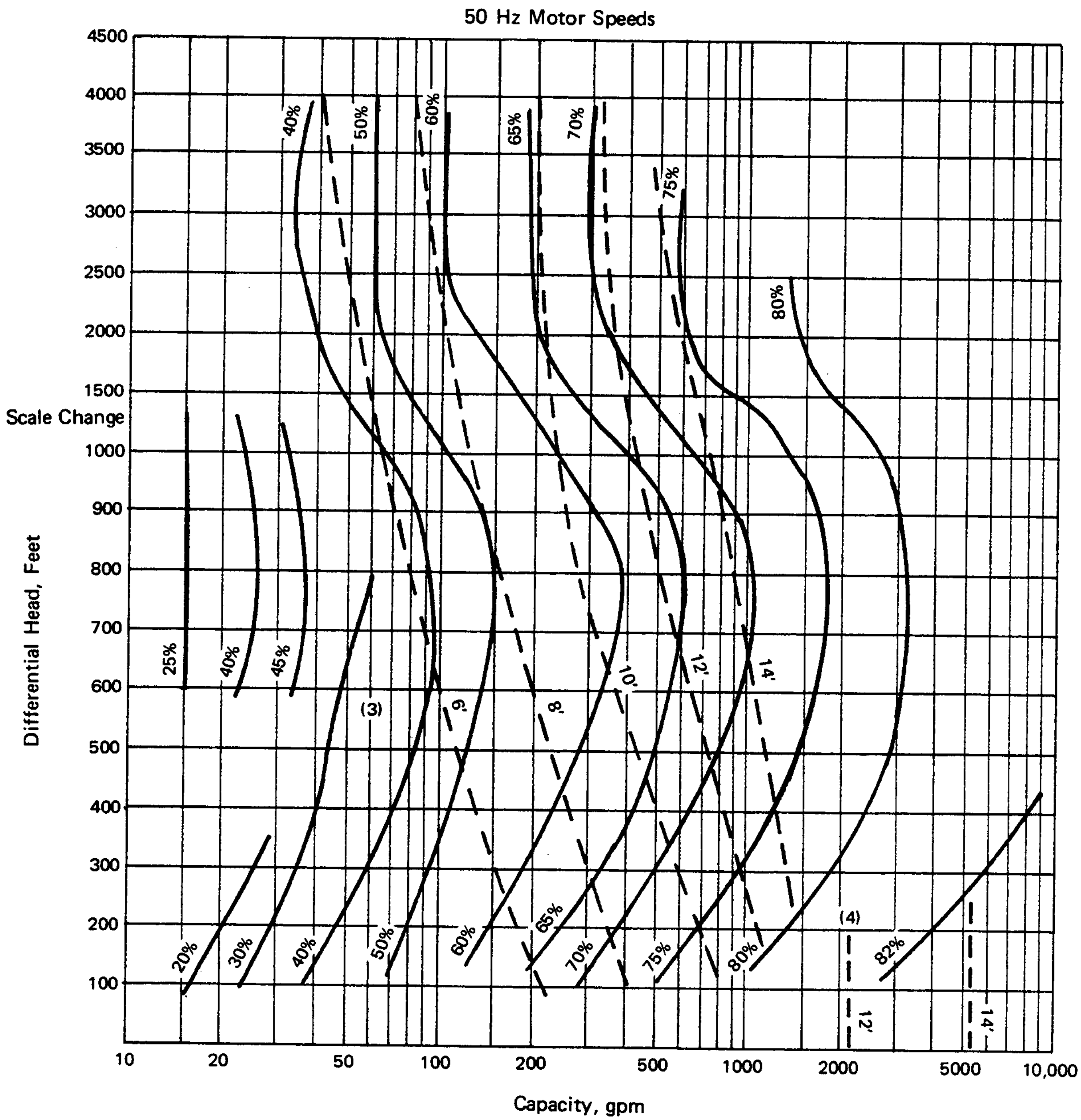
1. All values are based on $f_f = 0.005$.

   For Other Values of f_f, $K' = K \cdot \frac{f_f}{0.005}$

2. Refer to Crane, pages A-26 thru A-31, for additional resistance coefficient (K) values.

Addendum 6.06
PUMP NPSH REQUIRED, PUMP EFFICIENCY, MOTOR EFFICIENCY

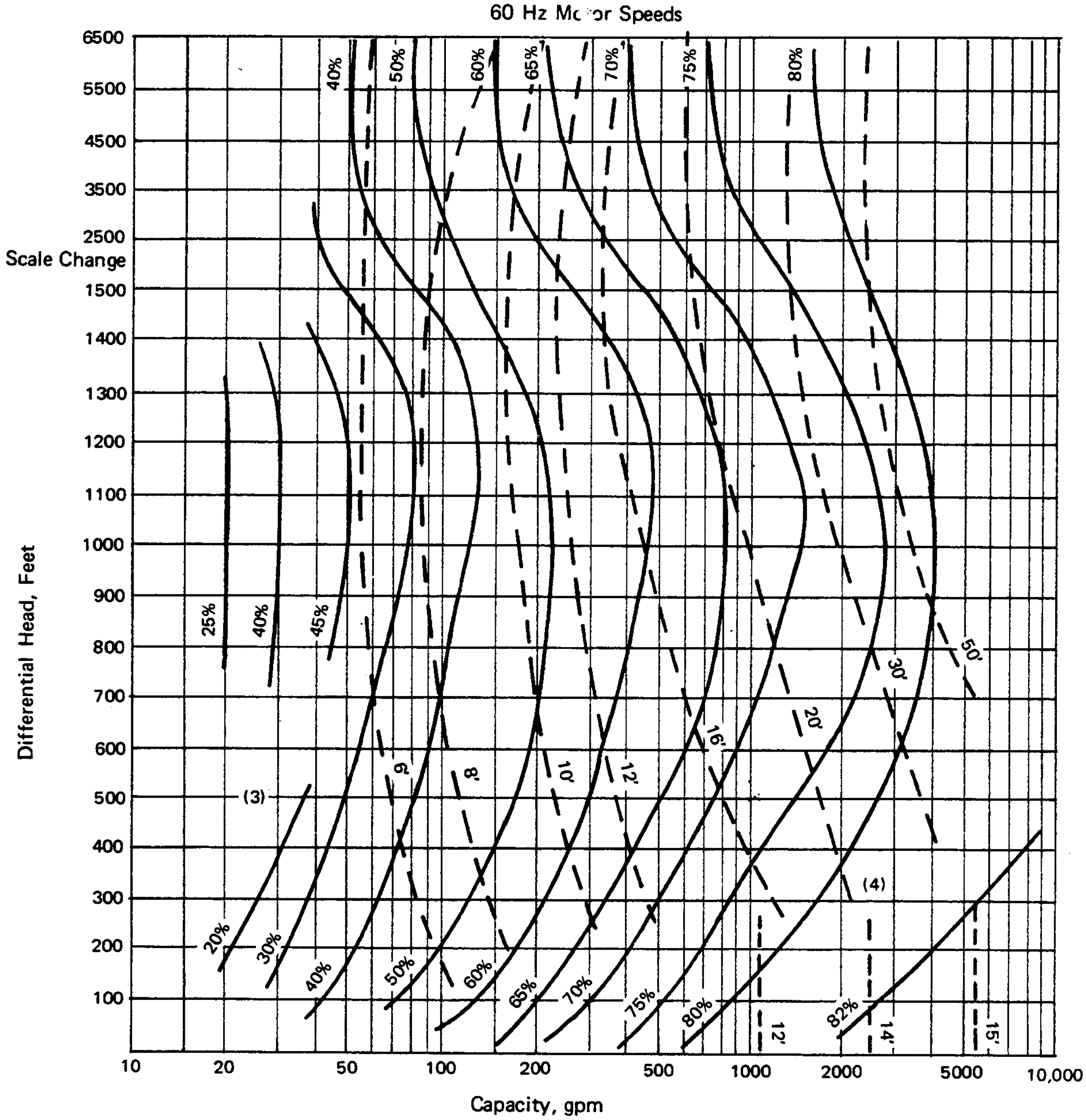
Figure 1
CENTRIFUCAL PUMP PERFORMANCE DATA



Notes:

1. Efficiency has % notation.
2. $NPSH_R$ has foot (') notation.
3. Do not assume less than 6 ft $NPSH_R$ without machinery specialist consultation.
4. $NPSH_R$ discontinuity in this region is due to change in pump speed from 1450 rpm to 2950 rpm.

Figure 2
CENTRIFUGAL PUMP PERFORMANCE DATA

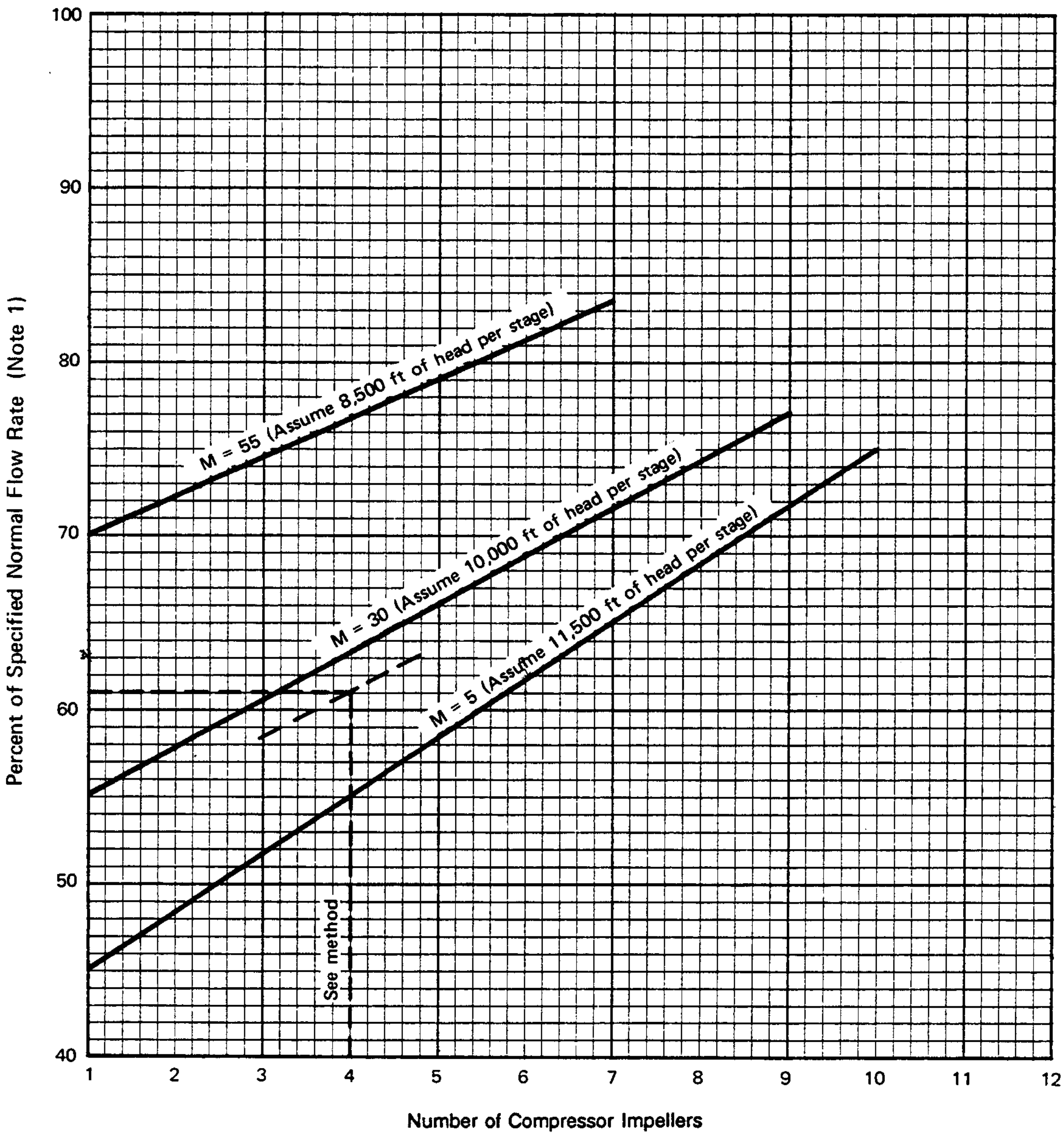


Notes:

1. Efficiency has % notation.
2. $NPSH_R$ has foot (') notation.
3. Do not assume less than 6' $NPSH_R$ without machinery specialist consultation.
4. $NPSH_R$ discontinuity in this region is due to change in pump speed from 1750 rpm to 3550 rpm.

Addendum 6.07
CENTRIFUGAL COMPRESSOR CURVE CHARACTERISTICS

Figure 1
CENTRIFUGAL COMPRESSOR STABILITY RANGE



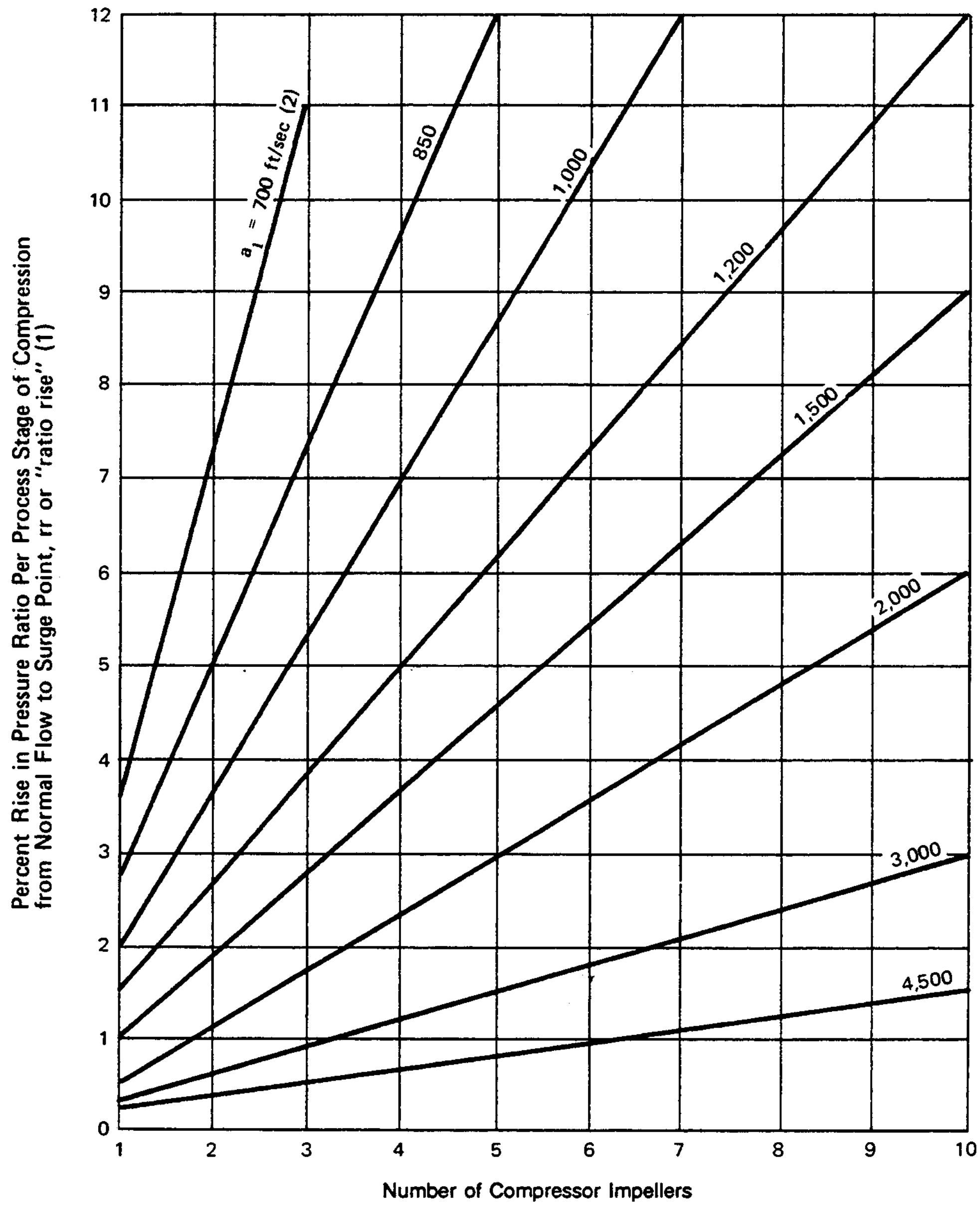
Method:

- Estimate number of stages, based on total head requirement.
- Interpolate between molecular weight lines for actual M.
- Read maximum expected surge flow, as a percent of normal flow. "Stability" equals 100% minus this value.

Note:

(1) A well-designed centrifugal compressor can be expected to have its surge points no higher than this percent of normal flow rate.

Figure 2
CENTRIFUGAL COMPRESSOR CURVE RISE



Notes:

(1) A well-designed compressor can normally exceed the value read from this chart. Specify this value as the minimum acceptable.

(2) a_1 = sonic velocity of gas at compressor inlet conditions

$$= \sqrt{g\ Z_1 k_1 R\ T_1} = \sqrt{\frac{49{,}750\ Z_1 k_1 T_1}{M}}\ \text{, ft/sec.}$$

FUNCTIONS OF PROCESS CONTROL

- Maintain Stability of Operating Conditions at Key Points in the Process
- Provide the Operator and Computer with Information on These Conditions and Means for Adjusting Them
- Automating Operations to Reduce Operator Attention
- Insuring that Operations Are Safe

7.01

PROCESS DESIGN CONCERN

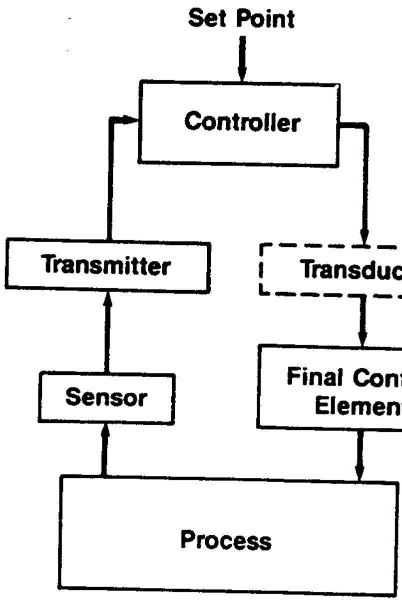
We Will be Concerned With:

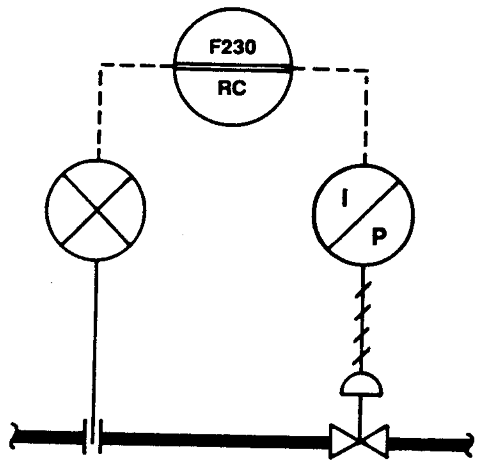
- Type of Control Scheme
- Measuring Element
- Control Element

Leave Controller and Hardware to Instrument Engineer

7.02

ELEMENTS OF CONTROL SYSTEM





7.03

BASIC TYPES OF CONTROL

- Flow
- Level
- Pressure
- Temperature
- Analysis

7.04

FLOW CONTROL

- Provides Unit Material Balance Measurement/Control
- Various Metering Devices Are Available
- Common Applications:
  - — Fractionator Feed and Reflux
  - — Compressor Antisurge
  - — Furnace Pass Distribution

7.05

FLOW MEASURING ELEMENTS

- Orifices Plates
  - — Used Predominantly
  - — 3/1 Rangeability
- Venturi Tubes
  - — Used When Net Head Loss is Very Expensive or for Slurries
- Flow Nozzles
  - — Used in High Velocity Lines
- Rotameters
  - — Used in Very Small Lines, and for Viscous and Fouling Fluids
  - — 10/1 Rangeability

7.06

LEVEL MEASUREMENT

- External Displacer - 14 to 48 in Range
- Differential Pressure - No Range Restrictions
- Ball Float - Occasionally for Alarm or Cutoff Service
- Gage Glass - Utilized to Check Level Measurement

7.07

LEVEL CONTROL

- Objective is Flow Stability Not Constant Level
- Common Applications
  - — Fractionator Bottoms Product
  - — BFW Makeup to Steam Drum
- Holdup Times
  - — Process Feed 5 to 15 Minutes
  - — Products to Tankage 2 Minutes
  - — Reflux 5 Minutes

7.08

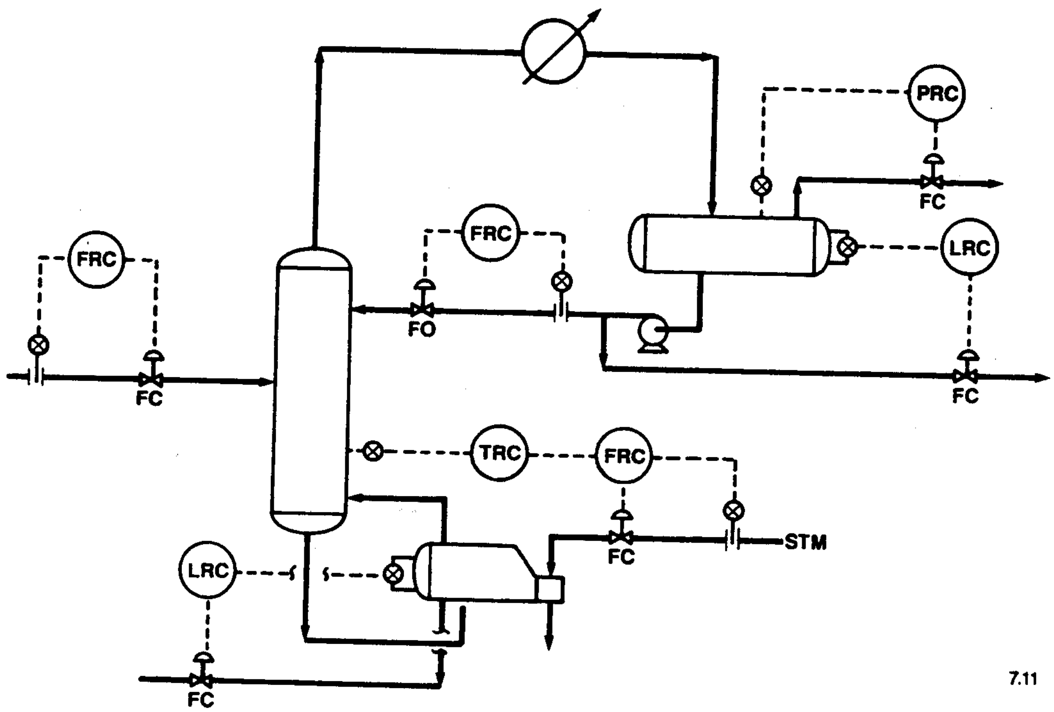
PRESSURE CONTROL

- In Process Design and Operation Pressure Measurement Gives Few Problems
- Common Applications
  - Fuel Gas to Furnaces
  - Fractionator Overhead
  - Compressor Discharge Suction

TEMPERATURE CONTROL

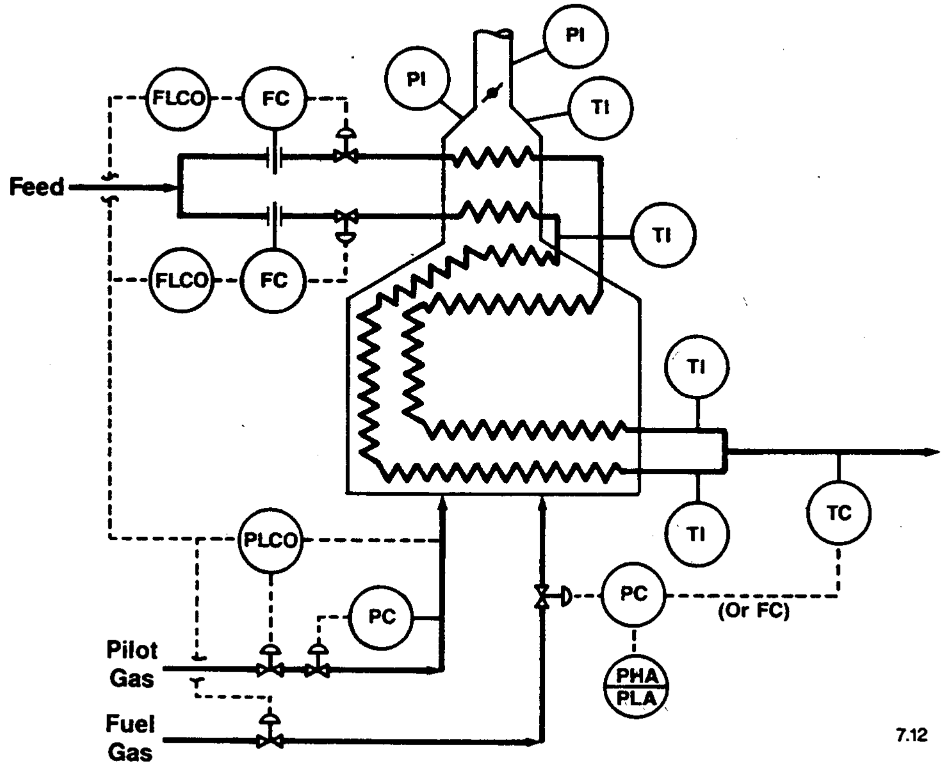
- Temperature Response to Control Valve Movement is Relatively Slow and Complex
- Most Temperature Measured Electrically (Thermocouples)
- Common Applications
  - Furnace Coil Outlet
  - Distillation Tower Overhead Temperature
  - Feed Preheat

FUNDAMENTAL TYPES OF CONTROL



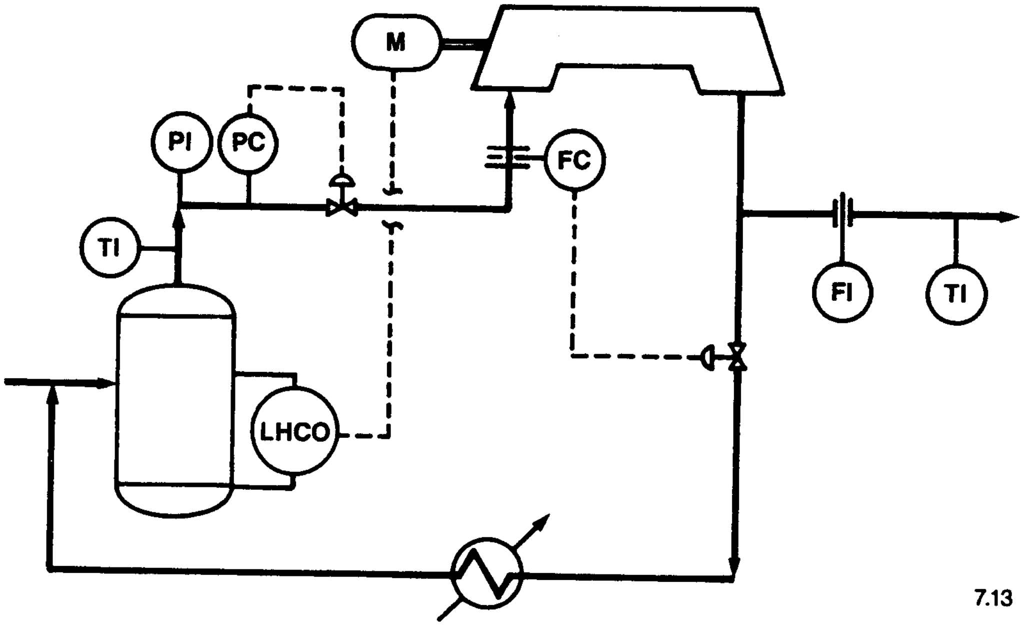
7.11

FURNACE CONTROL



7.12

CENTRIFUGAL COMPRESSOR CONTROL



7.13

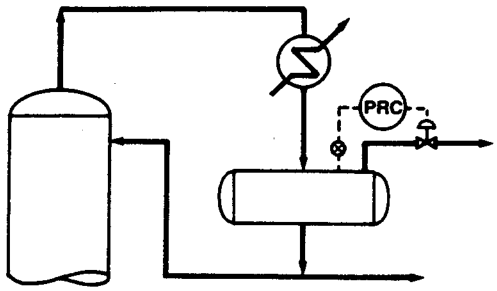
FRACTIONATOR CONTROL PRINCIPLES

- Objectives of Fractionator Control
  - — Maintain Desired Separation
  - — Provide Means for Operator to Adjust Separation as Product Specification Demands Change
- Constant Pressure Required for Satisfactory Control
- Two Types of Fractionator Control
  - — Material Balance
  - — Composition
    - + Analysis
    - + Temperature (Inferred)

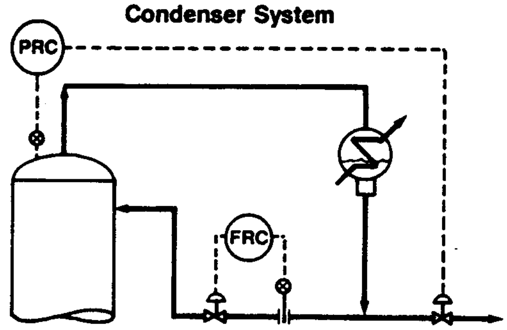
7.14

FRACTIONATOR PRESSURE CONTROL

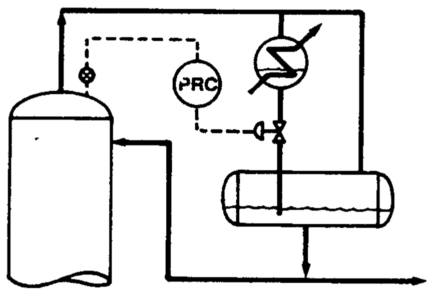
No Compression, Partial Condensation of Overhead Product



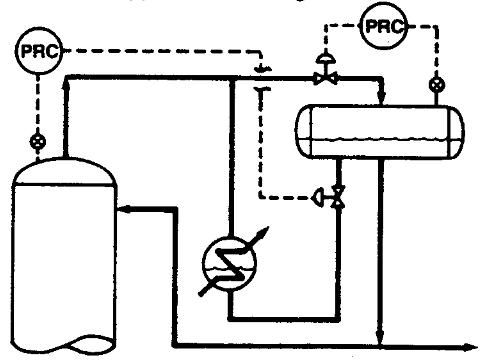
Pressure-Product, Drumless Flooded Condenser System



Flooded Condenser Method, With Drum



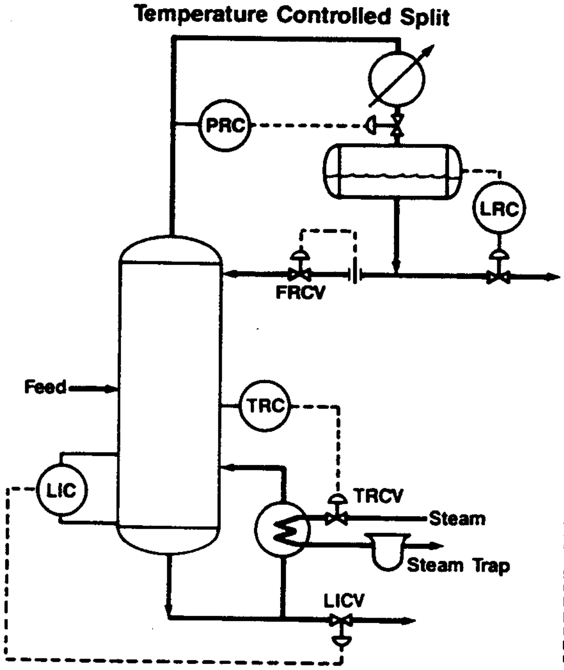
Hot Gas Bypass, Submerged Condenser



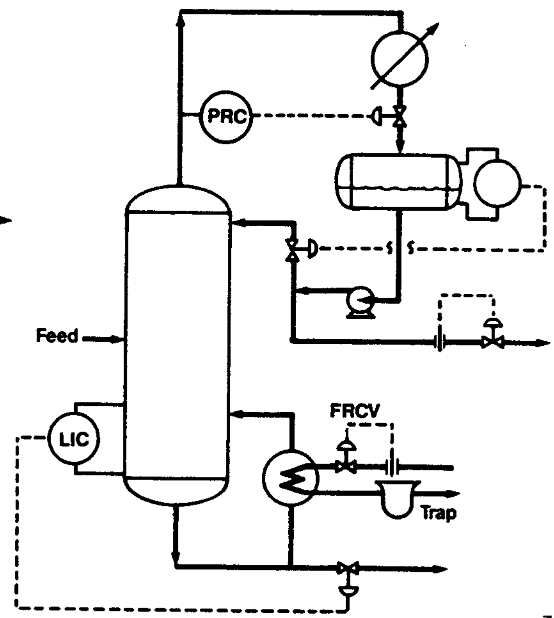
7.15

COMPOSITION vs. MATERIAL BALANCE CONTROL

Temperature Controlled Split



Material Balance Controlled Split



7.16

CONTROL VALVE SPECIFICATION

- **Specify Operating Conditions**
  - **Normal Rate (Max. If > 125% of Normal)**
  - **Pressure Drop at Normal Rate**
  - **Upstream Temp., Press., S.G. or Mol Wt.**

- **Is Fluid: Corrosive? Contains Solids?**

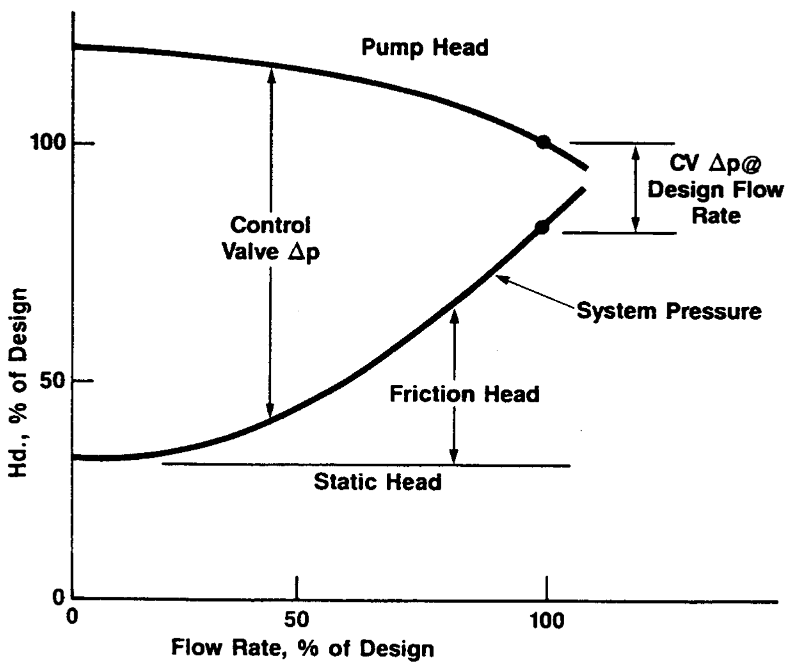
- **Valve Body Type and Flow Characteristics**

- **Action on Failure**

- **Flashing Service**
  - **Specify "FS" for the Valve Pressure Drop.**
  - **Specify that the Fisher Controls Km Method Shall be Used for Valve Sizing. This Method Severely Limits the Pressure Drop Which May be Used in Calculating C_V.**

7.17

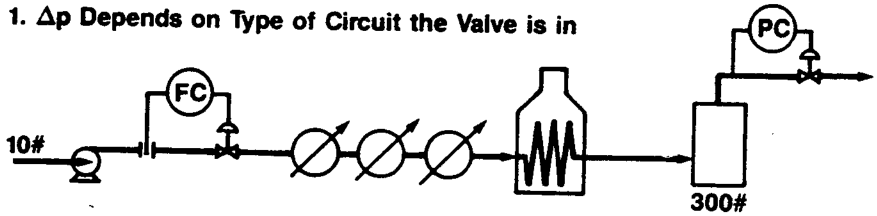
SYSTEM HEAD-CAPACITY RELATIONSHIP



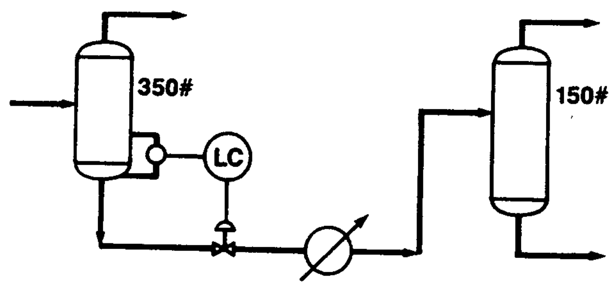
7.18

ASSIGNING CONTROL VALVE ΔP

1. Δp Depends on Type of Circuit the Valve is in



2.



7.19

CONTROL VALVE PRESSURE DROP

- DP · 20% of Circuit Friction Loss (Excl. CV) at Normal Rate (Design) Plus

Press. Drop	For Static Press. (DP)
10%	<200
20 PSI	200-400
5%	>400

- For 3-Way Valves - Use 50% of Exchanger DP

4.5 4 02/164

7.20

CONTROL VALVE SIZING

- Calculate Flow Coefficient C_V at Normal (Design) Flow Rate and Pressure Drop to Get Design C_V.

  Valve C_V = Design $C_V/0.8$

- If Max. Rate > 125% of Normal (Design) Calculate Valve CV for Max. Rate

- Check to be Sure that Valve Size is not Larger Than Line Size. If it is, Increase Allotted ΔP to Reduce Valve Size to Line Size or Less

SIZING CONTROL VALVES

Liquids

$$C_v = Q_L \sqrt{\frac{G_L}{\Delta_P}}$$

Steam

$$C_v = \frac{Q_s}{82} \sqrt{\frac{T}{\Delta P P_2}}$$

Gases

$$C_v = \frac{Q_g}{1360} \sqrt{\frac{G_g\, T\, \mu}{\Delta P \left(\frac{P_1 + P_2}{2}\right)}}$$

C_v = Valve Capacity Rating

Q_L = Liq. Rate @ T, GPM

Q_s = Stm. Rate, #/Hr.

Q_g = SCFH

G_L = Liq. Sp. Gr. @ T

G_g = Gas Sp. Gr. Relative to Air

P = Pressure, PSIA

$\Delta P = P_1 - P_2$

T = Temp., °R, @ Inlet

μ = Gas Comp., @ P_2 & T_2

CONTROL VALVE CHARACTERISTICS

- Equal Percentage (Logarithmic) Provided Unless Otherwise Specified
- Linear Characteristic Specified When Pressure Drop is Constant
- On/Off Should be Specified When Appropriate

7.23

CONTROL VALVE BODY SELECTION

- Refinery Standard is Double Seated Globe Valve
- Butterfly Valves Are Used for High Capacity Low Pressure Drop
- Single Seated Globe Valves Are Utilized When Tight Shutoff is Required
- For Streams Containing High Concentrations of Solids Angle Valves Are Specified

7.24

CONTROL VALVE UTILITY ACTION

Goals in Failure of Control Valve

- **Require Minimum Operator Attention to Put Unit in Safest Standby Condition**
- **Minimize Impact on Other Units**
- **Facilitate Returning to Normal Operation When Failure is Corrected**

DRUM DESIGN

Types of Separator Drums

- Vapor-Liquid
- Liquid-Liquid
- Liquid and Vapor from a Third Liquid Phase

Course Will Concentrate on Vapor-Liquid Separation Since it is far More Common

8.01

DRUMS—BASIC DESIGN CONSIDERATIONS

- Critical Entrainment Velocity
- Drum Orientation—Vertical or Horizontal
- Liquid Holdup Requirements
- Liquid Level Measurement Requirements
- Inlet Nozzle Type, Number and Orientation

8.02

DRUMS—BASIC DESIGN CONSIDERATIONS (Continued)

- Inlet Distributor, If Any
- Anti-Vortex Baffles Needed?
- Relation of Length to Diameter
- Possibility of Liquid Re-entrainment
  —Need for Crinkled Wire Mesh Screen (CWMS)
- Severe Foaming Service?

T → ↑V sedim [illegible] burbujas.

$V_{sedimentacion}$ se aplica [illegible]

8.03

CRITICAL ENTRAINMENT VELOCITY

- Ensures Vapor Velocity is Sufficiently Low to Prevent Excessive Liquid Carryover. Not Related to Sonic Velocity.
- $V_c = 0.157 \left(\frac{\rho_L - \rho_G}{\rho_G}\right)^{0.5}$ ρ_G = Gas Density, lb/ft^3 ρ_L = Liquid Density, lb/ft_3

CRITICAL ENTRAINMENT VELOCITY (Continued)

Typical Values of V_c with CWMS:

- Surge Drums, Distillate Drums — 100-125%
- Compressor Suction & Interstage — 100-225%
- Fuel Gas K.O. — 100%
- Steam Drums — 100%

Minimum Vapor Space Should be 12 Inches or 20% of the Drum Diameter.

Also a Minimum Distance From the LLL to the Outlet Nozzle Entrance (Details Later)

DRUM ORIENTATION

- Horizontal
  + With One Inlet Nozzle More Efficient Than Vertical Drums (Cross-Flow vs Countercurrent Flow)
  + Almost Always Used When Relatively Large Quantities of Liquid and Vapor Must Be Separated
  + More Flexible in Choice of Nozzle Arrangement (e.g., Split Flow)
  + Always Chosen for Liquid-Liquid Systems and When Flow in Inlet Piping is Slug or Bubble Flow
  + Smaller in Volume for High Liquid Loading Service
- Vertical
  + Commonly Used for Very Low Liquid Loading (Fuel Gas K.O.)
  + Main Advantage is Smaller Plot Area

TYPICAL LIQUID HOLDUP REQUIREMENTS

• Surge Drums, Distillate Drums	— 2-15 Minutes
• Compressor Suction & Interstage	— 5-10 Minutes or 10 Minutes of Liquid Spill From Upstream Unit.
• Fuel Gas K.O.	Volume of 50 Feet of Inlet Line; But After Absorber, Use 5 Min. of Lean Oil.
• Steam Drums	— 2 Minutes on Feedwater or ⅓ the Volume of Steam Generator and Piping

LIQUID LEVEL MEASUREMENT

External Displacers—Measures the Change in Buoyancy of the Displacer Tube as the Level Changes Over the Length of the Tube.

- Available in 8, 14, 32, 48, 60, 72, 84, 96 and 120″. Standard Ranges
- Exxon Does Not Use E.D. Greater Than 48″ Range. (For larger Ranges, Use Differential Head Devices.)
- The Range is the Distance Between HLL and LLL. This Range Must Equal a Standard Range When Using an External Displacer, and Should Be Shown on the Drawing
- E.D. Connections are Preferably Side-Side, But Top-Side and Bottom-Side are Acceptable Where Necessary

INLET NOZZLE/DISTRIBUTOR ARRANGEMENT

- Surge Drums, Distillate Drums
  - 90° Elbow Directed at Head (Horizontal Orientation)
  - Slotted Distributor
- Most Others,
  - Slotted Distributor (Especially for Vertical Orientation)

8.09

ANTI-VORTEX BAFFLES

- Consist of Three Square Sections of Subway Grating Evenly Spaced One Above the Other
- Grating Bars are 1″ by 1/8″ on a 1″ by 4″ Spacing.
- Maximum Distance Between Section = 6″. In an Exxon Job, Details are Covered by BP5-2-1.
- For Vertical Drums, Lowest Section = d/2″ Above Liquid Outlet Nozzle
- For Horizontal Drums, Lowest Section = 2″ Above Liquid Outlet Nozzle

(d = Outlet Nozzle ID)

- Length of a Side Should Be 4 Times the Outlet Nozzle Diameter, Or Half the Drum Diameter, Whichever is Smaller

8.10

ANTI-VORTEX CONSIDERATIONS

- To Reduce Furthur the Possibility of Vapor Carryunder, Calculate h_{LL}:

$$h_{LL} = \frac{8.4\ Q^{0.4}}{(1 - \frac{\rho_G}{\rho_L})^{0.2}}$$

Q = Liquid Discharge Rate ft^3/s
ρ_G = Vapor Density lb/ft^3
ρ_L = Liquid Density lb/ft^3

h_{LL} = Minimum Distance Between LLL and the Outlet Nozzle Entrance (Not Always the Bottom of Drum)

- If h_{LL} is Less Than 9", Then Use 9" as the Minimum Distance

OVERALL GEOMETRY

Ratio of Length to Diameter

- Very Wide Range of Values Found — 1 to 5
- Most Common (and Economical) — 2.5 to 4

LIQUID RE-ENTRAINMENT

- After Collection in the Drum, Liquid Can Be Swept from the Surface and Entrained to the Outlet (Especially at High Pressure and Temperature)

- Factors Involved:
  - — Velocity in the Inlet Nozzle
  - — Distance from Inlet Nozzle to Liquid Surface
  - — Physical Properties of Fluids
  - — Inlet Nozzle Type (Flush, 90° Elbow, Slotted T)

- Factors Define a Maximum Inlet Velocity

- Can Increase Inlet Nozzle Size to Greater than Line Size to Prevent Entrainment

8.13

DESIGN PROCEDURE

(1) Tabulate Rates and Physical Properties of the Fluids to be Separated (H&MB)

(2) From the Type of Service and Prior Experience, Select the Preferred Geometry for the First Pass Design. For Exxon See EDP V-A Pages 27-28.

(3) Calculate Vapor Flow Area Required

(4) If CWMS is Required, Calculate Size and Position

(5) Calculate Liquid Volume (and Liquid Cross Section) Required. Calculate Drum Diameter and Length.

8.14

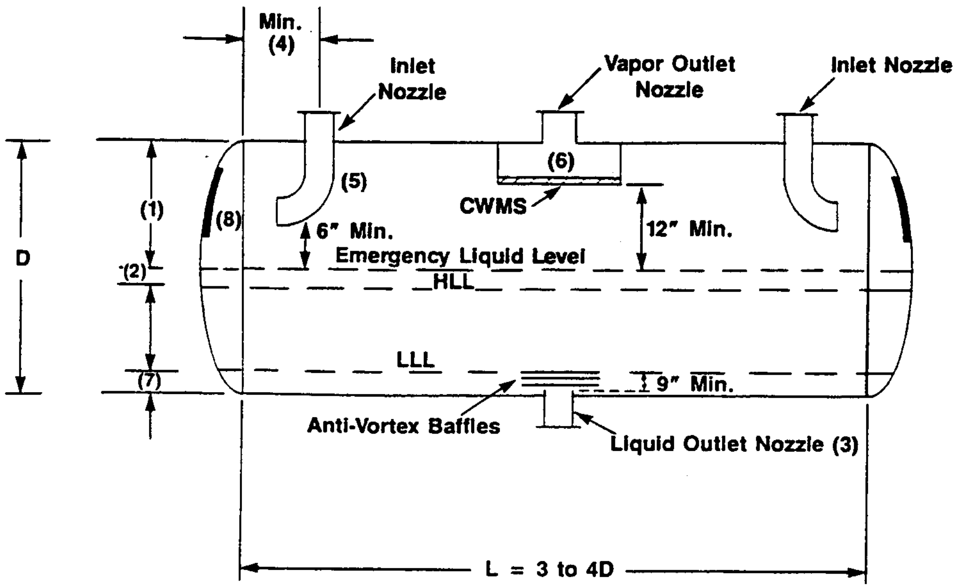
DESIGN PROCEDURE
(Continued)

(6) Select Type of Inlet Nozzle/Distributor; Calculate Diameter of Nozzle to Prevent Re-entrainment

(7) Calculate Size and Positions of Anti-Vortex Baffles

(8) Prepare a Sketch Showing Nozzles, Instrument Tap Locations and Positions of Interface (LLL, HLL and ELL if Required)

TYPICAL DIMENSIONS OF HORIZONTAL CYLINDRICAL DRUMS



See Next Text Page for Notes

8.16

NOTES:

1. Design for Appropriate % V_C. Minimum Vapor Space Is 12 Inches or 20% of Drum Diameter Whichever Is Greater.
2. Ten Minutes Holdup If Applicable, Otherwise ELL Is HLL.
3. If Water Drawoff Is Present the Hydrocarbon Liquid Outlet Nozzle Should Extend 4" above the Bottom of the Drum.
4. Minimum Distance Considering Reinforcement and Fabrication Requirements
5. Inlet Nozzle Selection Based on Prior Experience. For Exxon Jobs See EDP V-A Pages 9-16 and Table 3 Page 31.
6. The Minimum Distance Above CWMS Is Calculated From

$$ho = \frac{12\ D_{CWMS} - do}{2}$$

7. The Minimum Distance Between LLL and The Outlet Nozzle Is h_{LL} or 9 Inches, as Previously Discussed.
8. Impingement Baffles Should be Installed Opposite 90° Elbow Inlet Nozzles to Protect the Drum Shell. The Baffle Diameter Should be Twice the Inlet Nozzle Diameter.

8.17

GENERAL PROCESS
DESIGN CONSIDERATIONS

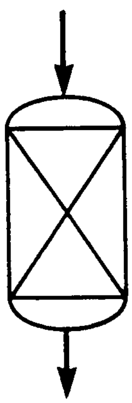
Will Cover—

- Design Temperatures
- Design Pressures
- Piping Flange Ratings (Classification)
- Equipment Spacing and Location

9.01

OPERATING TEMPERATURE & PRESSURE

- The Process Fluid Temperature Predicted for Normal Operations.
- The Pressure to Which Equipment or Piping is Normally Subjected While in Service.



Ho	Kero.	Naph	Psig
625°	625°	500°	350
650°	625°	600°	330

9.02

MAXIMUM/MINIMUM OPERATING TEMPERATURES & PRESSURES

- Normal Fluid Temp. and Pressure
  ± Deviations From Normal
- Include Startup, Shut Down, Depressuring, Alternate Operations, Control Requirements, Upsets and Flexibility.

9.03

DESIGN TEMPERATURE

- The Metal Temperature Representing the Most Severe Condition of Coincident Pressure and Temperature.
- Can be a Maximum or Minimum. Sometimes Shown as Both.
- Metal Temperatures are Same as Fluid Temp. for No Internal Insulation.
- Used for Mechanical Design of Equipment.

9.04

CRITICAL EXPOSURE TEMPERATURE (CET)

- The Minimum Metal Temperature at Which a Component Will be Subjected to More than 25% of its Design Pressure.
- Usually Occurs During Startups, Shutdowns or Depressuring.
- Important for Auto Refrigerating Systems and Cryogenic Equipment.
- Results in Minimum Impact Requirements that Assure Adequate Material Toughness to Prevent Catastrophic Brittle Fracture.

9.05

DESIGN PRESSURE

- The Maximum or Minimum (Vacuum) Pressure Used to Determine Minimum Wall Thickness
- Design Pressure is Specified at the Top of the Vessel Unless Stated Otherwise.
- A Sufficient Margin Must be Allowed Above Maximum Normal Operating Pressure to Avoid Leakage or Frequent Opening of Safety Valves

9.06

SHORT-TIME DESIGN BASIS FOR PIPING

- Variations From Normal Set Design Conditions
- If Variations are Infrequent & of Short Duration, Pressure-Temperature Ratings May be Adjusted
- 10 Hours/Time or 100 Hrs/Year
- Permits an increase of 33% in Pressure Rating or Allowable Stress
- Applies for Piping Only

9.07

INTERMEDIATE-TIME DESIGN BASIS FOR PIPING

- Variations From Normal Design Conditions Usually Associated With an Emergency Situation or Alternative Operating Condition
- 50 Hours/Time or 500 Hrs/Year
- Permits an Increase of 20% in Pressure Rating or Allowable Stress
- Applies for Piping Only

9.08

DESIGN TEMPERATURE GUIDELINES FOR VESSELS

A. Design Temperatures 120°F and Below

1. Brittle Fracture Becomes a Problem
2. Process Designer Should be Realistic in Setting Minimum Temp.
3. Break Points for Toughness Requirements are 60°, 32°, −20°, −50°F, −150°F (ie. −20 Requires Killed Carbon Steel)
4. Where Min. Operating Temp. are Above Ambient, in Warm Climates Use Lowest One-Day Mean Temperature (Usu. 60 to 70°F)

DESIGN TEMPERATURE GUIDELINES FOR VESSELS (Continued)

B. Design Temperatures Above 120°F

1. *120 to 650°F.* Stresses for C.S. are Essentially Constant; Therefore, 50°F is Normally Added to Operating Temperature to Get Design Temperature
2. *650 to 850°F.* Use Very Minimum Increment for C.S. Because Allowable Stresses Decrease With Increasing Temperature
3. *850°F+.* Minimum Temperature Should be Added and Alternatives of Internally Insulated or Alloy Equipment Should be Considered
4. Careful Selection of Design Temperature and Pressure Can Minimize Costs by Not Arbitrarily Going to the Next Higher Piping Class or Higher Cost Materials

DESIGN TEMPERATURE GUIDELINES FOR VESSELS (Continued)

5. Design Metal Temperature for *Internally Insulated* Vessels is Normally Set at 650°F for Fluid Temperatures Above 650°F
6. Equipment Subject to Steamout Should be Designed for the Steam Temperatures
7. Consider Effect of Cooling Water Failure, Air-Fin Failure, Reflux Pump Failure, Decoking, etc.

DESIGN TEMPERATURE GUIDELINES FOR VESSELS (Continued)

C. Materials for Low Temperature Service

1. Down to −20°F Use Steel That Meets Proper Impact Requirements
2. From −20° to −50°F Use Fully Killed and Fine Grained Steels
3. From −50° to −150°F, 3½% Ni Alloy Steels Would Be Used
4. Below −150°F, Use 9% Ni Alloy Steel Down to −320°F or Austenitic Stainless Steels (304 or 316) Down to −425°F

DESIGN TEMPERATURE GUIDELINES FOR PIPING

1. Normally Do Not Specify Design Temperature
2. Do Specify Operating Conditions—Normal and Maximum to Predict Required Flange Class.Must Consider Startup, Shutdown, Steamout and Alternate Operations
3. Design Stress is a Function of Temperature Throughout the Range of 100 to 650°F; Therefore, Must be Careful in Setting Operating Conditions
4. Design Temperatures for Uninsulated Flanges Can Be 10% Less Than the Maximum Operating Fluid Temperature
5. Low Temperature Considerations for Vessels Also Apply to Piping

DESIGN PRESSURE CONSIDERATIONS

Maximum Operating Pressure Should Be Determined Based on Consideration of Pressure Variations Due to Changes in

1. Vapor Pressure
2. Static Head
3. System Pressure Drop
4. Pump or Compressor Shut Off Pressure
5. Ambient Temperature Changes
6. Feedstock Changes
7. Density Changes

DESIGN PRESSURE CONSIDERATIONS (Continued)

- Vessels—D.P. is the Greater of
  1. Max. Operating Pressure (psig) + 10%
  2. Max. Operating Pressure (psig) + 25 psi

  This Applies When a Conventional S.V. is Used
  Exceptions are—
  a. For Stable Operations Below 250 psig Use Greater of Max. Operating Pressure Plus 10% or Max. Operating Pressure + 15 psig
  b. Self-Limiting Pressure Systems. A Blower, Operating at 6 psig, that Can Develop Only 9 psig. The D.P. Would be 9 psig
- A Min. Vessel D.P. of 16 psig is Generally Used to Fall Within ASME Code Unless There are Strong Incentives to do Otherwise

9.15

DESIGN PRESSURE CONSIDERATIONS (Continued)

- Piping—D.P. for Piping Must be Consistent With the Design Pressure for Vessels and Equipment to Which it is Attached. On a Short-Time Basis, it is Permissible to Increase the Pressure at Operating Temperature by 33% or 20% for Intermediate-Time Basis
- Equipment Downstream of Centrifugal Pumps
  D.P. is the Greater of:
  1. Normal Pump P_1 + 1.2 (Normal Pump ΔP)
  2. Max. Pump P_1 + Normal Pump ΔP
  3. Actual Pump Shutoff Pressure Plus Normal Pump P_1
  4. If Shutting Off Pump Flow <u>Causes</u> Suction Pressure to Rise to Maximum, then Design Equipment for Max. Pump P_1 + 1.2 (Normal Pump ΔP). Example: Failure Closed of a Reflux Control Valve.
- Suction Piping of Parallel Pumps Must be Good for 75% of the Discharge Side D.P. and D.T.
- Pump Casing Design Pressure is the Sum of the Max Suction Pressure and the Max Differential Pressure (1.2 Times Normal DP) at Rated Conditions

9.16

PIPING FLANGE RATINGS

The Designer Determines the Appropriate Flange Rating Corresponding to the Required Design conditions. By Reducing Maximum Specified Temperature or Pressure only Slightly, it Might Be Possible to Reduce Flange Rating.

9.17

PIPING FLANGE RATINGS

- Flanges or Piping Components May Be Used Freely at or Below Their Pressure-Temperature Rating.
- Classes Up to 24 in. Diameter are 150, 300, 400, 600, 900, 1500 and 2500 for Carbon Steel and Alloy Flanges
- Flanges Now Designated by "ANSI Class", Such as "Class 150" or "150 ANSI".
- Years Ago, Flanges Were Rated in PSIG, and One Still Hears "150 Pound Flange", or "600 Pound Flange Rating"

9.18

PIPE FLANGE RATING-EXAMPLE

Max. Cont. Oper.	Max. Cont. Oper.
DT 388°F	422°F
DP 210 psig	210 psig
F.D.T.=0.9(388)=350°F*	0.9(422)=380*
Max. Press. Class 150 = 215 psig for C.S. Use 150# Flange	Max. Press. for Class 150 = 206 psig Use 300# Flange

*Uninsulated Flanges

EDPM Sect 11 Pg 21
(ASME/ANSI B16.5—1988)–See Handout

9.19

PIPE FLANGE RATING-EXAMPLE

- Max. Continuous Operating Pressure=660 psig
- Max. Continuous Operating Temperature=500°F
- Short Term Operating Pressure=960 psig
- Pipe is Carbon Steel & Flanges Uninsulated
- Determine Flange Rating to Use
  1. Max. Continuous D.P. for Class 300 Flanges
  2. Max. Pressure to Which the Selected Flange May Be Subjected on a Short-Time Basis
  3. Max. Pressure to Which the Selected Flange May Be Subjected on an Intermediate Term Basis

9.20

PIPE FLANGE RATING-EXAMPLE (Continued)

D.T. for Uninsulated Flange=0.9(500)=450°F

Pressure =660 psig

From Table 1 Sect. II, EDP, Page 21 (Or Handout) Flange Rating=400

From Table 1 Sect. II, EDP, (Or Handout) Maximum Pressure for Class 300 Flanges=617 psig

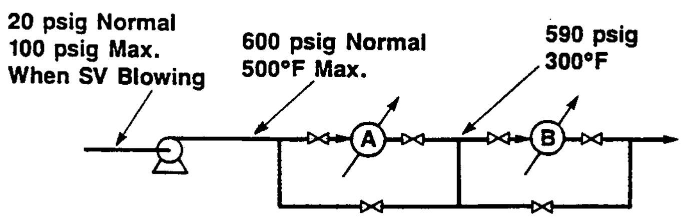
On Short-Time Basis, Can Overpressure Class 400 Flanges by 1/3=(822) (1.33)=1093 psig

On Intermediate-Time Basis, Can Overpressure by 1/5 (822) (1.20)=986 psig

Since Class 400 Flanges are Good for 1093 psig, They are OK for Short-Time as Well as Continuous Conditions

9.21

DESIGN PRESSURE & TEMPERATURES DOWNSTREAM OF PUMP



Max. Pump Discharge Press = 20+1.2(580)=716 psig ←
or 100+580 =680 psig

Design Pressure of Exch. A=716 psig
Design Pressure of Exch. B=716 psig
Design Temperature of Exch. A=500°F
Design Temperature of Exch. B=500°F

Use Class 400 Flanges for C.S.

9.22

D.T. & D.P. FOR TOWERS

DeC$_3$, Butane Splitters, Gasoline Rerun—

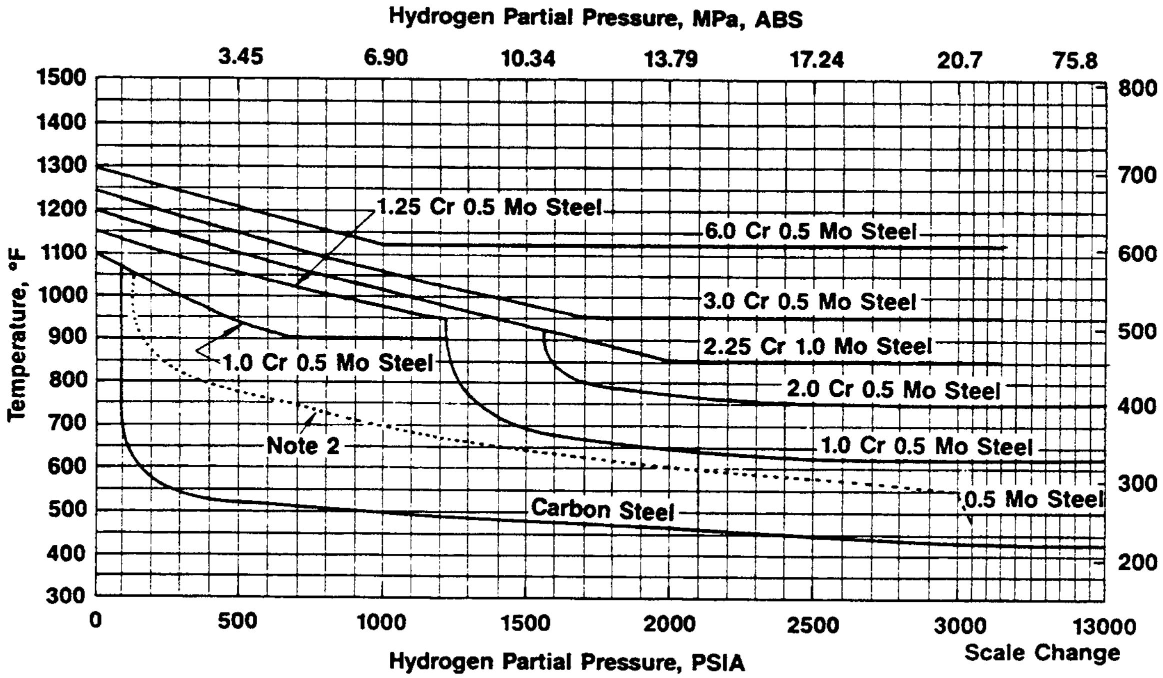
Min. Temp. Considerations:

Can the Towers be Autorefrigerated?
Will the Towers be Tested With Water?

Design Press. Considerations:

What is Maximum Liquid Level in Bottom of Tower?

CORROSION DESIGN CURVE 6A
OPERATING LIMITS FOR STEELS IN HYDROGEN SERVICE
(API 941, JUNE 1977)



EQUIPMENT SPACING

Objectives of Spacing Recommendations are:

- To Permit Access for Firefighting
- To Permit Operator Access for Emergency Shutdown
- To Minimize Impact of Fire on Adjacent Facilities
- To Ensure Emergency Facilities are Not Subject to Fire Damage
- To Segregate High Risk Facilities
- To Separate Continuous Ignition Sources From Probable Sources of Flammable Release
- To Avoid Danger to Facilities Beyond the Adjacent Property Line
- To Permit Access for Normal Operation and Maintenance
- To Ensure Site Security

9.25

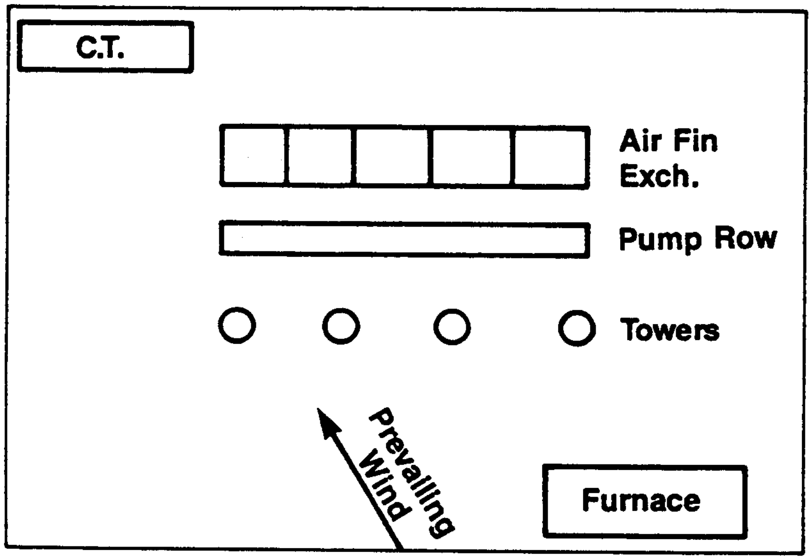
RELATIVE EQUIPMENT LOCATION

When Flexibility Permits, Use the Following Guidelines—

1. Furnaces—Locate Upwind of Other Equipment
2. Cooling Towers—Locate Downwind
3. Air Fin Exchangers—Locate Opposite Side of Fractionation Tower From Furnaces
4. Pumps—
   Locate Opposite Side of Fractionation Tower From Furnace

9.26

EQUIPMENT LOCATION



9.27

FOR USE WITH PROBLEM NO. 11

PRESSURE-TEMPERATURE RATINGS FOR NORMAL CARBON STEEL FLANGES, FLANGED VALVES, AND FITTINGS (1, 2)

Pressures are in pounds per square inch, gage (psig)

Temperature, °F	Class 150	Class 300	Class 400	Class 600	Class 900	Class 1500	Class 2500
-20 to 100	285	740	990	1480	2220	3705	6170
125	279	724	967	1447	2171	3622	6034
150	272	707	945	1415	2122	3540	5897
175	266	691	922	1382	2074	3457	5761
200	260	675	900	1350	2025	3375	5625
225	252	670	894	1341	2011	3351	5586
250	245	665	887	1332	1997	3327	5547
275	238	660	881	1324	1984	3304	5509
300	230	655	875	1315	1970	3280	5470
325	222	650	867	1304	1952	3252	5422
350	215	645	860	1292	1935	3225	5375
375	208	640	852	1281	1917	3197	5327
400	200	635	845	1270	1900	3170	5280
425	192	626	834	1252	1874	3126	5207
450	185	617	822	1235	1847	3082	5135
475	177	609	811	1217	1821	3039	5062
500	170	600	800	1200	1795	2995	4990
525	162	587	782	1174	1756	2930	4882
550	155	575	765	1147	1717	2865	4775
575	147	562	747	1121	1679	2800	4667
600	140	550	730	1095	1640	2735	4560
625	132	542	722	1085	1625	2710	4517
650	125	535	715	1075	1610	2685	4475
675	117	535	712	1070	1605	2675	4457
700	110	535	710	1065	1600	2665	4440
725	102	520	690	1037	1555	2592	4320
750	95	505	670	1010	1510	2520	4200
775	87	457	610	917	1372	2290	3815
800	80	410	550	825	1235	2060	3430
825	72	340	452	680	1020	1700	2830
850	65	270	355	535	805	1340	2230
875	57	220	292	440	660	1100	1830
900	50	170	230	345	515	860	1430
925	42	137	185	276	412	687	1145
950	35	105	140	205	310	515	860
975	27	77	105	155	232	387	645
1000	20	50	70	105	155	260	430

NOTES:
1. Based on ANSI B16.5-1977.
2. Linear interpolation is permissible to find pressure ratings at temperatures between listed temperatures.

REFERENCES FOR SAFETY IN PROCESS DESIGN

1. API Recommended Practices 520, Parts I and II, and 521.
2. ASME Code ANSI Standard B31.1.
3. Air Pollution Calculations—Dispersion Models
4. PC Models Available for Network Analyses:
   - TRI-FLARE.
   - F37 Flare Network Program.
   - 'SAFE' Flare Network & Contingency Analysis
5. Within Exxon:
   - Exxon Design Practices, Section XV, completamente [illegible] INTCANT
   - ER&E Basic Practices
   - Local Engineering Standards (e.g. EXES)

10.01

SAFETY IN PROCESS DESIGN

- General Principles
  - The Design of Safety Into a Process is the Responsibility of the Process Design Engineer.
  - Every Design Must be Safe Against Reasonable Causes of Failure. Adequate Facilities Must be Incorporated Into the Design to Prevent Fires, Explosions, and Accidents.
  - All Process Designs (Grass-Roots and Revisions) are Reviewed With the Safe Operations Committee to Ensure that Safety Standards are Being Followed.
- General Requirements per ASME Pressure Vessel Code
  - Equipment is Protected From Exceeding Design Pressure By:
    - + 10% (Single Pressure Relief Valve, Excluding Fire Contingency)
    - + 16% (Multiple Pressure Relief Valves)
    - + 21% (Fire Contingency)

10.02

DEFINITIONS

1. AUTOIGNITION TEMPERATURE—The Lowest Temperature Required to Cause Self-Sustaining Combustion, Without Ignition by Spark or Flame. (Typical 600°F)
2. CONTINGENCY—An Abnormal Event that is the Cause of an Emergency Condition. (e.g., Loss of Cooling Water)
3. EMERGENCY—An Interruption From Normal Operation in Which Personnel or Equipment are Endangered or Equipment is Subject to Overpressure.
4. EXPLOSIVE LIMITS—Limits of the Flammable Range. Minimum and Maximum Concentrations of Flammable Vapor in Air.
5. FLASH POINT—Lowest Temperature at Which Liquid Exposed to Air Gives Off Sufficient Vapor to Form a Flammable Mixture.
6. HIGH FLASH STOCKS—Flash Points 130°F or Greater.
7. LOW FLASH STOCKS—Flash Points Less Than 130°F or Stocks at Temperatures Above or Within 15°F of its Flash Point.

DEFINITIONS (Continued)

8. LIGHT ENDS—Material Having an RVP ≥ 15 psia. (Reid Vapor Pressure, i.e., Vapor Pressure @ 100°F)
9. PYROPHORIC MATERIAL—A Material that is Spontaneously Combustible When Exposed to Air at Ambient Temperature.
10. TOXIC MATERIAL—A Material Capable of Causing Injury on Reaching Sites in or on the Human Body. (i.e., Phenol, H2S, HF Acid, Benzene, NH3, etc.)
11. SINGLE RISK—The Equipment Affected by a Single Contingency. (e.g., Fire)
12. FIRE ZONE—Area Containing the Smallest Group of Equipment That Can be Approached From All Sides by Fire-Fighting Equipment and Personnel. Regardless of Accessibility, Vessels With a Horizontal Distance of 20 Feet of Each Other are in the Same Fire Zone. Maximum Area Normally Limited to 5,000 Sq. Ft.

DEFINITIONS (Continued)

13. SET PRESSURE—The Inlet Pressure at Which the Pressure Relief Valve Is Set to Open.
14. MAWP—The Highest Pressure to Which a Vessel May be Subjected Continuously.
15. DESIGN PRESSURE—That Used as a Basis for Determining Minimum Shell Thickness, Usually the Same as MAWP (MAWP $\geq$ Design Pressure).
16. ACCUMULATION—The Pressure Increase Over MAWP During Discharge Through a Pressure-Relief Valve.
17. OVERPRESSURE—The Pressure Increase Over Set Pressure During Discharge Through a Pressure Relief Valve.
18. BACKPRESSURE—The Pressure on the Discharge Side of a Pressure-Relief Valve.

DEFINITIONS (Continued)

19. SUPERIMPOSED BACKPRESSURE—The Pressure on the Discharge Side of a Pressure-Relief Valve Before it Opens.
20. BUILT-UP BACKPRESSURE—Increase in Pressure at Valve Discharge Resulting From Flow Through That Valve.
21. DIFFERENTIAL SPRING PRESSURE—Set Pressure Minus the Superimposed Backpressure for a Conventional Valve. For a Balanced Bellows Valve, the Spring Pressure Equals the Set Pressure.

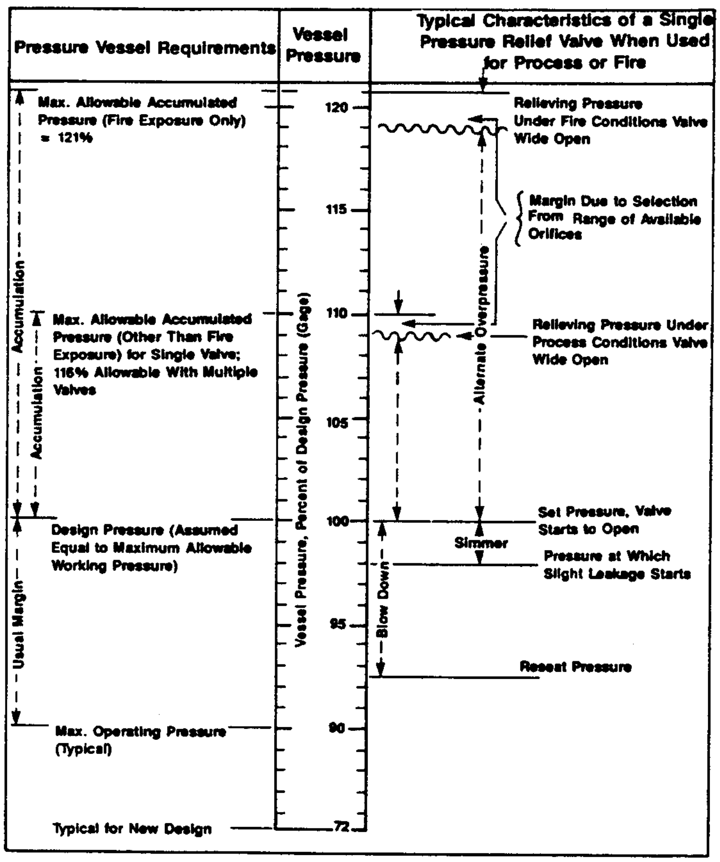
SUMMARY OF ALLOWABLE PRESSURES

	ΔP Spring	Max ΔP Builtup	Max Back Pres.	Superimposed Backpressure Maximum Allowable	Superimposed Backpressure Charac-teristics
• Conventional SV's	Set Press. Minus SIBP	10%	—	25%	Fixed
• Bellows	Set Press.	—	50% Normal 75% Retrofit	75%	Variable

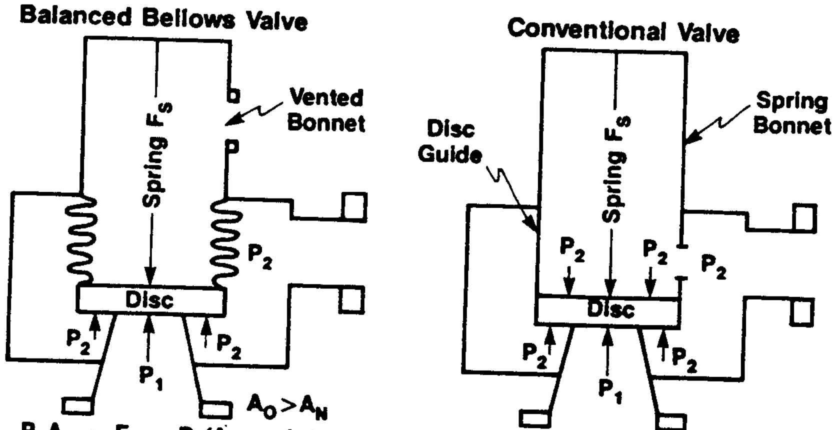
These Limits Will Be Reviewed Again in Later Slides

10.07

DISCHARGE CHARACTERISTICS OF TYPICAL PRESSURE RELIEF VALVE



FORCES ACTING ON DISCS OF CONVENTIONAL AND BALANCED BELLOWS SAFETY RELIEF VALVES



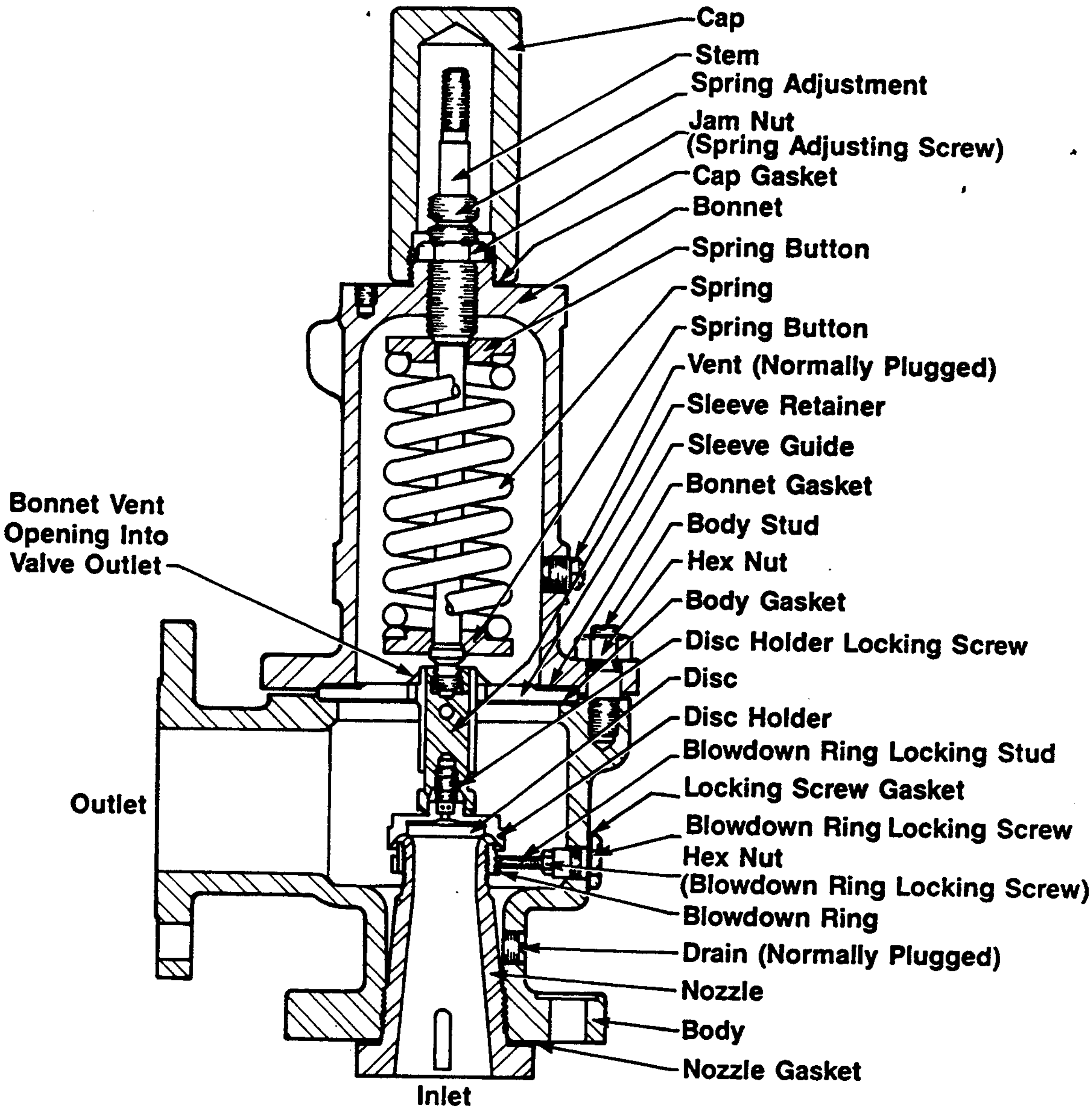
$P_1A_N = F_s - P_2(A_D - A_N)$

Back Pressure Has Very Little Effect on Set Pressure

$P_1A_N = F_S + P_2A_N$

Back Pressure Increases Set Pressure

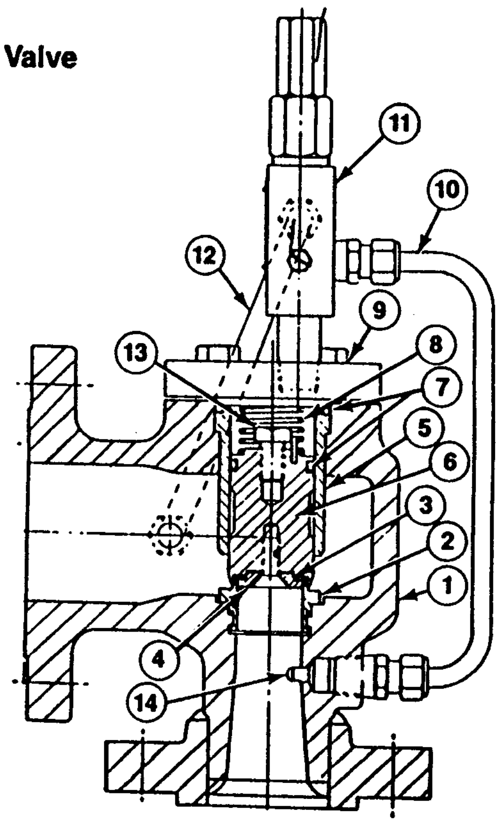
TYPICAL CONVENTIONAL PRESSURE RELIEF VALVE



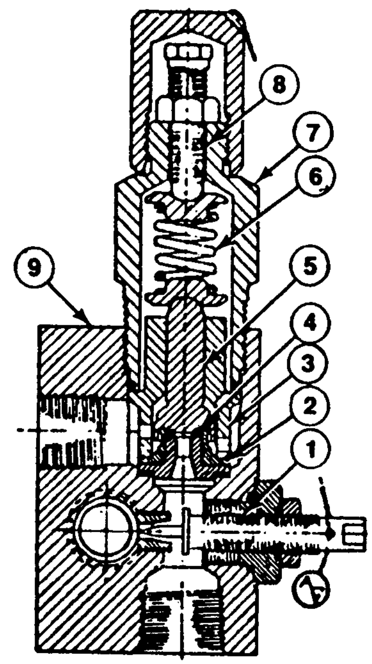
TYPICAL PILOT-OPERATED PR VALVE

Main Valve

No.	Part Name
1	Body
2	Nozzle
3	Seat
4	Seat Retainer
5	Liner
6	Piston
7	Piston Seal
8	Shipping Spring
9	Cap
10	Supply Tube
11	Pilot Valve
12	Exhaust Tube
13	Lift Adjustment Screw
14	Dipper Tube



Pilot Valve

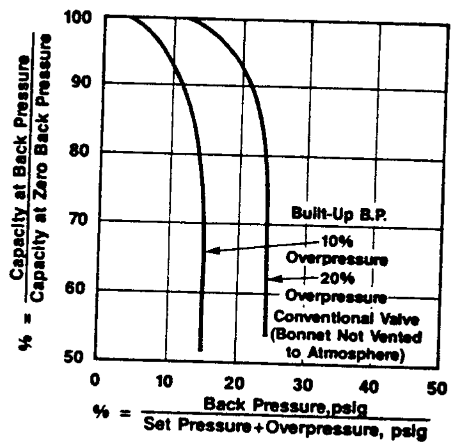


No.	Part Name
1	Blowdown Adjustment
2	Two Piece Nozzle
3	Spindle Guide
4	O-Ring Seat
5	Spindle
6	Spring
7	Bonnet
8	Adjusting Screw
9	Body

- The Pilot Valve Vents Pressure From the Top of the Disc in the Main Valve

10.11

TYPICAL EFFECTS OF VARIABLE BACK PRESSURE ON CAPACITY OF CONVENTIONAL PRESSURE RELIEF VALVES



10.12

DESIGN GUIDELINES

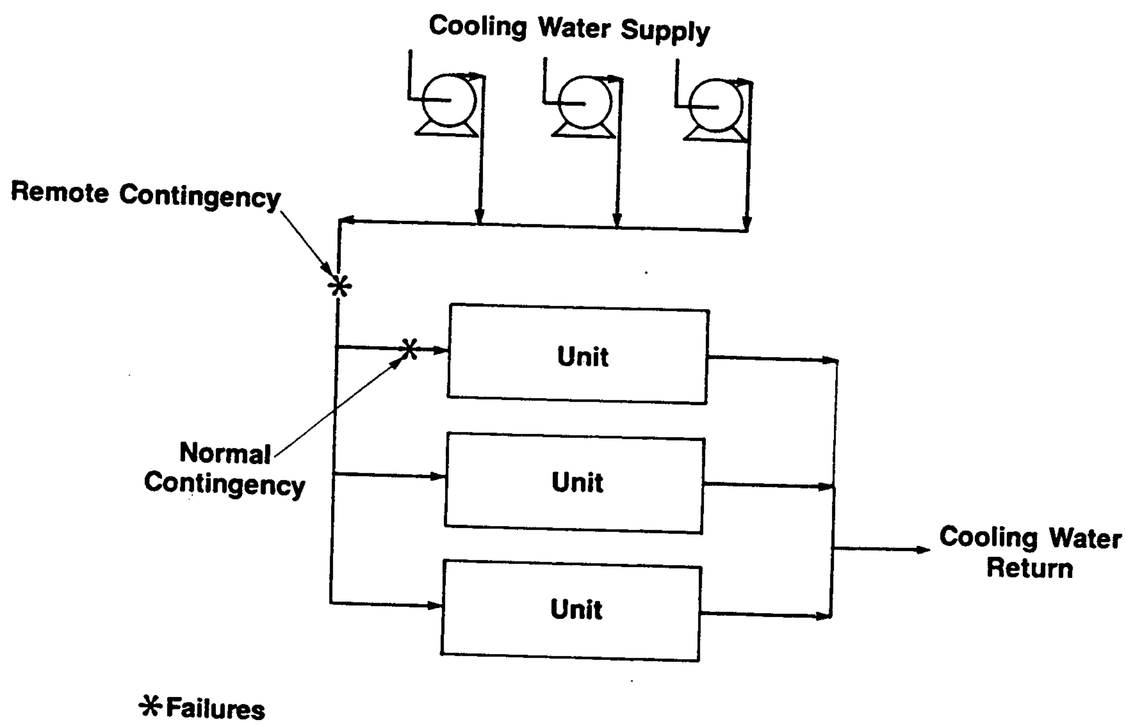
- Consider Only One Contingency at a Time. An Assumption that Two UNRELATED Contingencies Will Occur Simultaneously is Not Warranted.
- Assume That Immediately Prior to an Emergency, the Plant was Operating in a Normal Operating Condition.
- All Equipment Normally in Operation Will Continue to Function if it is Not Directly Part of the Contingency.
- Blowdown Valves and Pressure Control Valves Normally Closed Should Not be Assumed to be Operable in an Emergency and Credit Should Not be Taken for Their Capcity When Determining Required Relieving Capacity.
- Valves Which are Normally Open and Which are Not Directly Part of the Contingency are Assumed to Remain Open. The Credit Contributed by the Valve May be Considered.

10.13

TYPES OF CONTINGENCIES
(What Can Go Wrong?)

- Fire
  - Pressure in Equipment Enveloped in a Fire is Increased Through Vaporization of Liquid
- Utility Failures
  - Cooling Water–Failure to Remove Heat Causes Pressure to Increase
  - Electric Power–Generally Results in Loss of Heat Removal
  - Steam–Similar to Power
  - Air–Instruments Fail; Control Valves Fail According to Mode Specified (AO = Air to Open . . . Same as FC = Fails Closed)
- Mechanical Failure
  - Control Valves Can Fail in ANY Position; Pump Seizing, Tube Failures (Heat Exchangers, Furnaces); Plugged Lines; Plugged Reactors
- Operating Failure (Error)
  - Opening and Closing of Wrong Valves

10.14

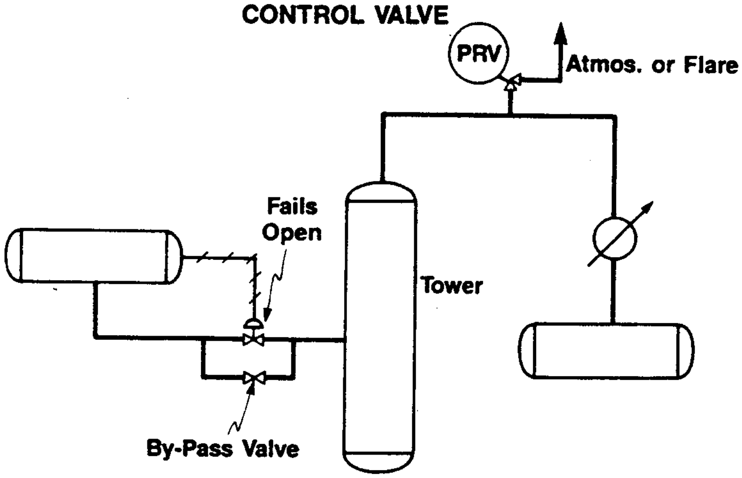


PLANT COOLING WATER FAILURES
FOR PRV SIZING

10.15

EMERGENCY CONDITIONS
EXAMPLE OF EQUIPMENT FAILURE

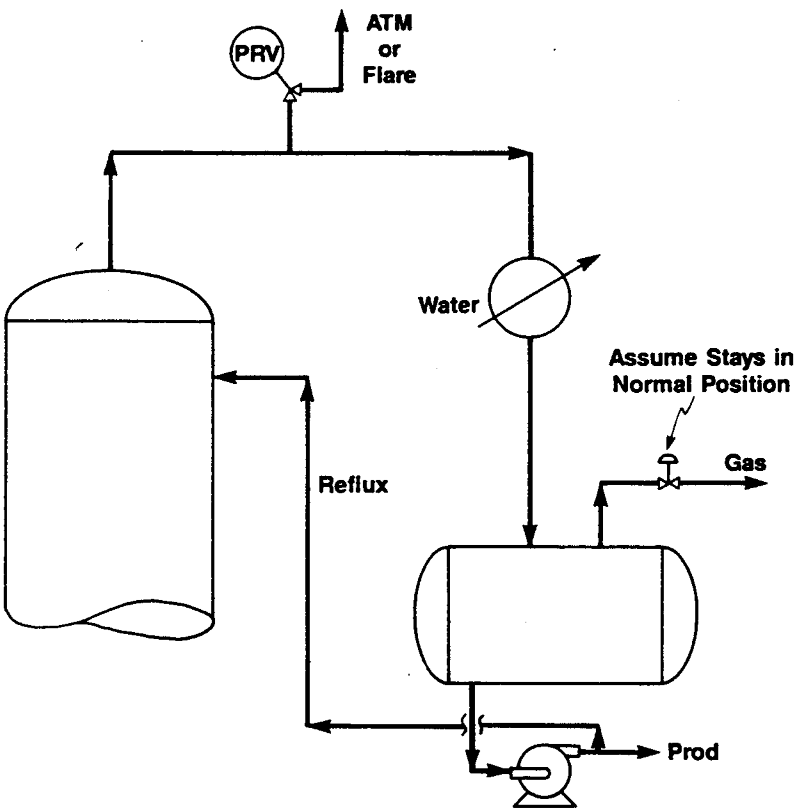
CONTROL VALVE



1. Assume By-Pass 1/2 Open—Limit Tower Pressue to 110% of Design Press. (Normal Contingency)
2. Assume By-Pass Full Open—Limit Tower Overpressure to 150% of Design Press. (Remote Contingency)

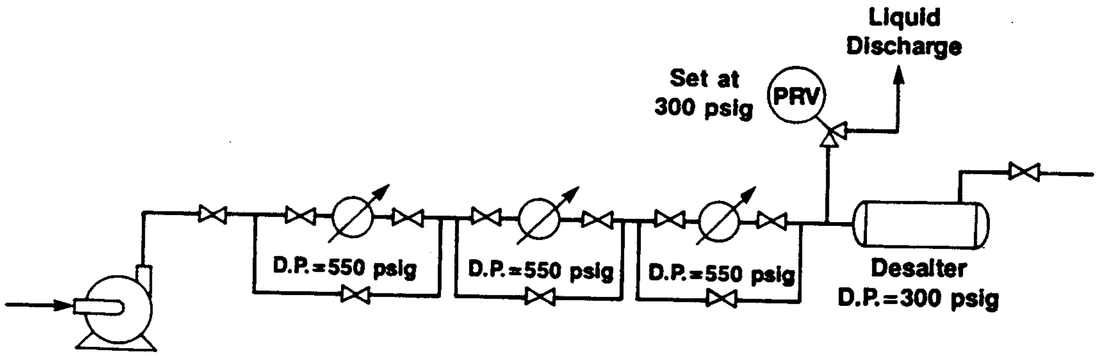
10.16

EMERGENCY CONDITIONS
EXAMPLE OF UTILITY FAILURE (C.W.)



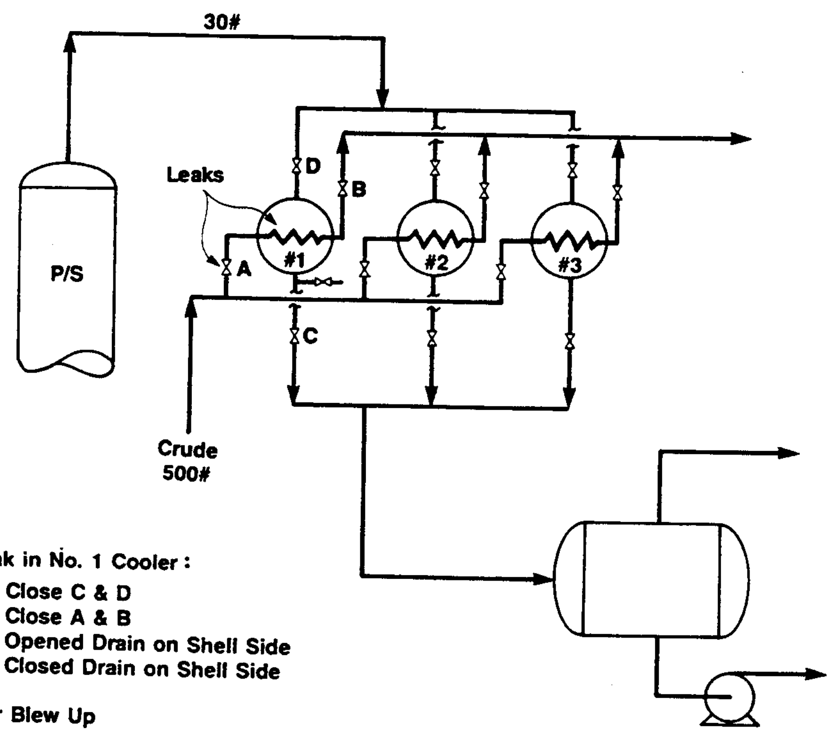
Overhead Condenser Water Failure

EMERGENCY CONDITIONS
EXAMPLE OF OPERATING FAILURE



Pump Shutoff ΔP = 500 psi
Max. Suction Pressure = 50 psig
∴ Max. Discharge Pressure = 550 psig

EXPLOSION CAUSED BY OPERATING ERROR
(CLOSING THE DRAIN ON A BLOCKED-IN, UNBLINDED EXCHANGER)



Crude Leak in No. 1 Cooler:
Operation Close C & D
Operation Close A & B
Operation Opened Drain on Shell Side
Operation Closed Drain on Shell Side

Exchanger Blew Up

SOME REMOTE CONTINGENCIES PERMITTING 150% ACCUMULATION

1. **Split Exchanger Tube**

   Low Pressure Side of Heat Exchanger Must be Protected by a PRV if:

   - Design Pressure on High Side is More Than 1.5 Times the Lower Design Pressure, and
   - The Piping on the Low Side Cannot Handle the Discharge From a Split Tube Without the Pressure Exceeding 1.5 Times The Lower Design Pressure. (See Addendum 10.01 for Calculation Procedure.)

2. Maloperation of a Car Sealed Open/Closed Valve

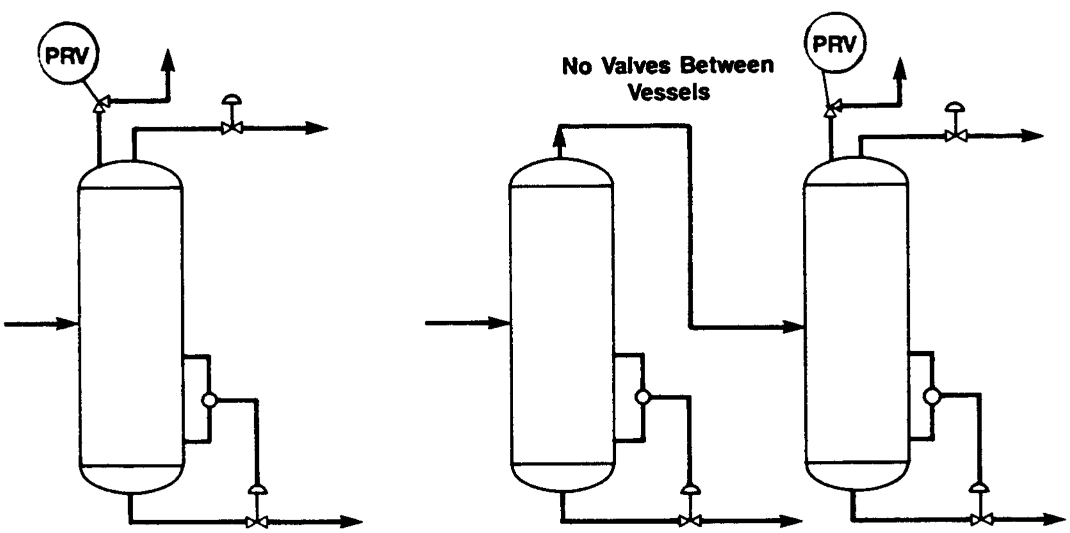
3. Fixed Bed Reactor "Plugging"

4. "Double" Contingencies With Some Remote Interrelationship (e.g. Control Valve Fails Open With Bypass 100% Open)

10.20

REQUIREMENTS FOR PRESSURE RELIEF VALVES

- Vessels
  - + Any Vessel That Can Be Overpressured Must Be Protected By a PRV



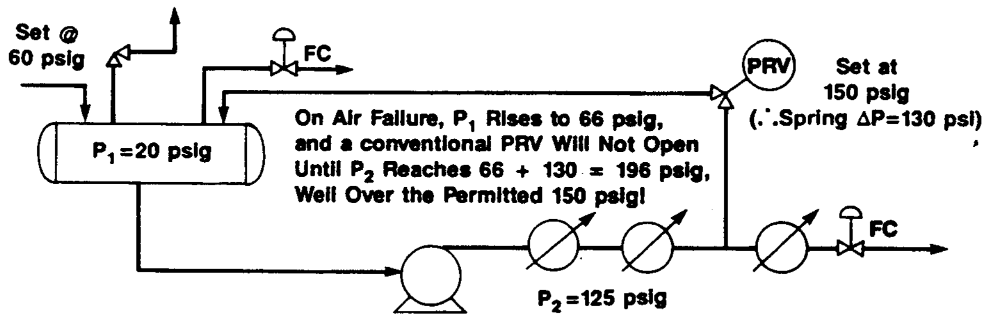
+ Vessels 2 Ft in Diameter and Smaller Made of Pipe Do Not Require Protection Against Fire

10.21

REQUIREMENTS FOR PRV'S (Continued)

- **Pumps**

  PRV's are Required on Discharge When Downstream Facilities Can be Overpressured. Typically it is Not Permitted to Discharge the PRV Back to the Suction Source of the Pump, as Demonstrated Here, But May be the Solution for Debottlenecking.

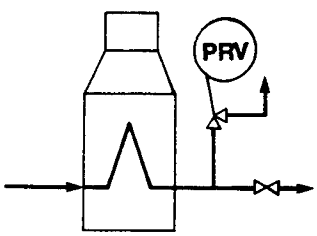


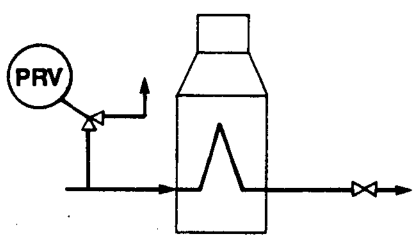
- **A Bellows Type PRV May be Acceptable in this Situation.**

REQUIREMENTS FOR PRV'S (Continued)

- **Furnaces**

  If There is a Restriction or Valve in the Outlet Line, PRV's are Required.
  If the Outlet Valve May be Car Sealed Open (CSO), a PRV MIGHT be Avoided.
  However, an outlet block valve is not normally necessary.





— Normally Preferred
— Required if Feed is All Vapor (To Provide Continued Flow)

— Only When Feed Contains Liquid, and:
- PRV Would Coke on Outlet
- PRV Cannot be Purged

SPECIAL CONTINGENCIES

- Presence of Highly Volatile Materials (e.g. Water in Feed)
  - — Difficult to Apply PR Valve Protection
  - — Design/Operation Should Minimize Likelihood
- Chemical Reactions
  - — Temperature "Runaway"
  - — Decomposition Reactions
- Thermal Expansion
  - — Piping "Blocked In" and Heated by Steam Tracing, Solar Radiation
  - — Heat Exchanger "Blocked In" on Cold Side With Flow Continuing on the Hot Side

CALCULATING PRV SIZES

- VAPOR RELEASE

The Maximum Flow Thru a Restriction Such as the Nozzle of a Pressure Relief Valve Will Occur When the Velocity Thru the Nozzle Equals the Speed of Sound or "Sonic Velocity". This Condition is Known as "Choked" or "Critical" Flow. In This Situation, the Flow Rate is Independent of Downstream Pressure.

$$\frac{Px}{P_1} = \left[\frac{2}{K+1}\right]^{\frac{K}{K-1}}$$

Px = psia at Sonic Velocity
P_1 = Safety Valve Inlet Pres, psia (Set + Overpressure)
K = Cp/Cv at Inlet Conditions

If Downstream Pressure is Less than Px, Critical Flow Will Be Achieved

$$\frac{Px}{P_1} = \text{Approx. } 0.5$$

PRV SIZING (Continued)

At Critical Pressure Drop Across the Pressure Relief Valve: (This Equation Must NOT be Used for Non-Critical Flow)

$$\frac{1}{A} = \frac{520 \cdot Kd \cdot Kb \cdot P_1}{W} \cdot \sqrt{\frac{M \cdot K}{\mu \cdot T_1} \cdot \left[\frac{2}{K+1}\right]^{\frac{K+1}{K-1}}}$$

A = Safety Valve Orifice Diameter, Sq In
W = Lbs/Hr
T_1 = Inlet Temperature, deg R*
P_1 = Safety Valve Inlet Pressure, psia (Set + Overpressure)
M = Molecular Weight
μ = Compressibility Factor
Kd = Orifice Discharge Coefficient (Usually 0.975)
Kb = Correction Factor For Backpressure (1.0 for Conventional Valves)
K = Cp/Cv; If Unknown, Use 1.001 (Conservative)

*For Fire, T1 = Mid Boiling Point at P1

PRV SIZING (Continued)

If a PRV Must be Designed to Relieve Under Non-Critical Flow Conditions, the Following May be Used.

$$A = \frac{W\sqrt{\mu T}}{735\, F_2\, K_d\, K_b \sqrt{M P_1 (P_1 - P_2)}}$$

Where

$$F_2 = \sqrt{\left(\frac{K}{K-1}\right)\left(\frac{P_2}{P_1}\right)^{\frac{2}{K}} \frac{\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{K-1}{K}}\right]}{\left[1-\left(\frac{P_2}{P_1}\right)\right]}}$$

and P_2 = Pressure at PRV Outlet (Superimposed + Builtup) psia

PRV SIZING (Continued)

- **LIQUID RELEASE—There Is No Critical Flow Pressure for Liquids, Unless Vaporization Occurs.**
  Equation for Old Style PRV:

$$\frac{1}{A} = \frac{38 \cdot Kd \cdot Kp \cdot Ku \cdot Kw}{L} \cdot \sqrt{\frac{1.25 \cdot P_1 - P_2}{S}}$$

L = GPM at Inlet Conditions
S = S.G. at Inlet Temperature
P_1 = Set Pressure + Accumulation, psig
P_2 = Total Back Pressure, psig
Kd = Discharge Coefficient (Typical = 0.64)
Kp = Correction For Accumulation Less Than 25%; Kp = 0.6 for 10%
Ku = Viscosity Correction Factor
Kw = Backpressure Correction for Balanced Bellows Valve
A = Area, In^2

PRV SIZING (Continued)

- **LIQUID RELEASE—There Is No Critical Flow Pressure for Liquids, Unless Vaporization Occurs.**
  Equation for New Style PRV: NO CHAN [illegible] VAPOR

$$\frac{1}{A} = \frac{38 \cdot Kd \cdot Ku \cdot Kw}{L} \cdot \sqrt{\frac{P_1 - P_2}{S}}$$

L = GPM at Inlet Conditions
S = S.G. at Inlet Temperature
P_1 = Set Pressure + Accumulation, psig
P_2 = Total Back Pressure, psig
Kd = Discharge Coefficient (Typical = 0.64)
Ku = Viscosity Correction Factor
Kw = Backpressure Correction for Balanced Bellows Valve
A = Area, In^2

PRV SIZING (Continued)

- Release of Liquid/Vapor Mixtures
  - — Complex Calculations–Will Not be Covered.
  - — A Suggested Method Within Exxon is Given in Report EE.28E.90
- PRV Type—Normally Use Conventional Valve, Unless:
  - — Superimposed Back Pressure is Not Constant
  - — Built-Up Back Pressure Exceeds 10% of Set Pressure
  - — Fluid Service is Fouling or Corrosive

    These Situations Require Considering Balanced Bellows Valve...But Don't Specify Unless Absolutely Necessary...Bellows Valves Have Inherent Mechanical Limitations

10.30

PRV "CHATTERING"

- Can Occur in Both Liquid and Vapor Services
- Causes:
  - — Valve Orifice Area is Too Large
    - + 25% Minimum of PRV Design Flow Capacity is Necessary to Avoid "Chattering"
    - + Consider Multiple PRV's With Staggered Set Pressures to Meet Total Area Requirements
  - — Excessive Inlet Pressure Drop
    - + Maximum Pressure Drop Must be Limited to 3% Set Pressure by Appropriate Lines
  - — Excessive Built-Up Back Pressure
    - + Maximum Built-Up Pressure Must be Limited to 10% Set Pressure by Appropriate Line Size

10.31

DESIGN OF PRV INLET PIPING

- PRESSURE DROP
  - Less Than 3% of Set Pressure (psig)
- SIZING
  - At Least Size of PR Valve Inlet
  - For Multiple PR Valves, Cross-Sectional Area of Manifold Line Equal to Sum of All Inlets
- ORIENTATION
  - Must Drain Freely Back to Source of Fluid (No Traps)
- PREVENTION OF PLUGGING
  - Heat Tracing if Plugging by Ice or Wax
  - For Coking, Provide Continuous Purge of Clean Fluid
- REMOVAL OF PR VALVE DURING OPERATION
  - Install Bleeder Between Inlet Block Valve and PR Valve

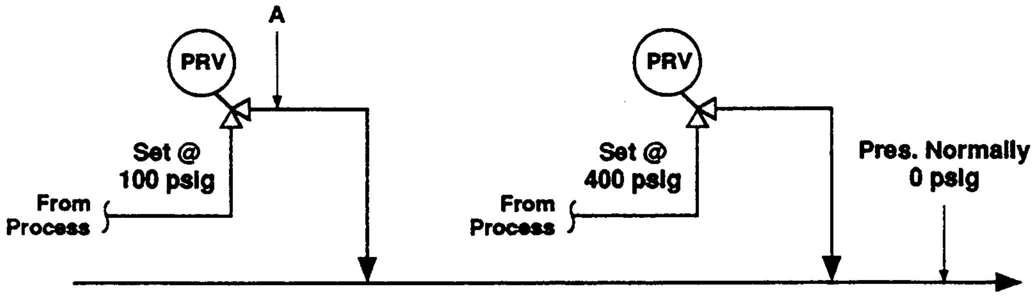
10.32

BACK PRESSURE

- For Single or Multiple PRV Releases Discharging Under a Single-Risk Contingency:
  - Built-Up Back Pressure is Limited to:
    - + 10% of Set Pressure for Conventional Type PR Valves for Operating Contingencies
    - + 20% of Set Pressure for Conventional Type PR Valves for Fire Contingencies
  - Total Back Pressure (Built-Up + Superimposed) is Limited to 50% of Set Pressure for Balanced Bellows Type PR Valves for Either Operating or Fire Contingencies
- Superimposed Back Pressure on the Non-Discharging PR Valves in the System During a Maximum System Release (Single Contingency) Shall Not Exceed:
  - 25% of Lowest Set Pressure for Conventional Type PR Valves
  - 75% of Lowest Set Pressure for Balanced Bellows Type PR Valves
- Mechanical Design of the PR Valve Shall Take Into Account Any Limitations Imposed by the Back Pressure

10.33

PRV DISCHARGE PIPING

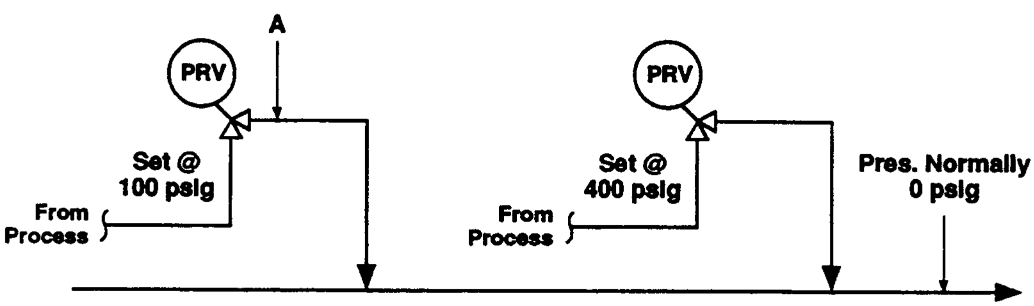


Both Valves are Relieving

- — For Conventional Safety Valve, Max pres @ A = 10 psig
- — High Outlet Line Pressure Drop Will Cause Valve "Chatter"
- — For Balanced Bellows Valve, Max Pres @ A = 50 psig
- — Discharge Line Size Should Not Be Less Than Safety Valve Outlet Flange Size
- — Velocity in Discharge Piping Should Not Exceed 75% of Sonic
- — Discharge Piping Should Not Contain Any Liquid Traps and Should Slope Downwards to the Collection Header

10.34

OTHER BACK PRESSURE CONSIDERATIONS IN DISCHARGE CIRCUIT



PR Valve at A is Not Relieving

- — For Conventional Safety Valve, Max pres @ A = 25 psig
- — For Balanced Bellows Valve, Max Pres @ A = 75 psig

10.35

DESIGN OF PR VALVE OUTLET PIPING
ATMOSPHERIC DISCHARGE

- Discharge Riser
  - 10 Ft. Above Top Platform
  - 50 Ft. Horizontal Distance From Other Equipment
  - Discharge Vertically
  - No Restrictions: Check Valves, Flame Arrestors or Orifice Plates
  - Maximum Velocity = 75% of Sonic
  - Minimum Velocity = 100 Ft/Sec. (If Flammable)
- Install Snuffing Steam if Above Autoignition
- Install Toroidal Ring if Hydrogen or Methane

10.36

MULTIPLE PRV INSTALLATIONS

- Considerations
  - Use When Release is Greater Than Largest Available PRV
  - Use to Get Better Match of Total Area Requirement and Avoid Potential "Chattering"
  - Using Two Small Valves May be More Economic Than One Large Valve (>8×10), Because of Mechanical Design Considerations
- ASME Code Requirements for Set Pressures With Multiple Valves
  - Only ONE PRV Needs to be Set at Vessel Design Pressure
  - Other Valves Can be Set Up to 105% of Vessel Design Pressure
  - All PRV's Can be Sized at Design Pressure + 16% Accumulation (105% × 110% = 116%)
  - If Fire is Limiting Contingency, Accumulation Can be 121%

10.37

OPEN AND CLOSED EMERGENCY RELEASE SYSTEMS

Open System — The Discharge Piping of Pressure Relief Valves That Release Directly to Atmosphere
— (Regulations in Some Areas May Require a Rupture Disc Upstream of Atmospheric PRV's)

Closed System — The Discharge Piping for Pressure Relief Valves that Go to a Collection System Such as a Blowdown Drum or Flare

Open vs Closed System:
- — Liquid Release – Always Discharge to a Closed System
- — Vapor – Several Factors to Consider:
  - + Corrosivity of Vapors That Condense
  - + Toxicity
  - + Pollution Abatement
  - + Ignition of a Vapor Cloud – Check Radiant Heat Density
  - + Possibility of Liquid Fallout From Condensing Vapors

10.38

SELECTION OF ATMOSPHERE OR CLOSED DISCHARGE FOR PR VALVES

PRV's Which Must Discharge to a Closed System:

- — Liquid or Partial Liquid Discharges
- — Vapor Service PR Valves Which in Single Contingency Could Discharge Flammable or Toxic Liquids
- — PR Valves on Drums With Liquid Hold-Up Less Than 15 Minutes
- — PR Valves Handling Toxic or Corrosive Vapors Which Condense (e.g., Phenol)
- — PR Valves in Toxic Service Where Calculated Concentrations at Working Area Exceed Threshold Limit Value
- — Releases of Vapor to Atmosphere if Ignited Would Result in Heat Densities Exceeding 6000 $BTU/HR/FT^2$

10.39

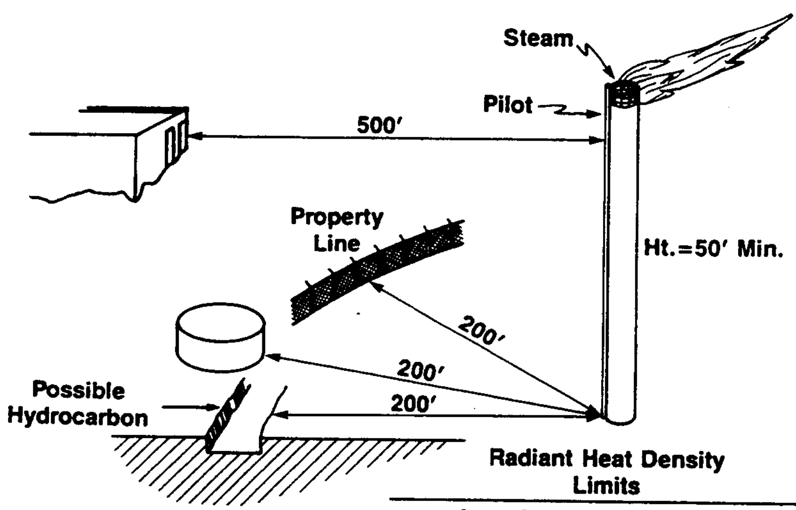
CONDENSIBLE BLOWDOWN DRUMS AND FLARE STACKS

Discharge to a Condensible Blowdown Drum if:

- Liquid Would Condense in the Flare Seal Drum and Flare Stack if Sent to a Flare System
- Reduction in Flare System Size is Necessary
- Vapor is Heavy Enough to Cause Smoke Problems When Flared
- Necessary to Reduce Maximum Temperature of Flared Gas and Minimize Thermal Expansion Problems in the Flare Stack

10.40

LIMITS ON RADIANT HEAT DENSITY



Radiant Heat Density Limits

Location	BTU/Hr/Sq. Ft.
Property Line	500
At Grade Below Flare	3,000
Equipment	10,000

10.41

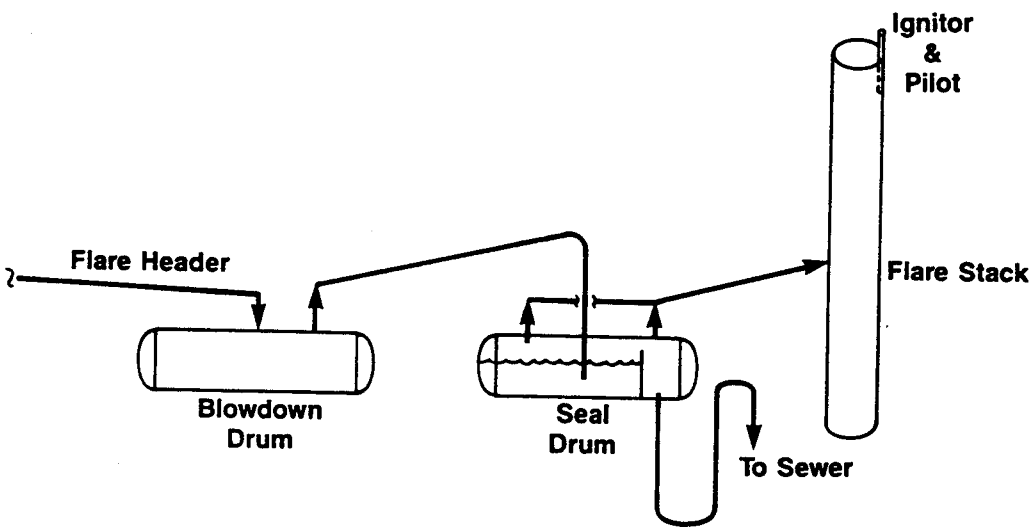
DESIGN OF CLOSED SYSTEMS

Routing of Closed Systems

- To a Conventional Flare System Via Blowdown Drum
- To Atmosphere Via Condensible Blowdown Drum
- To Segregated H_2S Flaring System
  - Use Any for H_2S Venting, Not PR Valves in H_2S Service (H_2S System Can Become Plugged)
- To Special Segregated Closed Systems
  - Severely Corrosive Materials
  - High Value Materials
  - Vapors Which React to From Solids

10.42

TYPICAL COMPONENTS OF A VAPOR DISCHARGE SYSTEM



10.43

DESIGN OF CLOSED SYSTEMS

- Routing
  - — Main Lines Outside Process Areas
  - — Avoid High Risk Areas
  - — Provide Isolating CSO Valves
- Liquid Drainage
  - — PR Outlet Piping Slope to Header—No Traps
  - — Slope Header to BD Drum
  - — Between BD and Flare Seal–Slope to BD (0.2% Slope)
  - — Steam Trace for Wax, Ice or Viscous Liquids
- Thermal Expansion
  - — Only Pipe Loops in Onsite Areas
  - — Sliding Type Expansion Joints Acceptable in Offsite (Low Risk) Areas
- Design Temperature
  - — Use Max Design Temp of Protected Vessel. However Piping Must be Designed for Thermal Expansion Caused by the Fire Contingency Release Temperature
- Maximum Line Velocity = 75% of Sonic

10.44

DESIGN OF CLOSED SYSTEMS

Sizing of Flare System

- Consider All Releases
  - — PR Valves
  - — Knockout Drum and Closed Drainage
  - — Vapors From Water Disengaging Drums
  - — Vapor and Liquid Pulldowns
- Maximum Flow From Largest Single Contingency
- Component Pressure Drops
  - — PR Valve Laterals and Headers
  - — Blowdown Drum
  - — Flare Seal Drum (Dipleg Submergence)
  - — Flare Stack
  - — Flare Tip (Typically 1 psi)

10.45

FIRE CONTINGENCY

- All Facilities in a Fire Zone are Assumed to be Affected by the Fire
- Assume Pneumatic Tubing and Wiring Destroyed
- Relieving Capacity

$W = \frac{Q}{L}$

W = pounds/hour
Q = heat Input/Btu/hour
L = latent heat of vaporization, Btu/pound

$Q = 21{,}000\ F\ (A)^{0.82}$

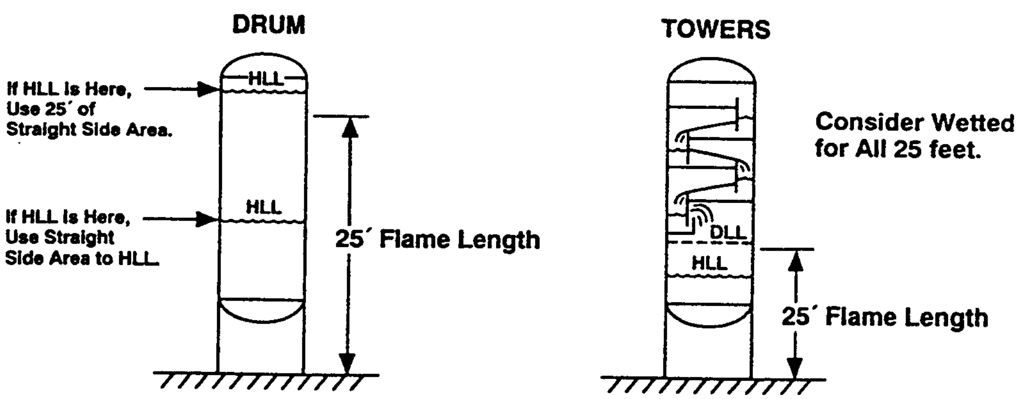
F = f (insulation)
= 1.0 for bare wall
A = wetted area, ft^2

- L is Elevated at Set Pressure + Accumulation
- Accumulation for Fire is 21%

10.46

CALCULATION OF WETTED AREA

Wetted Area is Total Wetted Surface Within 25 Feet of Grade or Elevation at Which Fire Could be Sustained.



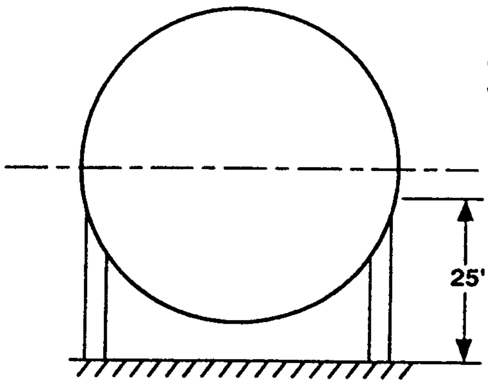
HLL = High Liquid Level

DLL = Dumped Liquid Level

10.47

WETTED AREA (CONTINUED)

SPHERES AND SPHEROIDS



"A" Based on Total Exposed Surface to a Height of 25 Feet or to the Elevation of the Equator, whichever is Greater.

Addendum 10.01

SECTION 10
SAFETY

— INDEX —

CALCULATION OF FLOW RATE FROM A SPLIT TUBE

Calculate split-tube flow as follows:

$W = 1.566 \times A \sqrt{\rho_2(P_1 - P_2)}$ (This equation contains a discharge coefficient of 0.65)

Where: W = Flow in klb/h

A = Split-tube flow area $= 2 \cdot \dfrac{\pi \cdot (\text{Tube ID})^2}{4}$, in².
(two cross-sections)

P_1 = High-pressure-side pressure, psia.

P_2 = Low-pressure-side pressure, psia.

<u>For Vapors</u>, P_2 depends on critical flow pressure, P_x.

1. If P_2 is less than or equal to P_x, use P_x as P_2.
   (All units in psia.)

2. If P_2 is greater than P_x, use P_2 as P_2.
   (All units in psia.)

3. $P_x = P_1 \left(\dfrac{2}{k+1}\right)^{k/(k-1)}$

ρ_2 = Fluid density at the vena contracta in pounds/cubic foot.
If the fluid is a liquid at the high pressure, use the liquid density. If the fluid is a vapor at the high pressure, the density is determined as follows:

$$\rho_2 = \rho_1 \left(\frac{P_2}{P_1}\right)^{1/k}, \text{ where } K = \frac{C_p}{C_v}$$

$\rho_1 = \dfrac{P_1 M}{10.731 ZT}$ = density high-pressure side, pounds/cubic feet.

where: M = molecular weight.
Z = compressibility factor.
T = temperature at high-pressure side, in °R.

Table 10.01

SPECIFIC HEAT RATIOS OF VARIOUS GASES AT 60°F AND 1.0 ATM.

Gas	Specific Heat Ratio $K = c_p/c_v$	Critical Flow Pressure Ratio P_x/P_1
Methane	1.31	0.54
Ethane	1.19	0.57
Propane	1.13	0.58
Butane	1.09	0.59
Air	1.40	0.53
Ammonia	1.31	0.53
Benzene	1.12	0.58
Carbon dioxide	1.29	0.55
Hydrogen	1.41	0.52
Hydrogen sulfide	1.32	0.53
Phenol	1.30*	0.54*
Steam	1.33	0.54
Sulfur dioxide	1.29	0.55
Toluene	1.09	0.59

*Estimated

Figure 10.01
CRITCIAL FLOW PRESSURE FOR HYDROCARBONS

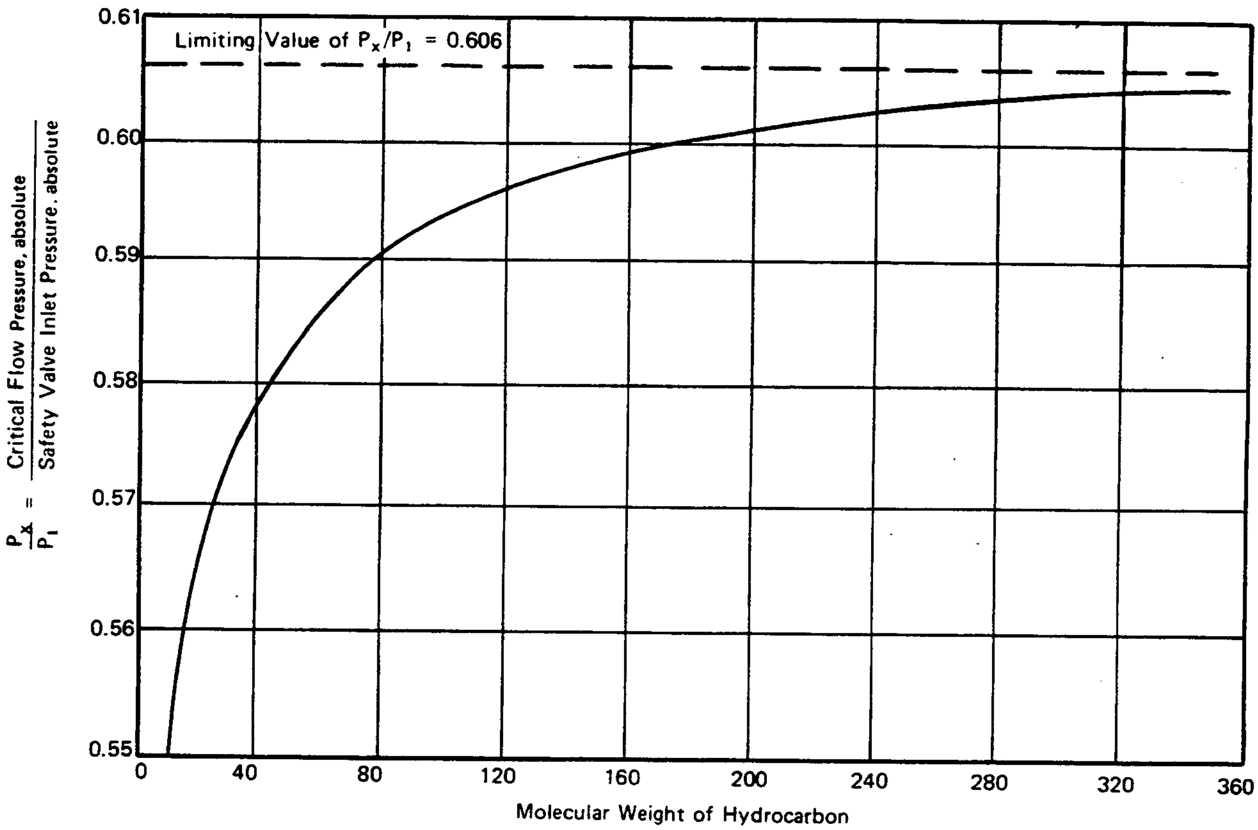


Figure 10.02

VARIABLE OR CONSTANT BACK PRESSURE SIZING FACTOR, K_b, FOR BALANCED BELLOWS PRESSURE RELIEF VALVES (VAPORS AND GASES)

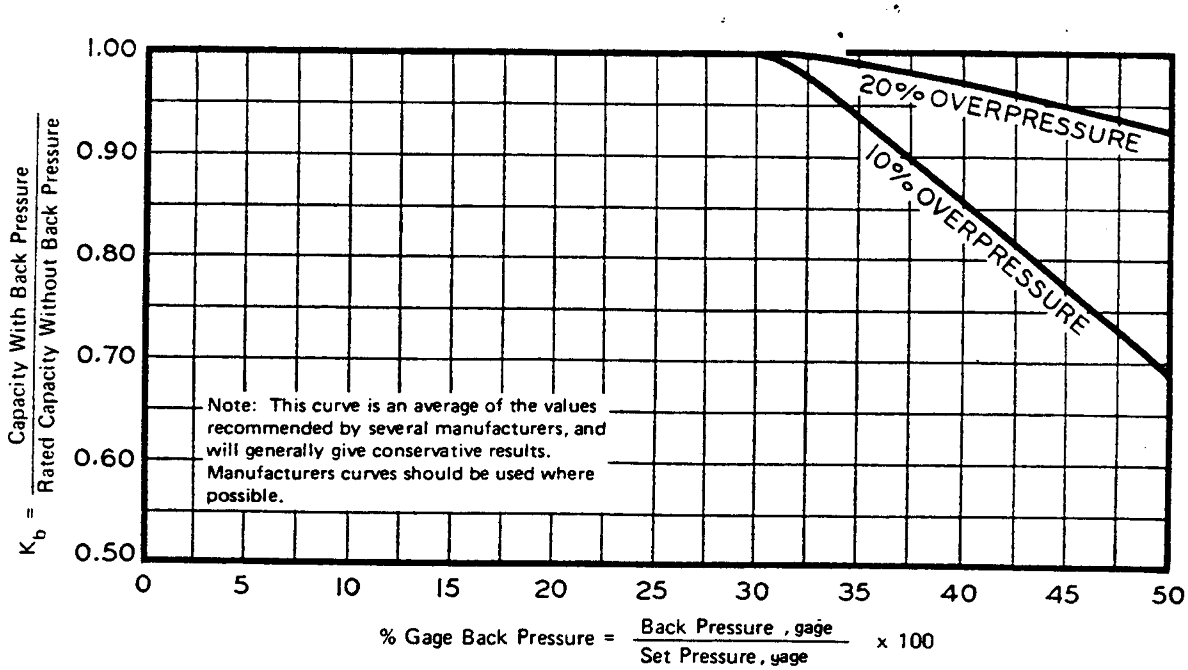
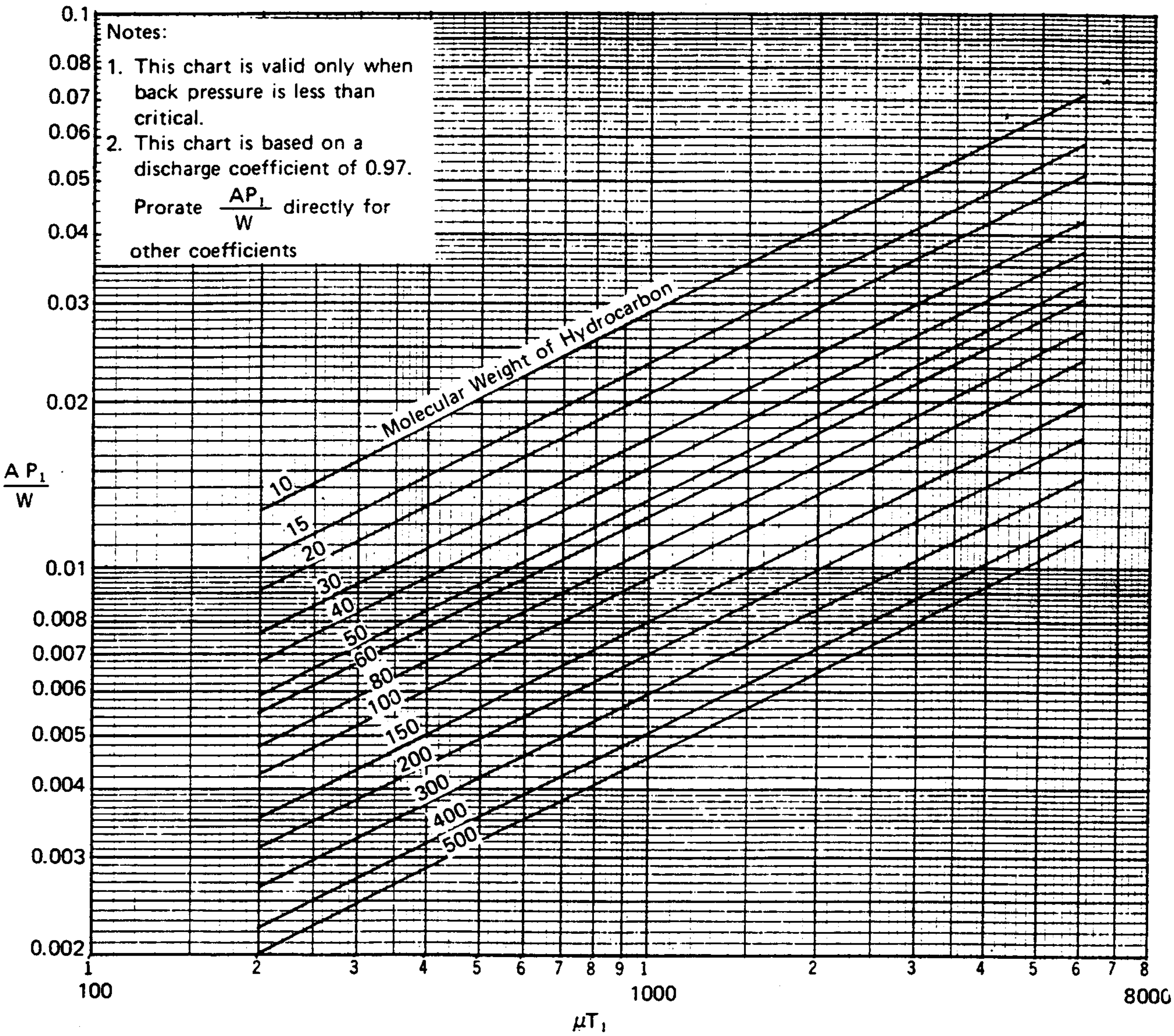


Figure 10.03

PRESSURE RELIEF VALVE ORIFICE AREA REQUIRED FOR HYDROCARBON VAPOR RELEASE



A = Orifice area, in^2

P_1 = Inlet press., psia (set press. + overpressure)

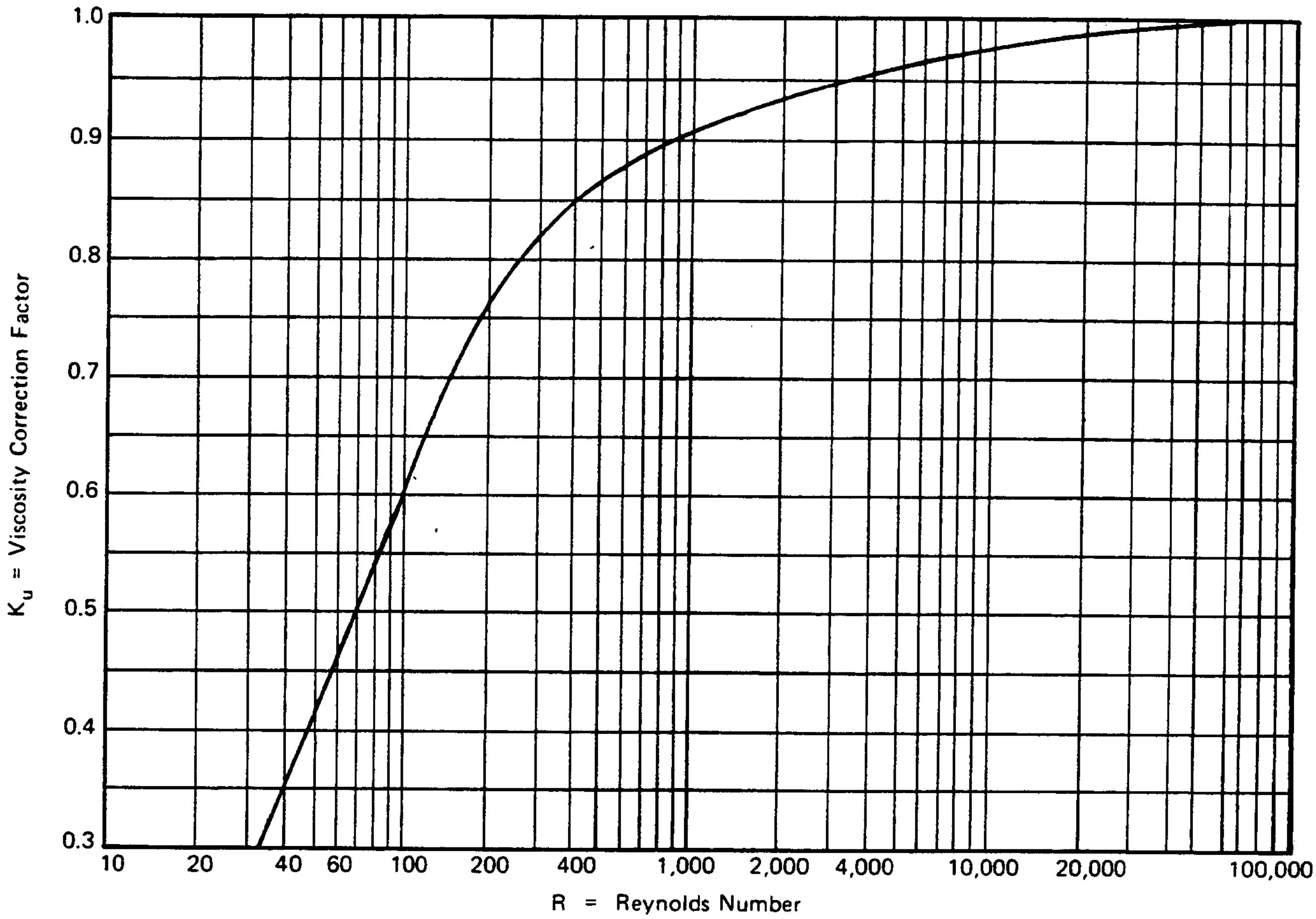
W = Rate of flow, lb/h

T_1 = Inlet temp., °Rankine

μ = Compressibility factor

Figure 10.04

VISCOSITY CORRECTION CHART
PR VALVES IN LIQUID SERVICE



To size a relief valve for viscous liquid service:

1. Determine area required without viscosity correction, A_o (K_u = 1), from Equation (8).
2. Select next larger standard orifice size from manufacturer's literature or Table III-2.
3. Determine Reynold's Number, R:

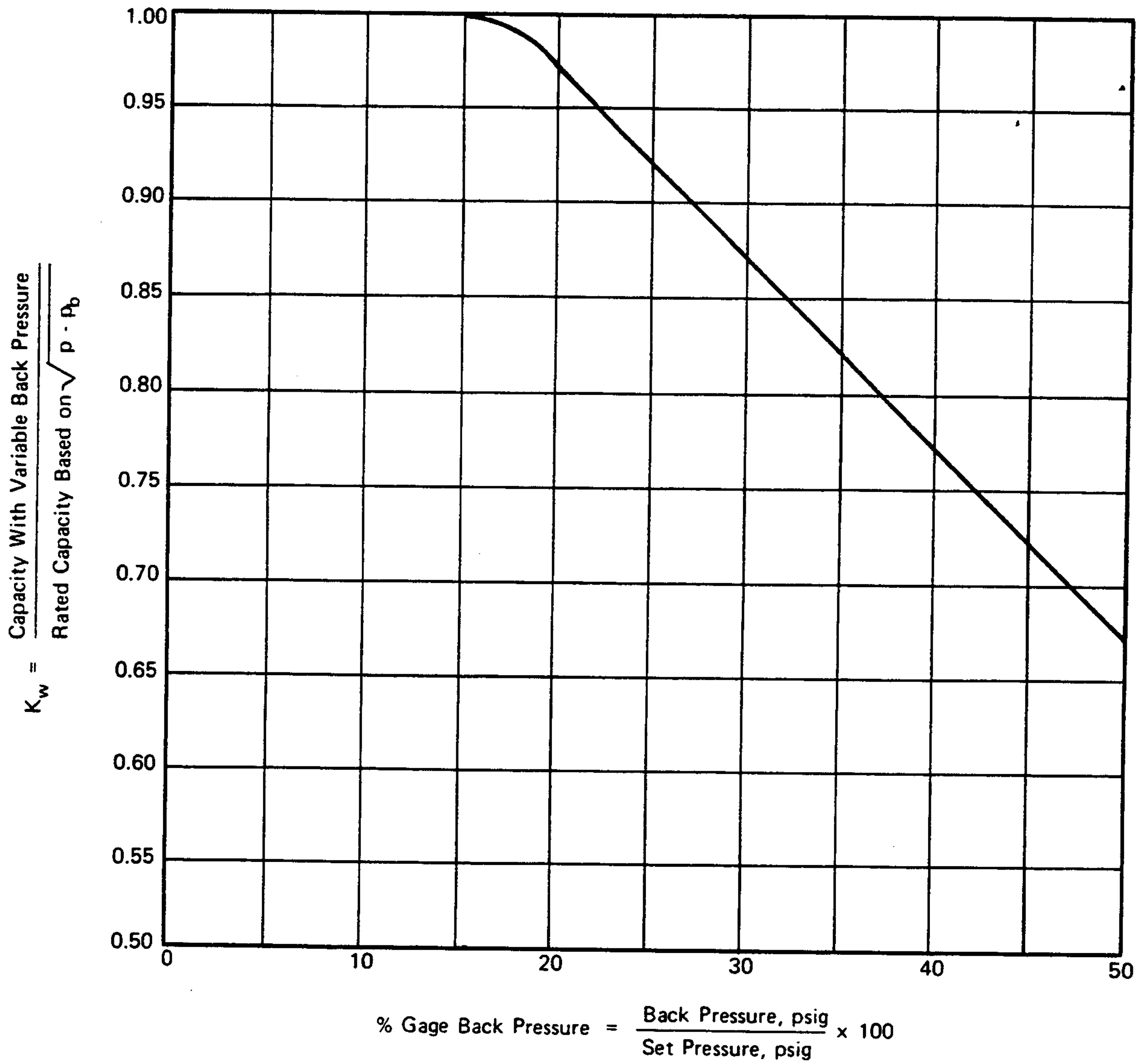
$$R = \frac{2800\ LS}{\mu\sqrt{A}} = \frac{12700\ L}{U\sqrt{A}}$$

where:
- L = Flow Rate, gpm
- S = Specific gravity at flowing temperature vs. water at 70°F.
- μ = Viscosity at flowing temperature, centipoises
- U = Viscosity at flowing temperature, SSU
- A = Effective orifice area, in^2 (from manufacturer's literature)

4. Find K_u, viscosity correction factor from chart.
5. Corrected area required is A_o/K_u ; if this exceeds A, repeat the calculation.
6. If the required corrected area is only slightly above a standard orifice size, consider using multiple smaller valves with staggered set pressures, to minimize tendency to chatter.

Figure 10.05

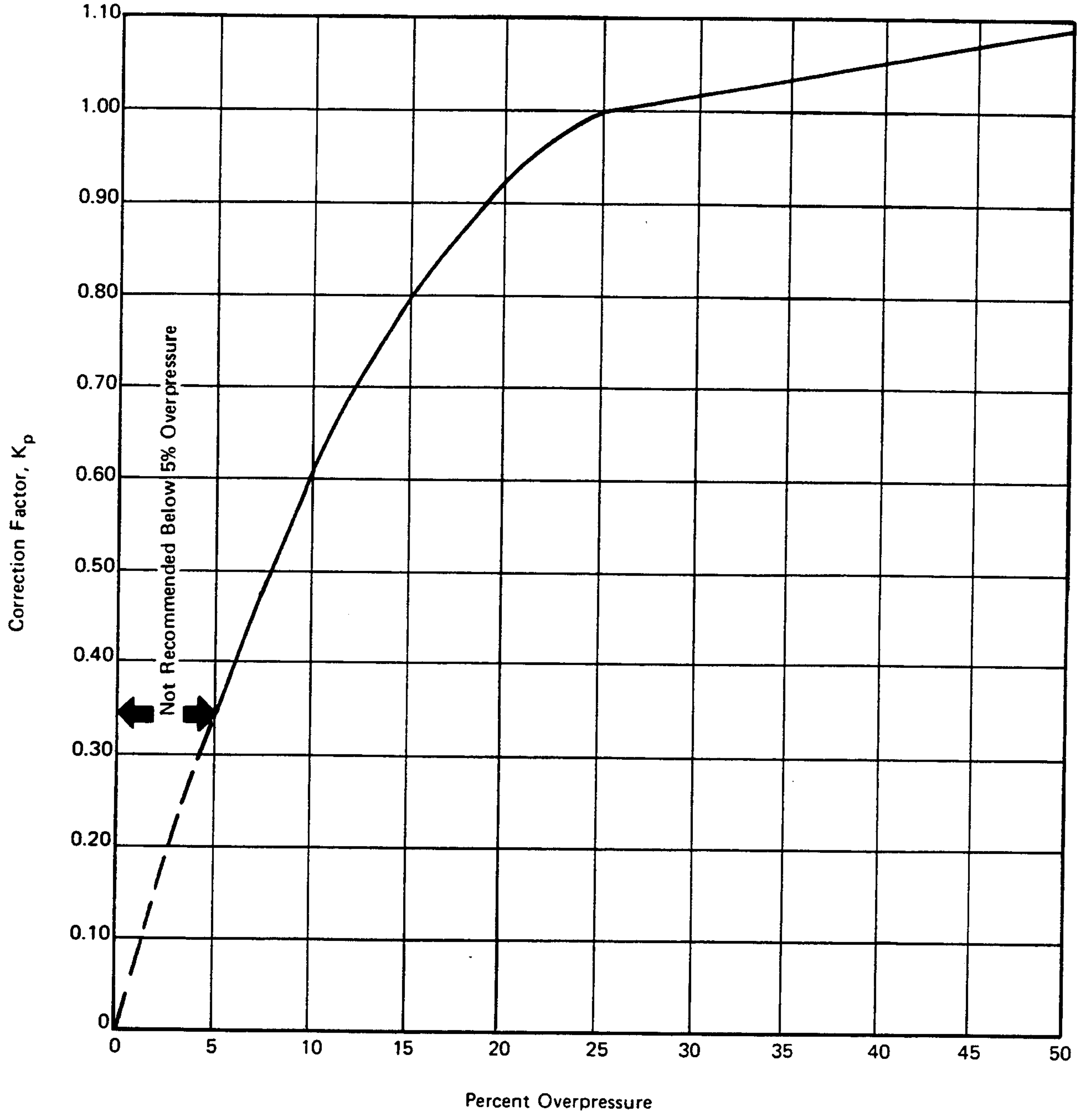
VARIABLE OR CONSTANT BACK PRESSURE SIZING FACTOR, K_w, FOR 25% OVERPRESSURE ON BALANCED BELLOWS PRESSURE RELIEF VALVES (LIQUIDS ONLY)



Note: The above curve represents a compromise of the values recommended by a number of relief valve manufacturers. This curve may be used when the make of the valve is not known. When the make is known, the manufacturer should be consulted for the correction factor.

Figure 10.06

OVERPRESSURE CORRECTION FACTOR FOR PR VALVES IN LIQUID SERVICE



Note: The above curve shows that up to and including 25 percent overpressure, capacity is affected by the change in lift, the change in orifice discharge coefficient, and the change in overpressure. Above 25 percent, the valve is at full lift and capacity is affected only by overpressure.

PR Valve Number ______________ Unit ______________

Service ______________

Vapor Flow Rate, klb/h = ______________

Vapor Flow Rate, lb/s = ______________ = W

Set Pressure, psig, P = ______________

Molecular Weight, M = ______________

INLET LINE

Allowable(1) $\Delta P = 0.03P$		______ psi
$P_1 = P(1 + \%\ \text{Overpress}/100) + 14.7 =$		______ psia
Temp=(+460), T_1	=	______ °R
Compress. Factor, μ	=	______
$\rho_1 = \dfrac{MP_1}{\mu RT_1} =$ ______	=	______ lb/ft³
$\dfrac{W^2}{\rho_1} = \dfrac{(\text{lb/s})^2}{\text{lb/ft}^3} =$ ______	=	______
Inlet Line Size,		______ in
ΔP per 100 ft,	=	______ psi
Equiv. Length, (assume 100 ft. min.)	=	______ ft
Piping Press. Drop	=	______ psi
Contraction Press. Drop(2) $\Delta P_f = \dfrac{K}{144}\left(\dfrac{\rho_1 v^2}{2gc}\right)$	=	______ psi
Total S.V. Inlet ΔP	=	______ psi

OUTLET LINE

Allowable Total Back Pressure,(3) P_a	=	______ psig
Superimp. Back Press, P_s	=	______ psig
Bellows $P_b = P_a - P_s$	=	______ psig
Conventional $P_b = 0.1\ P$	=	______ psig
Temp=(+460), T_2	=	______ °R
Compress. Factor, μ	=	______
$\rho_s = \dfrac{M(P_s + 14.7)}{\mu RT_2} =$ ______	=	______ lb/ft³
$\dfrac{W^2}{\rho_s} = \dfrac{(\text{lb/s})^2}{\text{lb/ft}^3} =$ ______	=	______
Outlet Line Size,		______ in
ΔP per 100 ft,	=	______ psi
Equiv. Length,	=	______ ft
S.V. Outlet Line ΔP	=	______ psi
Spec. Heat Ratio, k	=	______
Sonic Velocity: $68\sqrt{\dfrac{k(P_s + 14.7)}{\rho_s}}$	=	______ ft/sec
Outlet line Velocity: $\dfrac{W}{\rho_s \cdot A}$	=	______ ft/sec
Percent of Sonic(4)	=	______ %

(1) 1.5 psi for PRV set below 50 psig.

(2) Friction only. ΔP due to acceleration not to be included.

(3) Do not exceed Px. Check that bellows mechanical strength is not exceeded (correct for temperature above 400°F).

(4) Maximum allowed 75%.

ER&E GRAPHIC DESIGN 0790-81-022

Table 10.02

CROSBY AND FARRIS STEEL FULL NOZZLE RELIEF VALVES(1)

Orifice Size and Area in.2	Standard Connections ANSI Flanges(2)				Maximum Back Pressure, psig at 100 F(4)		450 F Maximum Service Temperature Carbon Steel Spring and Body				800 F Maximum Service Temperature Alloy Steel Spring, Carbon Steel Body				1000 F Maximum Service Temperature Alloy Steel Spring and Body			
							Valve Type		Pressure Limits, psig, for Inlet Temperatures of		Valve Type		Pressure Limits, psig, for Inlet Temperatures of		Valve Type		Pressure Limits,(5) psig, for Inlet Temperatures of	
	Inlet		Outlet		Conventional	Bellows	Crosby	Farris	20 to 100 F	450 F	Crosby	Farris	450 F	800 F	Crosby	Farris	800 F	1000 F
D 0.110 and E 0.196	1	150	2	150	230	230	JOS-15	A-10	275	165	JOS-16	A-20	165	80	-	-	-	-
	1	300	2	150	230	230	JOS-25	A-11	275	275	JOS-26	A-21	275	275	-	-	-	-
	1	300	2	150	230	230	JOS-35	A-12	720	615	JOS-36	A-22	615	365	JOS-37	A-32	410	215
	1	600	2	150	230	230	JOS-45	A-13	1440	1235	JOS-46	A-23	1235	730	JOS-47	A-33	815	430
	1½	900	2	300	600	-	JO-55-9	A-14	2160	1845	JO-56-9	A-24	1845	1100	JO-57-9	A-34	1225	645
	1½	1500	2	300	600	-	JO-55	A-15	3600	3080	JO-56	A-25	3080	1830	JO-57	A-35	2040	1070
	1½	2500	2½	300	600	-	JO-65	A-16	6000	5135	JO-66	A-26	5135	3050	JO-67	A-36	3400	1785
F 0.307	1½	150	2	150	230	230	JOS-15	A-10	275	165	JOS-16	A-20	165	80	-	-	-	-
	1½	300	2	150	230	230	JOS-25	A-11	275	275	JOS-26	A-21	275	275	-	-	-	-
	1½	300	2	150	230	230	JOS-35	A-12	720	615	JOS-36	A-22	615	365	JOS-37	A-32	410	215
	1½	600	2	150	230	230	JOS-45	A-13	1440	1235	JOS-46	A-23	1235	730	JOS-47	A-33	815	430
	1½	900	2½	300	600	500	JO-55-9	A-14	2160	1845	JO-56-9	A-24	1845	1100	JO-57-9	A-34	1225	645
	1½	1500	2½	300	600	500	JO-55	A-15	3600	3080	JO-56	A-25	3080	1630	JO-57	A-35	2040	1070
	1½	2500	2½	300	600	500	JO-65	A-16	5000	5000	JO-66	A-26	5000	3050	JO-67	A-36	3400	1785
G 0.503	1½	150	2½	150	230	230	JOS-15	A-10	275	165	JOS-16	A-20	165	80	-	-	-	-
	1½	300	2½	150	230	230	JOS-25	A-11	275	275	JOS-26	A-21	275	275	-	-	-	-
	1½	300	2½	150	230	230	JOS-35	A-12	720	615	JOS-36	A-22	615	365	JOS-37	A-32	410	215
	1½	600	2½	150	230	230	JOS-45	A-13	1440	1235	JOS-46	A-23	1235	730	JOS-47	A-33	815	430
	1½	900	2½	300	600	470	JO-55-9	A-14	2160	1845	JO-56-9	A-24	1845	1100	JO-57-9	A-34	1225	645
	2	1500	3	300	600	470	JO-65	A-15	3600	3080	JO-66-15	A-25	3080	1830	JO-67-15	A-35	2040	1070
	2	2500	3	300	600	470	JO-75	A-16	3600	3600	JO-66	A-26	3600	3050	JO-67	A-36	3400	1785
H 0.785	1½	150	3	150	230	230	JOS-15	A-10	275	165	JOS-16	A-20	165	80	-	-	-	-
	1½	300	3	150	230	230	JOS-25	A-11	275	275	JOS-26	A-21	275	275	-	-	-	
	2	300	3	150	230	230	JOS-35	A-12	720	615	JOS-36	A-22	615	365	JOS-37	A-32	410	215
	2	600	3	150	230	230	JOS-45	A-13	1440	1235	JOS-46	A-23	1235	730	JOS-47	A-33	815	430
	2	900	3	150	230	230	JO-55-9	A-14	2160	1845	JO-56-9	A-24	1845	1100	JO-57-9	A-34	1225	645
	2	1500	3	300	600	415	JO-75	A-15	2750	2750	JO-66	A-25	2750	1830	JO-67	A-35	2040	1070
J 1.287	2	150	3	150	230	230	JOS-15	A-10	275	165	JOS-16	A-20	165	80	-	-	-	-
	2	300	3	150	230	230	JOS-25	A-11	275	275	JOS-26	A-21	275	275	-	-	-	-
	2½	300	4	150	230	230	JOS-35	A-12	720	615	JOS-36	A-22	615	365	JOS-37	A-32	410	215
	2½	600	4	150	230	230	JOS-45	A-13	1440	1235	JOS-46	A-23	1235	730	JOS-47	A-33	815	430
	3	900	4	150	230	230	JO-65	A-13	2160	1845	JO-66-9	A-24	1845	1100	JO-57	A-34	1225	645
	3	1500	4	300	600	230	JO-75	A-15	2700	2700	JO-66	A-25	2700	1830	JO-67	A-35	2040	1070
K 1.838	3	150	4	150	230	150	JOS-15	A-10	275	165	JOS-16	A-20	165	80	-	-		
	3	300	4	150	230	150	JOS-25	A-11	275	275	JOS-26	A-21	275	275	-	-		
	3	300	4	150	230	150	JOS-35	A-12	720	615	JOS-36	A-22	615	365	JOS-37	A-32	410	215
	3	600	4	150	230	200	JOS-45	A-13	1440	1235	JOS-46	A-23	1235	730	JOS-47	A-33	815	430
	3	900	6	150	230	200	JO-65	A-14	2160	1845	JO-66-9	A-24	1845	1100	JO-57	A-34	1225	645
	3	1500	6	300	600	200	JO-75	A-15	2160	2160	JO-66	A-25	2160	1830	JO-67	A-35	2040	1070
L 2.853	3	150	4	150	230	100	JOS-15	A-10	275	165	JOS-16	A-20	165	80	-	-	-	-
	3	300	4	150	230	100	JOS-25	A-11	275	275	JOS-26	A-21	275	275	-	-	-	-
	4	300	6	150	230	170	JOS-35	A-12	720	615	JOS-36	A-22	615	365	JOS-37	A-32	410	215
	4	600	6	150	230	170	JOS-45	A-13	1000	1000	JOS-46	A-23	1000	730	JOS-47	A-33	815	430
	4	900	6	150	230	170	JO-55	A-14	1500	1500	JO-66	A-24	1500	1100	JO-57	A-34	1225	645
	4	1500	6	150	230	170					JO-76	A-25	1500	1500	JO-67	A-35	1500	1070

Table 10.02 (Cont'd)

CROSBY AND FARRIS STEEL FULL NOZZLE RELIEF VALVES(1)

Orifice Size and Area in.^{2}	Standard Connections ANSI Flanges(2)		Maximum Back Pressure, psig at 100F (4)(5)		450°F Maximum Service Temperature Carbon Steel Spring and Body				800°F Maximum Service Temperature Alloy Steel Spring, Carbon Steel Body				1000°F Maxin .m Service Temperature Alloy Steel Spring and Body			
					Valve Type		Pressure Limits,(5) psig, for Inlet Temperatures of		Valve Type		Pressure Limits,(5) psig, for Inlet Temperatures of		Valve Type		Pressure Limits,(5) psig, for Inlet Temperatures of	
	Inlet	Outlet	Conven-tional	Bellows	Crosby	Farris	-20 to 100°F	450°F	Crosby	Farris	450°F	800°F	Crosby	Farris	800°F	1000°F
	4 - 150	6 - 150	230	80	JOS-15	A-10	275	165	JOS-16	A-20	165	80	—	—	—	—
	•4 - 300	6 - 150	230	80	JOS-25	A-11	275	275	JOS-26	A-21	275	275	—	—	—	—
M 3.60	•4 - 300	6 - 150	230	160	JOS-35	A-12	720	615	JOS-36	A-22	615	365	JOS-37	A-32	410	215
	4 - 600	6 - 150	230	160	JOS-46	A-13	1100	1100	JOS-46	A-23	1100	730	JOS-47	A-33	815	430
	4 - 900	6 - 150	230	160	—	—	—	—	JO-56	A-24	1100	1100	JO-57	A-34	1100	645
	4 - 150	6 - 150	230	80	JOS-15	A-10	275	165	JOS-16	A-20	165	80	—	—	—	—
	•4 - 300	6 - 150	230	80	JOS-25	A-11	275	275	JOS-26	A-21	275	275	—	—	—	—
N 4.34	•4 - 300	6 - 150	230	160	JOS-35	A-12	720	615	JOS-36	A-22	615	365	JOS-37	•A-32	410	215
	4 - 600	6 - 150	230	160	JOS-45	A-13	1000	1000	JOS-46	A-23	1000	730	JOS-47	A-33	815	430
	4 - 900	6 - 150	230	160	—	—	—	—	—	A-24	1000	1000	—	A-34	1000	645
	4 - 150	6 - 150	230	80	JOS-15	A-10	275	165	JOS-16	A-20	165	80	—	—	—	—
	•4 - 300	6 - 150	230	80	JOS-25	A-11	275	275	JOS-26	A-21	275	275	—	—	—	—
P 6.38	•4 - 300	6 - 150	230	150	JOS-35	A-12	525	525	JOS-36	A-22	525	365	JOS-37	A-32	410	215
	4 - 600	6 - 150	230	—	JOS-45	A-13	1000	1000	JOS-46	A-23	1000	730	JOS-47	A-33	815	430
	4 - 900	6 - 150	230	—	—	—	—	—	—	A-24	1000	1000	—	A-34	1000	645
	6 - 150	8 - 150	115	70	JOS-15	A-10	165	165	JOS-16	A-20	165	80	—	—	—	—
Q 11.05	•6 - 300	8 - 150	115	70	JOS-25	A-11	165	165	JOS-26	A-21	165	165	—	—	—	—
	•6 - 300	8 - 150	115	115	JOS-35	A-12	300	300	JOS-36	A-22	300	300	JOS-37	A-32	165	165
	6 - 600	8 - 150	115	115	JOS-45	A-13	600	600	JOS-46	A-23	600	600	JOS-47	A-33	600	430
	6 - 150	8 - 150	60	60	JOS-15	A-10	100	100	JOS-16	A-20	100	80	—	—	—	—
R 16.0	•6 - 300	8 - 150	60	60	JOS-25	A-11	100	100	JOS-26	A-21	100	100	JOS-37	A-32	100	100
	•6 - 300	10 - 150	100	100	JOS-35	A-12	230	230	JOS-36	A-22	230	230	—	—	—	—
	6 - 600	10 - 150	100	100	JOS-45	A-13	300	300	JOS-46	A-23	300	300	JOS-47	A-33	300	300
T 26.0	8 - 150	10 - 150	30	30	JOS-15	A-10	65	65	JOS-16	A-20	65	65	—	—	—	—
	•8 - 300	10 - 150	30	30	JOS-25	A-11	65	65	JOS-26	A-21	65	65	—	—	—	—
	•8 - 300	10 - 150	60	60	JOS-35	A-12	120	120	JOS-36	A-22	120	120	—	A-32	120	120
	•8 - 300	10 - 150	60	60	JOS-45	H-A-12	300	300	JOS-46	H-A-22	300	300	JOS-37	H-A-32	300	215

Notes:

(1) Crosby valve information for JOS series from catalog 310 1982; for JO series from catalog 315 1980
Farris valve information from catalog FE - 336.
Crosby style: JOS and JO type are conventional; B replaces O for bellows.
Farris style : A type is conventional; B replaces A for bellows.
C replaces A for conventional with "O" ring.
D replaces A for bellows type with "O" ring.

(2) Raised face is standard: ring joint is optional

•(3) The proper valve must be selected on the basis of maximum temperature and pressure limitations. See the manufacturers' catalogs.

(4) For temperatures over 400°F, maximum back pressure allowable must be reduced per correction factors on p. 56.

(5) Pressure limits vary according to which manufactures is used. Lower valves are listed in this table.