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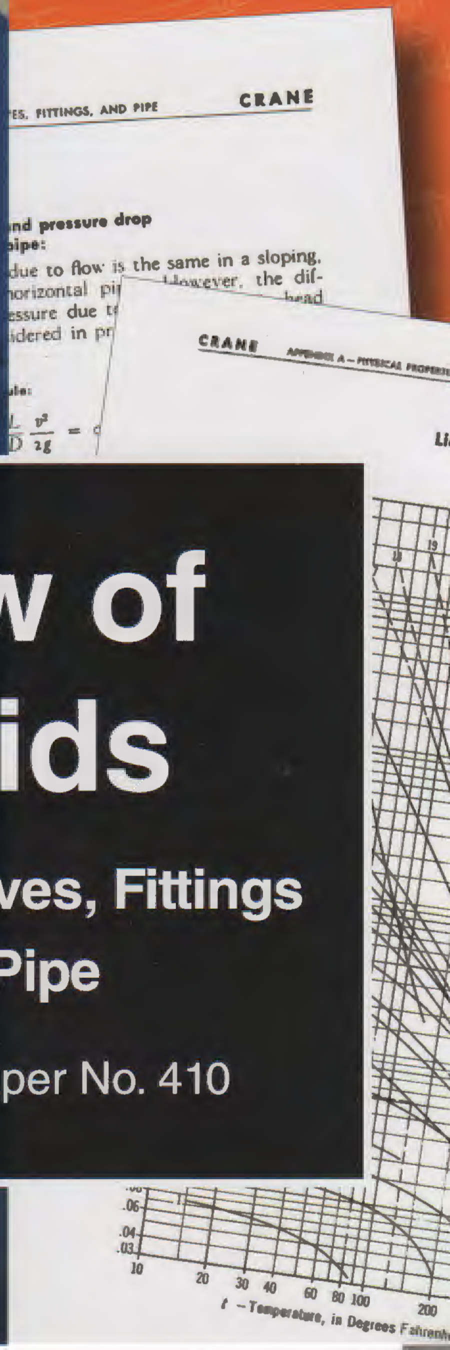

Flow of Fluids

Through Valves, Fittings and Pipe

Technical Paper No. 410

CRANE

CRANE®

FLOW OF FLUIDS

Through Valves, Fittings and Pipe

Technical Paper No. 410

By the Engineering Department

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Technical Paper No. 410

PRINTED IN U.S.A.

Reprinted 10/10

ISBN 1-40052-712-0

FH-CR-TB-EN-L13-00-1010

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Foreword

In the 21st century, the global industrial base continues to expand. Fluid handling is still at the heart of new, more complex processes and applications. In the 19th century, water was the only important fluid which was conveyed from one point to another in pipe. Today, almost every conceivable fluid is handled in pipe during its production, processing, transportation, or utilization. In the 1950's new fluids such as liquid metals i.e., sodium, potassium, and bismuth, as well as liquid oxygen, nitrogen, etc., were added to the list of more common fluids such as oil, water, gases, acids, and liquors that were being transported in pipe at the time. In the current decade of new technologies, heat-transfer fluids for solar plants, mineral slurries, and new chemical compounds expand the envelope of materials of construction, design, process pressures and temperature extremes as never before. Transporting fluids is not the only phase of hydraulics which warrants attention either. Hydraulic and pneumatic mechanisms are used extensively for the precise controls of modern aircraft, sea-going vessels, automotive equipment, machine tools, earth-moving and road-building machines, scientific laboratory equipment, and massive refineries where precise control of fluid flow is required for plant automation.

So extensive are the applications of hydraulic and fluid mechanics that most engineering disciplines have found it necessary to teach at least the elementary laws of fluid flow. To satisfy a demand for a simple and practical treatment of the subject of flow in pipe, Crane Co. in 1935, first published a booklet entitled Flow of Fluids and Heat Transmission. A revised edition on the subject of Flow of Fluids Through Valves, Fittings, and Pipe was published in 1942 as Technical Paper 409. In 1957, a completely new edition with an all-new format was introduced as Technical Paper No. 410. In T.P. 410, Crane endeavored to present the latest available information on flow of fluids, in summarized form with all auxiliary data necessary to the solution of all but the most unusual fluid flow problems.

The 1976 edition presented a conceptual change regarding the values of Equivalent Length L/D and Resistance Coefficient K for valves and fittings relative to the friction factor in pipes. This change had a relatively minor effect on most problems dealing with flow conditions that result in Reynolds numbers falling in the turbulent zone. However, for flow in the laminar zone, the change avoided a significant overstatement of pressure drop. Consistent with this conceptual revision, the resistance to flow through valves and fittings became expressed in terms of resistance coefficient K instead of equivalent length L/D, and the coverage of valve and fitting types was expanded. Further important revisions included updating of steam viscosity data, orifice coefficients, and nozzle coefficients. As in previous printings, nomographs were included for the use of those engineers who preferred graphical methods of solving some of the more simple problems.

In the 2009 edition of Technical Paper 410, Crane Co. has now included new flow control and measurement components to the pages of this paper. Pumps and Control Valves, critical elements of fluid handling, are included for the first time, as well as Flow Meters, and several additional types of valves and fittings. We have added new illustrations and updated the content throughout. Many of the nomographs have been replaced with online calculators. Visit www.flowoffluids.com for the latest data.

Originally, data on flow through valves and fittings were obtained by carefully conducted experiments in the Crane Engineering Laboratories. For this 2009 update, additional tests were performed within Crane to increase the number of valves with defined resistance coefficients. In addition, industry research was also gathered and refined to provide the reader with the latest methods for calculating hydraulic resistance. Resistance values for fittings were correlated with existing industry research and, when appropriate, more updated methods are provided in this paper, particularly seen with the new treatment of Tees and the addition of Wyes.

Since the last major update of TP-410, personal computers and Web applications have become the computational tools of choice. To meet the needs of today's engineers we have presented a variety of proven computational methods to simplify fluid flow calculations for those interested in developing custom spreadsheets or computer programs. In addition, Flow of Fluids has its own web site (www.flowoffluids.com) with a variety of Web based tools to simplify your most common fluid flow calculations.

The 2009 version of the Technical Paper 410 employs the most current references and specifications dealing with flow through valves, fittings, pipes, pumps, control valves and flow meters. The fluid property data found in Appendix A has been updated to reflect the current research on estimating fluid property data with references for the data cited throughout the paper.

From 1957 until the present, there have been numerous printings of Technical Paper No. 410. Each successive printing is updated, as necessary, to reflect the latest flow information available. This continual updating, we believe, serves the best interests of the users of this publication. The Flow of Fluids software and updated web site provide users with electronic tools and a source for the latest information. We welcome your input for improvement.

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Nomenclature

Unless otherwise stated, all symbols used in this book are defined as follows:

A = cross sectional area (ft^2)
a = cross sectional area (in^2)
bhp = brake (shaft) horsepower (hp)
C = flow coefficient for orifices and nozzles
C_d = discharge coefficient for orifices and nozzles
C_V = flow coefficient for valves or piping components
c = speed of sound in a fluid (ft/s)
c_p = specific heat at constant pressure (Btu/lb ·°R)
c_v = specific heat at constant volume (Btu/lb ·°R)
D = internal diameter (ft)
D_H = equivalent hydraulic diameter (ft)
d = internal diameter (in)
d_{nom} = nominal pipe or valve size (in)
E = efficiency factor (unitless)
ehp = electrical horsepower (hp)
F_F = liquid critical pressure ratio factor (unitless)
F_K = specific heat ratio factor (unitless)
F_L = liquid pressure recovery factor (unitless)
F_{LP} = combined piping geometry and liquid pressure recovery factor (unitless)
F_P = piping geometry factor (unitless)
f = Darcy friction factor (unitless)
f_T = friction factor in zone of complete turbulence (unitless)
g = gravitational acceleration = 32.174 ft/s²
H = total head or fluid energy, in feet of fluid (ft)
h = static pressure head at a point, in feet of fluid (ft)
h_f = specific enthalpy of saturated liquid (Btu/lb)
h_{fg} = specific latent heat of evaporation (Btu/lb)
h_g = specific enthalpy of saturated vapor (Btu/lb)
h_L = loss of static pressure head due to fluid flow (ft)
h_w = static pressure head, in inches of water (in H_20)
K = resistance coefficient (unitless)
K_B = Bernoulli coefficient (unitless)
K_V = flow coefficient or flow factor (unitless)
k = ratio of specific heat at constant pressure (c_p) to specific heat at constant volume (c_v)
L = length of pipe (ft)
L/D = equivalent length of a resistance to flow, in pipe diameters
L_m = length of pipe, in miles (mi)
M = Mach number (unitless)
M_r = relative molecular mass
NPSHa = Net Positive Suction Head available (ft)
NRPD = Non-Recoverable Pressure Drop (psid)
n_a = number of moles of a gas
P = gauge pressure, in lb/in² (psig)
P' = absolute pressure, in lb/in² (psia)
P'_b = absolute pressure at standard conditions = 14.7 psia
P'_c = fluid critical pressure (psia)
P'_t = absolute tank surface pressure (psia)
P'_v = absolute fluid vapor pressure (psia)
P'_{vc} = absolute pressure at the vena contracta (psia)
p = gauge pressure, in lb/ft² (psfg)
p' = absolute pressure, in lb/ft² (psfa)
Q = rate of flow (gpm)
q = rate of flow at flowing conditions, in ft³/s (cfs)
q' = rate of flow at standard conditions (14.7 psia and 60°F) (ft³/s, scfs)
q_d = rate of flow at flowing conditions, in millions of cubic feet per day (MMcfd)
q'_d = rate of flow at standard conditions (14.7 psia and 60°F), in millions of cubic feet per day (MMscfd)
q_h = rate of flow at flowing conditions, in ft³/hr (cfh)
q'_h = rate of flow at standard conditions (14.7 psia and 60°F), in ft³/hr (scfh)
q_m = rate of flow at flowing conditions, in ft³/min (cfm)
q'_m = rate of flow at standard conditions (14.7 psia and 60°F), in ft³/min (scfm)
R = individual gas constant = $\overline{R}/M_r$ (ft · lb/lb_m ·°R)
$\overline{R}$ = universal gas constant = 1545.35 ft · lb/lbmol ·°R
R_e = Reynolds number (unitless)
R_H = hydraulic radius (ft)
r_c = critical pressure ratio for compressible flow
S = specific gravity of liquids at specified temperature relative to water at standard temperature (60°F) and pressure (14.7 psia)(unitless)
S_g = specific gravity of a gas relative to air = the ratio of the molecular weight of the gas to that of air (unitless)

Nomenclature

Unless otherwise stated, all symbols used in this book are defined as follows:

T = absolute temperature, in degrees Rankine (°R)
T_b = absolute temperature at standard condition = 520 °R
t = temperature, in degrees Fahrenheit (°F)
t_s = saturation temperature at a given pressure (°F)
V = mean velocity of flow, in ft/min (fpm)
$\overline{V}$ = specific volume of fluid (ft^3/lb)
V_a = volume (ft^3)
v = mean velocity of flow, in ft/s (fps)
v_s = sonic (or critical) velocity of flow of a gas (ft/s)
W = rate of flow (lb/hr)
w = rate of flow (lb/s)
w_a = weight (lb)
x = pressure drop ratio (unitless)
x_T = critical pressure drop ratio factor without fittings (unitless)
x_{TP} = critical pressure drop ratio factor with fittings (unitless)
Y = net expansion factor for compressible flow through orifices, nozzles, venturi, control valves or pipe (unitless)
Z = potential head or elevation above reference level (ft)
Z_f = compressibility factor (unitless)
Z_s = elevation at pump suction (ft)
Z_t = elevation at tank surface (ft)

Greek Letters
Alpha
α = angle (degrees)
Beta
β = ratio of small to large diameter in orifices and nozzles, and contractions or enlargements in pipes
Delta
Δ = differential between two points
Epsilon
ε = absolute roughness or effective height of pipe wall irregularities (ft)
Eta
η_m = motor efficiency (unitless)
η_p = pump efficiency (unitless)
η_{vsd}= variable speed drive (vsd) efficiency (unitless)
Mu
μ = absolute (dynamic) viscosity, in centipoise (cP)
μ_e = absolute viscosity, in pound mass per foot second (lbm/ft · s) or poundal seconds per square foot (pdl · s/ft^2)
μ'_e = absolute viscosity, in slugs per foot second (slug/ft · s) or in pound force seconds per square foot (lb · s/ft^2)
Nu
ν = kinematic viscosity, in centistokes (cSt)
ν' = kinematic viscosity (ft^2/s)
Phi
φ = potential energy term to account for elevation changes in isothermal compressible flow equations
Rho
ρ = weight density of fluid (lb/ft^3)
ρ' = mass density of fluid (g/cm^3)
ρ_a = weight density of air at standard conditions (14.7 psia and 60°F)
Sigma
Σ = summation
Theta
θ = angle of convergence or divergence in enlargements or contractions in pipes

Subscripts for Diameter
(1) defines smaller diameter
(2) defines larger diameter

Subscripts for Fluid Property
(1) defines inlet (upstream) condition
(2) defines outlet (downstream) condition

Subscript for Average Value
(avg) defines average condition

www. This symbol = online calculators are available at www.flowoffluids.com.

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Chapter 1

Theory of Flow in Pipe

The most commonly employed method of transporting fluid from one point to another is to force the fluid to flow through a piping system. Pipe of circular cross section is most frequently used because that shape offers not only greater structural strength, but also greater cross sectional area per unit of wall surface than any other shape. Unless otherwise stated, the word "pipe" in this book will always refer to a closed conduit of circular cross section and constant internal diameter.

Only a few special problems in fluid mechanics (laminar flow in pipe, for example) can be entirely solved by rational mathematical means; all other problems require methods of solution which rest, at least in part, on experimentally determined coefficients. Many empirical formulas have been proposed for the problem of flow in pipe, but these are often extremely limited and can be applied only when the conditions of the problem closely approach the conditions of the experiments from which the formulas were derived.

Because of the great variety of fluids being handled in modern industrial processes, a single equation which can be used for the flow of any fluid in pipe offers obvious advantages. Such an equation is the Darcy* formula. The Darcy formula can be derived rationally by means of dimensional analysis; however, one variable in the formula (the friction factor) must be determined experimentally. This formula has a wide application in the field of fluid mechanics and is used extensively throughout this paper.

*The Darcy formula is also known as the Weisbach formula or the Darcy-Weisbach formula; also, as the Fanning formula, sometimes modified so that the friction factor is one-fourth the Darcy friction factor.

Physical Properties of Fluids

The solution of any flow problem requires a knowledge of the physical properties of the fluid being handled. Accurate values for the properties affecting the flow of fluids (namely, viscosity and weight density) have been established by many authorities for all commonly used fluids and many of these data are presented in the various tables and charts in Appendix A.

Viscosity: Viscosity expresses the readiness with which a fluid flows when it is acted upon by an external force. The coefficient of absolute viscosity or, simply, the absolute viscosity of a fluid, is a measure of its resistance to internal deformation or shear. Molasses is a highly viscous fluid; water is comparatively much less viscous; and the viscosity of gases is quite small compared to that of water.

Although most fluids are predictable in their viscosity, in some, the viscosity depends upon the previous working of the fluid. Printer's ink, wood pulp slurries, and catsup are examples of fluids possessing such thixotropic properties of viscosity.

Considerable confusion exists concerning the units used to express viscosity; therefore, proper units must be employed whenever substituting values of viscosity into formulas. In the metric system, the unit of absolute viscosity is the poise which is equal to 100 centipoise. The poise has the dimensions of dyne seconds per square centimeter or of grams per centimeter second. It is believed that less confusion concerning units will prevail if the centipoise is used exclusively as the unit of viscosity. For this reason, and since most handbooks and tables follow the same procedure, all viscosity data in this paper are expressed in centipoise.

The English units commonly employed are "slugs per foot second" or "pound force seconds per square foot"; however, "pound mass per foot second" or "poundal seconds per square foot" may also be encountered. The viscosity of water at a temperature of 68°F is:

$$\mu = \quad 1 \text{ centipoise*} = \begin{cases} 0.01 \text{ poise} \\ 0.01 \text{ gram per cm second} \\ 0.01 \text{ dyne second per sq cm} \end{cases}$$

$$\mu_e = \begin{cases} 0.000\ 672 \text{ pound mass per foot second} \\ 0.000\ 672 \text{ poundal second per square foot} \end{cases}$$

$$\mu'_e = \begin{cases} 0.000\ 0209 \text{ slug per foot second} \\ 0.000\ 0209 \text{ pound force second per square ft} \end{cases}$$

Kinematic viscosity is the ratio of the absolute viscosity to the mass density. In the metric system, the unit of kinematic viscosity is the stoke. The stoke has dimensions of square centimeters per second and is equivalent to 100 centistokes.

Equation 1-1

$$\nu \text{ (centistokes)} = \frac{\mu \quad \text{(centipoise)}}{\rho' \text{ (grams per cubic cm)}} = \frac{\mu}{S_{4°C}}$$

By definition, the specific gravity, S, in the foregoing formula is based upon water at a temperature of 4°C (39.2°F), whereas specific gravity used throughout this paper is based upon water at 60°F. In the English system, kinematic viscosity has dimensions of square feet per second.

Factors for conversion between metric and English system units of absolute and kinematic viscosity are given on page B-3 of Appendix B.

The measurement of the absolute viscosity of fluids (especially gases and vapors) requires elaborate equipment and considerable experimental skill. On the other hand, a rather simple instrument can be used for measuring the kinematic viscosity of oils and other viscous liquids. The instrument adopted as a standard is the Saybolt Universal Viscometer. In measuring kinematic viscosity with this instrument, the time required for a small volume of liquid to flow through an orifice is determined; consequently, the "Saybolt viscosity" of the liquid is given in seconds. For very viscous liquids, the Saybolt Furol instrument is used.

Other viscometers, somewhat similar to the Saybolt but not used to any extent, is the Engler, the Redwood Admiralty, and the Redwood. The relationship between Saybolt viscosity and kinematic viscosity is shown on page B-4; equivalents of kinematic, Saybolt Universal, Saybolt Furol, and absolute viscosity can be obtained from the chart on page B-5.

The viscosities of some of the most common fluids are given on pages A-2 to A-6. It will be noted that, with a rise in temperature, the viscosity of liquids decreases, whereas the viscosity of gases increases. The effect of pressure on the viscosity of liquids and ideal gases is so small that it is of no practical interest in most flow problems. Conversely, the viscosity of saturated, or only slightly superheated, vapors is appreciably altered by pressure changes, as indicated on page A-2 showing the viscosity of steam. Unfortunately, the data on vapors are incomplete and, in some cases, contradictory. Therefore, it is expedient when dealing with vapors other than steam to neglect the effect of pressure because of the lack of adequate data.

*Actually the viscosity of water at 68°F is 1.005 centipoise.

Physical Properties of Fluids

Weight density, specific volume, and specific gravity: The weight density or specific weight of a substance is its weight per unit volume. In the English system of units, this is expressed in pounds per cubic foot and the symbol designation used in this paper is ρ (Rho). In the metric system, the unit is grams per cubic centimeter and the symbol designation used is ρ' (Rho prime).

The specific volume $\overline{V}$, being the reciprocal of the weight density, is expressed in the English system as the number of cubic feet of space occupied by one pound of the substance, thus:

$$\overline{V} = \frac{1}{\rho}$$ **Equation 1-2**

The variations in weight density as well as other properties of water with changes in temperature are shown on page A-7. The weight densities of other common liquids are shown on page A-8. Unless very high pressures are being considered, the effect of pressure on the weight of liquids is of no practical importance in flow problems.

The weight densities of gases and vapors, however, are greatly altered by pressure changes. For ideal gases, the weight density can be computed from the ideal gas equation:

$$\rho = \frac{144\,P'}{RT}$$ **Equation 1-3**

The individual gas constant R is equal to the universal gas constant, $\overline{R}$ = 1545, divided by the molecular mass of the gas,

$$R = \frac{1545.35}{M_r}$$ **Equation 1-4**

Values of R, as well as other useful gas constants are given on page A-9. The weight density of air for various conditions of temperature and pressure can be found on page A-11.

In steam flow computations, the reciprocal of the weight density, which is the specific volume, is commonly used; these values are listed in the steam tables shown on pages A-12 to A-20.

Specific gravity is a relative measure of weight density. Since pressure has an insignificant effect upon the weight density of liquids, temperature is the only condition that must be considered in designating the basis for specific gravity. The specific gravity of a liquid is the ratio of its weight density at specified temperature to that of water at standard temperature, 60°F.

Equation 1-5

$$S = \frac{\rho \text{ (any liquid at 60°F, unless otherwise specified)}}{\rho \text{ (water at 60°F)}}$$

A hydrometer can be used to measure the specific gravity of liquids directly. Three hydrometer scales are common; the API scale which is used for oils, and the two Baumé scales, one for liquids heavier than water and one for liquids lighter than water. The relationship between the hydrometer scales and specific gravity are:

For oils,

$$S\ (60°F/60°F) = \frac{141.5}{131.5 + \text{deg. API}}$$ **Equation 1-6**

For liquids lighter than water,

$$S\ (60°F/60°F) = \frac{140}{130 + \text{deg. Baumé}}$$ **Equation 1-7**

For liquids heavier than water,

$$S\ (60°F/60°F) = \frac{145}{145 - \text{deg. Baumé}}$$ **Equation 1-8**

For convenience in converting hydrometer readings to more useful units, refer to the table shown on page B-6.

The specific gravity of gases is defined as the ratio of the molecular mass of the gas to that of air, and as the ratio of the individual gas constant of air to that of the gas.

$$S_g = \frac{R\text{ (air)}}{R\text{ (gas)}} = \frac{M_r\text{ (gas)}}{M_r\text{ (air)}}$$ **Equation 1-9**

Vapor Pressure: Vapor pressure is the absolute pressure at which a liquid changes phase to a gas at a given temperature. For an enclosed fluid at rest, it is the pressure exerted on the liquid surface when the rate of evaporation from the liquid equals the rate of condensation of vapor above the surface. Vapor bubbles will form in a liquid when its absolute pressure is at or below its vapor pressure. Vapor pressure is dependent on fluid temperature and increases with increasing temperature. Vapor pressure is also referred to as the "saturation pressure" and is tabulated for water as a function of temperature on page A-7.

Nature of Flow in Pipe - Laminar and Turbulent

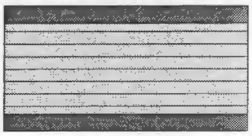


Figure 1-1: Laminar Flow
This is an illustration of colored filaments being carried along undisturbed by a stream of water.

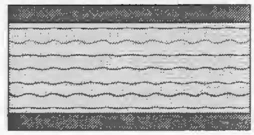


Figure 1-2: Flow in Critical Zone
At the critical velocity, the filaments begin to break up, indicating flow is becoming turbulent.

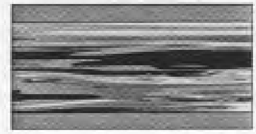


Figure 1-3: Turbulent Flow
This illustration shows the turbulence in the stream completely dispersing the colored filaments a short distance downstream from the point of injection.

A simple experiment (illustrated above) will readily show there are two entirely different types of flow in pipe. The experiment consists of injecting small streams of a colored fluid into a liquid flowing in a glass pipe and observing the behavior of these colored streams at different sections downstream from their points of injection.

If the discharge or average velocity is small, the streaks of colored fluid flow in straight lines, as shown in Figure 1-1. As the flow rate is gradually increased, these streaks will continue to flow in straight lines until a velocity is reached when the streaks will waver and suddenly break into diffused patterns, as shown in Figure 1-2. The velocity at which this occurs is called the "critical velocity." At velocities higher than "critical," the filaments are dispersed at random throughout the main body of the fluid, as shown in Figure 1-3.

The type of flow which exists at velocities lower than "critical" is known as laminar flow and, sometimes, as viscous or streamline flow. Flow of this nature is characterized by the gliding of concentric cylindrical layers past one another in orderly fashion. Velocity of the fluid is at its maximum at the pipe axis and decreases sharply to zero at the wall.

At velocities greater than "critical," the flow is turbulent. In turbulent flow, there is an irregular random motion of fluid particles in directions transverse to the direction of the main flow. The velocity distribution in turbulent flow is more uniform across the pipe diameter than in laminar flow. Even though a turbulent motion exists throughout the greater portion of the pipe diameter, there is always a thin layer of fluid at the pipe wall, known as the "boundary layer" or "laminar sub-layer," which is moving in laminar flow.

Mean velocity of flow: The term "velocity," unless otherwise stated, refers to the mean, or average, velocity at a given cross section, as determined by the continuity equation for steady state flow:

$$v = \frac{q}{A} = \frac{w}{A\rho} = \frac{w\bar{V}}{A}$$ **Equation 1-10**

"Reasonable" velocities for use in design work are given on page A-10.

Reynolds number: The work of Osborne Reynolds has shown that the nature of flow in pipe (whether it is laminar or turbulent) depends on the pipe diameter, the density and viscosity of the flowing fluid, and the velocity of flow. The numerical value of a dimensionless combination of these four variables, known as the Reynolds number, may be considered to be the ratio of the dynamic forces of mass flow to the shear stress due to viscosity. Reynolds number is:

$$R_e = \frac{Dv\rho}{\mu_e}$$ **Equation 1-11**

(other forms of this equation; page 6-2.)

For engineering purposes, flow in pipes is usually considered to be laminar if the Reynolds number is less than 2000, and turbulent if the Reynolds number is greater than 4000. Between these two values lies the "critical zone" where the flow (being laminar, turbulent, or in the process of change, depending upon many possible varying conditions) is unpredictable. Careful experimentation has shown that the laminar zone may be made to terminate at a Reynolds number as low as 1200 or extended as high as 40,000, but these conditions are not expected to be realized in ordinary practice.

Noncircular Conduit: When a conduit of noncircular cross section is encountered, the equivalent hydraulic diameter (equal to four times the hydraulic radius) should be used as a substitute for diameter in Reynolds number, friction factor, relative roughness and resistance value calculations.

$$R_H = \frac{\text{cross sectional area}}{\text{wetted perimeter}}$$ **Equation 1-12**

This applies to any ordinary conduit (partially full circular, oval, square or rectangular conduit) under turbulent flow, but does not apply to laminar flow conditions. For extremely narrow shapes such as annular or elongated openings, where width is small relative to length, hydraulic radius may not provide accurate results.

Equivalent diameter is the diameter of a circular pipe that gives the same area as a noncircular conduit and is substituted for diameter in equations where velocity and flow are calculated. This should not be confused with equivalent hydraulic diameter.

For example, to determine flow rate for a noncircular conduit using Equation 1-13:

$$q = \frac{\pi d^2}{4}\sqrt{\frac{2gh_L D}{fL}}$$ **Equation 1-13**

the value d is replaced with the equivalent diameter of the actual flow area and $4R_H$ (equivalent hydraulic diameter) is substituted for D.

General Energy Equation - Bernoulli's Theorem

The Bernoulli theorem is a means of expressing the application of the law of conservation of energy to the flow of fluids in a conduit. The total energy at any particular point, above some arbitrary horizontal datum plane, is equal to the sum of the elevation head, the pressure head, and the velocity head, as follows:

$$Z + \frac{144P}{\rho} + \frac{v^2}{2g} = H$$

Equation 1-14

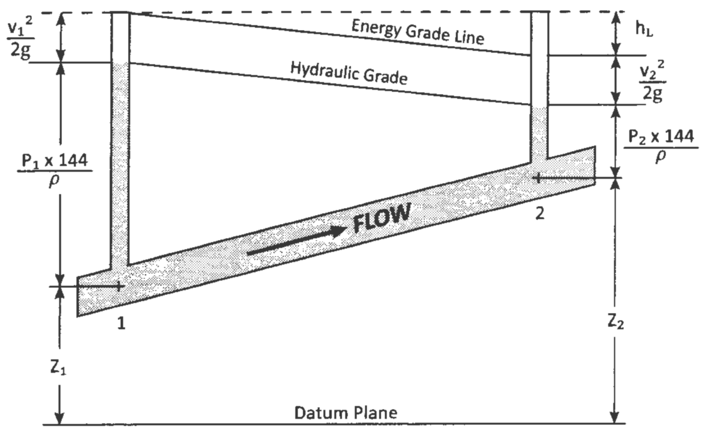


Figure 1-4: Energy Balance for Two Points in a Fluid[1]

If friction losses are neglected and no energy is added to, or taken from, a piping system (i.e., pumps or turbines), the total head, H, in the above equation will be a constant for any point in the fluid. However, in actual practice, losses or energy increases or decreases are encountered and must be included in the Bernoulli equation. Thus, an energy balance may be written for two points in a fluid, as shown in the example in Figure 1-4.

Note the pipe friction loss from point 1 to point 2 is h_L foot pounds per pound of flowing fluid; this is sometimes referred to as the head loss in feet of fluid.
The equation may be written as follows:

Equation 1-15

$$Z_1 + 144\frac{P_1}{\rho_1} + \frac{v_1^2}{2g} = Z_2 + 144\frac{P_2}{\rho_2} + \frac{v_2^2}{2g} + h_L$$

All practical formulas for the flow of fluids are derived from Bernoulli's theorem with modifications to account for losses due to friction.

Measurement of Pressure

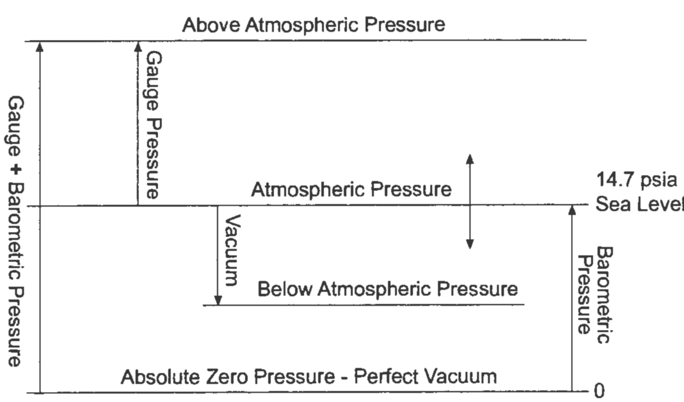


Figure 1-5: Relationship Between Gauge and Absolute Pressures[1]

Figure 1-5 graphically illustrates the relationship between gauge and absolute pressures. Perfect vacuum cannot exist on the surface of the earth, but it nevertheless makes a convenient datum for the measurement of pressure.

Barometric pressure is the level of the atmospheric pressure above perfect vacuum.

"Standard" atmospheric pressure is 14.696 pounds per square inch or 760 millimeters of mercury.

Gauge pressure is measured above atmospheric pressure, while absolute pressure always refers to perfect vacuum as a base.

Vacuum, usually expressed in inches of mercury, is the depression of pressure below the atmospheric level. Reference to vacuum conditions is often made by expressing the absolute pressure in inches of mercury; also millimeters of mercury and microns of mercury.

Head Loss and Pressure Drop Through Pipe

Flow in pipe is always accompanied by friction of fluid particles rubbing against one another, and consequently, by loss of energy available for work; in other words, there must be a pressure drop in the direction of flow. If ordinary Bourdon tube pressure gauges were connected to a pipe containing a flowing fluid, as shown in Figure 1-6, gauge P_1 would indicate a higher static pressure than gauge P_2.

Figure 1-6:

The general equation for pressure drop, known as Darcy's formula and expressed in feet of fluid, is:

$$h_L = f \frac{L}{D} \frac{v^2}{2g}$$ **Equation 1-16**

This equation may be written to express pressure drop in pounds per square inch, by substitution of proper units, as follows:

$$\Delta P = f \frac{L}{D} \frac{v^2}{2g} \frac{\rho}{144}$$ **Equation 1-17**

The Darcy equation is valid for laminar or turbulent flow of any liquid in a pipe. However, when extreme velocities occurring in a pipe cause the downstream pressure to fall to the vapor pressure of the liquid, cavitation occurs and calculated flow rates will be inaccurate. With suitable restrictions, the Darcy equation may be used when gases and vapors (compressible fluids) are being handled. These restrictions are defined on page 1-8.

Equation 1-17 gives the loss in pressure due to friction and applies to pipe of constant diameter carrying fluids of reasonably constant weight density in straight pipe, whether horizontal, vertical, or sloping. For inclined pipe, vertical pipe, or pipe of varying diameter, the change in pressure due to changes in elevation, velocity, and weight density of the fluid must be made in accordance with Bernoulli's theorem. See Example 7-25 for a calculation using this theorem.

Friction factor: The Darcy formula can be rationally derived by dimensional analysis, with the exception of the friction factor, *f*, which must be determined experimentally. The friction factor for laminar flow conditions ($R_e < 2000$) is a function of Reynolds number only; whereas, for turbulent flow ($R_e > 4000$), it is also a function of the character of the pipe wall.

A region known as the "critical zone" occurs between Reynolds number of approximately 2000 and 4000. In this region, the flow may be either laminar or turbulent depending upon several factors; these include changes in section or direction of flow and obstructions, such as valves, in the upstream piping. The friction factor in this region is indeterminate and has lower limits based on laminar flow and upper limits based on turbulent flow conditions.

At Reynolds numbers above approximately 4000, flow conditions again become more stable and definite friction factors can be established. This is important because it enables the engineer to determine the flow characteristics of any fluid flowing in a pipe, providing the viscosity and weight density at flowing conditions are known. For this reason, Equation 1-17 is recommended in preference to some of the commonly known empirical equations for the flow of water, oil, and other liquids, as well as for the flow of compressible fluids when restrictions previously mentioned are observed.

If the flow is laminar ($R_e < 2000$), the friction factor may be determined from the equation:

$$f = \frac{64}{R_e} = \frac{64\mu_e}{Dv\rho} = \frac{64\mu}{124dv\rho}$$ **Equation 1-18**

If this quantity is substituted into Equation 1-17, the pressure drop in pounds per square inch is:

$$\Delta P = 0.000668 \frac{\mu L v}{d^2}$$ **Equation 1-19**

which is Poiseuille's law for laminar flow.

When the flow is turbulent ($R_e > 4000$), the friction factor depends not only upon the Reynolds number but also upon the relative roughness, ε/D (the roughness of the pipe walls as compared to the diameter of the pipe). For very smooth pipes such as drawn brass tubing and glass, the friction factor decreases more rapidly with increasing Reynolds number than for pipe with comparatively rough walls.

Since the character of the internal surface of commercial pipe is practically independent of the diameter, the roughness of the walls has a greater effect on the friction factor in the small sizes. Consequently, pipe of small diameter will approach the very rough condition and, in general, will have higher friction factors than large pipe of the same material.

The most useful and widely accepted data of friction factors for use with the Darcy formula have been presented by L. F. Moody[2] and are shown on pages A-24 to A-26.

The friction factor, *f*, is plotted on page A-25 on the basis of relative roughness obtained from the chart on page A-24 and the Reynolds number. The value of *f* is determined by horizontal projection from the intersection of the ε/D curve under consideration with the calculated Reynolds number to the left hand vertical scale of the chart on page A-24. Since most calculations involve commercial steel pipe, the chart on page A-26 is furnished for a more direct solution. It should be kept in mind that these figures apply to clean new pipe.

Head Loss and Pressure Drop Through Pipe

Colebrook Equation: With the rising use of computers and software, it has become desirable to use equations that can be entered into a program or spreadsheet to solve for the turbulent friction factor rather than use the Moody diagram. The Colebrook equation offers an implicit, iterative solution that correlates well with the Moody Diagram.

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon}{3.7D} + \frac{2.51}{R_e\sqrt{f}}\right)$$ **Equation 1-20**

Explicit Approximations of Colebrook:[3] While the Colebrook equation provides the most accurate values for the friction factor, the iterative nature of the solution makes it problematic when looking at a system of pipes. For this reason there are a number of explicit approximations available to reach a direct solution for friction factor values. The Serghide equation offers a complex, but highly accurate explicit approximation of the Colebrook equation and is applicable over the full range of turbulent flow.

Equation 1-21

$$A = -2\log\left[\frac{\left(\frac{\varepsilon}{D}\right)}{3.7} + \frac{12}{R_e}\right]$$

$$B = -2\log\left[\frac{\left(\frac{\varepsilon}{D}\right)}{3.7} + \frac{2.51A}{R_e}\right]$$

$$C = -2\log\left[\frac{\left(\frac{\varepsilon}{D}\right)}{3.7} + \frac{2.51B}{R_e}\right]$$

$$f = \left[A - \frac{(B-A)^2}{C - 2B + A}\right]^{-2}$$

The Swamee-Jain is a simpler explicit approximation, but can vary up to 2.8% within its applicable range of Reynolds numbers between 5000 and 3×10^8, and relative roughness values from 10^{-6} to 0.01.

$$f = \frac{0.25}{\left[\log\left(\frac{\varepsilon}{3.7D} + \frac{5.74}{R_e^{0.9}}\right)\right]^2}$$ **Equation 1-22**

Hazen-Williams Formula for Flow of Water:[4] Although the Darcy formula for flow problems has been recommended in this paper due to its broad range of application, some industries prefer the use of empirical formulas like the Hazen-Williams formula. This formula is only appropriate for fully turbulent flow of fluids that are similar to 60°F water.

$$\Delta P_{per\_foot} = 4.52\,\frac{Q^{1.85}}{C^{1.85}\,d^{4.87}}$$ **Equation 1-23**

The calculation of this formula requires the use of the Hazen-Williams C factor which varies with the piping material. A table of some typical C factor values is included with the equations on page 6-3.

Effect of age and use on pipe friction: Pipe aging affects both the roughness and inside diameter of the pipe due to corrosion, sedimentation, encrustation with scale, tubercules, or other foreign matter. The rate of corrosion is dependent upon the fluid's chemical composition, temperature, pH, and concentration of dissolved gases. Other factors include the compatibility between the fluid and pipe material and the use of water chemistry controls to minimize corrosion. Processes that reduce the pipe diameter such as sedimentation and encrustation are dependent on variables such as the fluid velocity, the concentration of particulates in the fluid, and factors that influence biological growth inside the pipe.

Changes to the inside diameter have a much greater impact on the head loss than do changes in the roughness of the inside surface of the pipe wall, as shown in the graph of head loss vs. flow rate for the flow of 60°F water in a 100 foot length of 4 inch schedule 40 steel pipe (Figure 1-7). At 250 gpm, a new pipe with an absolute roughness of 0.0018 inch and an inside diameter of 4.026 inch will have 3.5 feet of head loss. If corrosion increases the roughness by 150% to 0.0045 inch (equivalent to the roughness of asphalted cast iron pipe), head loss at 250 gpm increases to 4.0 feet, or a 14% increase. If the pipe inside diameter is reduced by 5% to 3.825 inch, the head loss at 250 gpm increases to 4.5 feet, a 29% increase. If both the change in roughness and reduction in inside diameter are taken into account, head loss increases to 5.2 feet, or 49%.

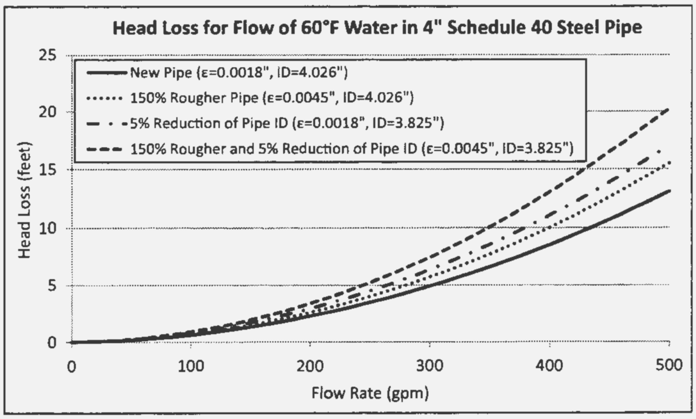


Figure 1-7: Head loss in aged pipe

It is difficult to predict the effects of pipe aging on head loss due to the number of highly variable factors over the life of the pipeline. Some studies attempt to calculate an annual growth rate of roughness based on the pH or calcium carbonate content of the fluid.[5,6] Other studies use field measurements taken over the age of the pipeline to determine a roughness growth rate or to show how the Hazen-Williams C factor changes over time.[7,8] Although the studies provide insight into the need to take into account the effect of pipe aging on head loss calculations, it is not prudent to apply any particular thumb rule to all piping systems. Changes to the pipe properties over time are usually taken into account by adding a design margin to the total head calculation for sizing and selecting a pump.

Principles of Compressible Flow in Pipe

An accurate determination of the pressure drop of a compressible fluid flowing through a pipe requires knowledge of the relationship between pressure and specific volume of the fluid. This is not easily determined in each particular situation and increases the complexity of compressible flow problems relative to incompressible. Compressible flow can often be described thermodynamically as polytropic, where $P'\bar{V}^n = C$ and both C and n are constants. For adiabatic flow (i.e., no heat transfer) $n = k$ or $P'\bar{V}^k = C$ and for isothermal flow (i.e., constant temperature) $n = 1$ or $P'\bar{V} = C$; which are commonly employed assumptions for compressible flow problems.

Definition of a perfect gas: The assumption of a perfect gas is often used to simplify compressible flow problems and is the basis for the equations presented in this paper. A perfect gas is one that obeys the ideal gas equation of state and has a constant specific heat ratio k. The ideal gas equation represents a hypothetical model of gas behavior that serves as an accurate approximation in many engineering problems:

$$p'\bar{V} = RT$$ **Equation 1-24**

It is generally stated that an ideal gas can be assumed for temperatures that are sufficiently above the critical temperature and pressures that are sufficiently below the critical pressure of the given gas.

The specific heat ratio varies only with temperature for an ideal gas and is defined as the ratio of the specific heat capacity of a fluid at constant pressure (c_p) to that at constant volume (c_v). The assumption of a perfect gas, for which the specific heats are constant, is justified since k has been shown to only change appreciably over a large temperature range and is almost unaffected by pressure.[9] A reasonable value of k for most diatomic gases is 1.4 and pages A-9 and A-10 give values of k for gases and steam.

Speed of sound and Mach number: Another important concept in compressible flow is the speed of sound (c) in a fluid. When a compressible fluid is disturbed, a signal in the form of a pressure wave propagates from the region of disturbance to other regions in the fluid. The speed of sound is the speed at which this pressure wave is transmitted and for a perfect gas is expressed as:

$$c = \sqrt{kgRT}$$ **Equation 1-25**

The speed of sound relative to the velocity of the fluid has many implications in compressible flow. A useful relationship is the Mach number (M), a dimensionless ratio of the velocity of the fluid to the speed of sound in the fluid at local conditions:

$$M = \frac{v}{c}$$ **Equation 1-26**

For sonic flow $M = 1$ and the fluid velocity is equal to the speed of sound. Supersonic flows occur at $M > 1$ whereas most compressible pipe flows are subsonic and occur at $M < 1$.

The Mach number can be helpful in determining the appropriate assumptions for a particular problem and is used extensively in more rigorous treatments for compressible flow of a perfect gas.

Approaches to compressible flow problems: It is common practice in gas flow problems to apply appropriate assumptions that simplify the problem while still providing a sufficiently accurate solution. Typically, flows in which the fluid specific volume varies by more than 5 to 10 percent are considered compressible.[10] This leads to the conclusion that cases in which the flow of a gas undergoes only slight changes in specific volume may be accurately treated as incompressible. If the problem is indeed compressible then it is usually treated as either isothermal or adiabatic, since these two types encompass most common compressible flow problems.

Isothermal flow is often assumed, partly for convenience but more often because it is closer to fact in piping practice. The flow of gases in long pipelines closely approximates isothermal conditions since there is usually adequate heat transfer to maintain a constant temperature.

Adiabatic flow is usually assumed in short perfectly insulated pipe, as seen in most industrial settings. This would be consistent since no heat is transferred to or from the pipe, except for the fact that the minute amount of heat generated by friction is added to the flow.

Application of the Darcy equation to compressible fluids: For cases dealing with compressible fluids in which the pressure drop is relatively low and the specific volume and velocity do not change appreciably, the flow may be treated as incompressible. Additionally, some resources indicate that the Mach number must not exceed 0.1 to 0.2 in order for compressibility effects to be assumed negligible.[10] For gas and vapor flow problems that can be treated as incompressible, the Darcy equation may be applied with the following restrictions:

1. If the calculated pressure drop (ΔP) is less than ~10% of the absolute inlet pressure P'_1, reasonable accuracy will be obtained if the specific volume used in the equation is based upon either the upstream or downstream conditions, whichever are known.

2. If the calculated pressure drop (ΔP) is greater than ~10%, but less than ~40% of the absolute inlet pressure P'_1, the Darcy equation will give reasonable accuracy by using a specific volume based upon the average of the upstream and downstream conditions; otherwise, the method given on page 1-11 may be used.

3. For greater pressure drops, such as are often encountered in long pipelines, the methods given on the next three pages should be used.

Principles of Compressible Flow in Pipe

Complete Isothermal Equation: The flow of gases in long pipelines closely approximates isothermal conditions. The pressure drop in such lines is often large relative to the inlet pressure, and solution of this problem falls outside the limitations of the Darcy equation. An accurate determination of the flow characteristics falling within this category can be made by using the complete isothermal equation:

Equation 1-27

$$w^2 = \left[\frac{144gA^2}{\bar{V}_1\left(f\frac{L}{D} + 2\ln\frac{P'_1}{P'_2}\right)}\right]\left[\frac{(P'_1)^2 - (P'_2)^2}{P'_1}\right]$$

The equation is developed from the generalized compressible flow equation by making the following assumptions;

1. Isothermal flow.
2. No mechanical work is done on or by the system.
3. Steady flow or the discharge is unchanged with time.
4. The fluid obeys the perfect gas laws.
5. The velocity may be represented by the average velocity at a cross section.
6. The friction factor is constant along the pipe.
7. The pipeline is straight and horizontal between end points.

Simplified Isothermal - Gas Pipeline Equation: In the practice of gas pipeline engineering, another assumption is added to the foregoing:

8. The pipeline is sufficiently long to neglect acceleration.

Then, the equation for discharge in a horizontal pipe may be written:

$$w^2 = \left(\frac{144gDA^2}{\bar{V}_1 fL}\right)\left[\frac{(P'_1)^2 - (P'_2)^2}{P'_1}\right]$$ **Equation 1-28**

This is equivalent to the complete isothermal equation if the pipeline is sufficiently long and also for shorter lines if the ratio of pressure drop to initial pressure is small.

Since gas flow problems are usually expressed in terms of cubic feet per hour at standard conditions, it is convenient to rewrite Equation 1-28 as follows:

Equation 1-29

$$q'_h = 3.2308\left(\frac{T_b}{P'_b}\right)\left[\frac{(P'_1)^2 - (P'_2)^2}{f\,L_m\,T_{avg}\,S_g}\right]^{0.5} d^{2.5}$$

The average temperature term (T_{avg}) in Equation 1-29 can be taken as the arithmetic mean of the upstream and downstream temperatures when the change in temperature is no more than 10°F. Methods to calculate T_{avg} for larger temperature variations are given in other sources.[11]

Other commonly used equations for compressible flow in long pipelines:

Weymouth equation:[11] **Equation 1-30**

$$q'_h = 18.062\left(\frac{T_b}{P'_b}\right)\left[\frac{(P'_1)^2 - (P'_2)^2}{L_m\,T_{avg}\,S_g}\right]^{0.5} d^{2.667}$$

This equation is commonly used for sizing gas pipelines and is applicable for fully turbulent flow.

Panhandle A equation:[11] **Equation 1-31**

$$q'_h = 18.161\,E\left(\frac{T_b}{P'_b}\right)^{1.0788}\left[\frac{(P'_1)^2 - (P'_2)^2}{L_m\,T_{avg}\,S_g^{0.8539}}\right]^{0.5394} d^{2.6182}$$

This equation was developed for natural gas pipelines and it is applicable for partially turbulent (hydraulically smooth) flow. In previous versions of this paper a specific gravity of 0.6 and a temperature of 60°F were assumed and built into the equation, since these were the typical conditions encountered in its application for natural gas pipelines. In this paper the inclusion of a temperature and specific gravity term in the equation allows for more general use.

Panhandle B equation:[11] **Equation 1-32**

$$q'_h = 30.708\,E\left(\frac{T_b}{P'_b}\right)^{1.02}\left[\frac{(P'_1)^2 - (P'_2)^2}{L_m\,T_{avg}\,S_g^{0.961}}\right]^{0.510} d^{2.53}$$

This equation is widely used for long transmission lines and is applicable for fully turbulent flow.

Both Panhandle equations incorporate the efficiency factor (E) which is used to correct for additional resistances in pipelines such as valves, fittings, and debris, as well as general age and condition. It is commonly associated with the Panhandle equations; however practitioners may apply it to the Weymouth equation as well. It is usually assumed to be 0.92 for average operating conditions and typically ranges in value from 1.00 to 0.85, depending on the condition and design of the system.

Comparison of equations for compressible flow in pipelines: Equations 1-29, 1-30, 1-31, and 1-32 are derived from the same basic isothermal flow equation, but differ in the selection of data used for the determination of the friction factors.

Friction factors in accordance with the Moody[2] diagram or the Colebrook equation are normally used with the simplified isothermal equation. However, if a friction factor is derived from the Weymouth or Panhandle equations and used in the simplified isothermal equation, identical answers will be obtained.

The Weymouth friction factor[11] is dependent only on pipe diameter and is defined as:

Equation 1-33

$$f = \frac{0.032}{d^{0.333}}$$

Principles of Compressible Flow in Pipe

This is identical to the Moody friction factor in the fully turbulent flow range for 20 inch I.D. pipe only. Weymouth friction factors are greater than Moody factors for sizes less than 20 inches, and smaller for sizes larger than 20 inches.

The Panhandle A and B friction factors are dependent on pipe diameter and Reynolds number and are defined respectively as:[11]

$$f = 0.0847 \frac{1}{R_e^{0.1461}} \qquad \textbf{Equation 1-34}$$

$$f = 0.0147 \frac{1}{R_e^{0.03922}} \qquad \textbf{Equation 1-35}$$

In the flow range to which the Panhandle A equation is applicable, Equation 1-34 results in friction factors that are lower than those obtained from either the Moody diagram or the Weymouth friction equation. As a result, flow rates obtained by solution of the Panhandle A equation are usually greater than those obtained by employing either the simplified isothermal equation with Moody friction factors, or the Weymouth equation.

An example of the variation in flow rates which may be obtained for a specific condition by employing these equations is given in Example 7-18.

Another popular method for determining the friction factor for gas pipelines was developed by the American Gas Association (AGA). In this case, the AGA defined friction factors for both partially and fully turbulent flows that can be used in the simplified isothermal Equation 1-29.[11]

The AGA friction factor for partially turbulent flow is given by the Prandtl smooth pipe law modified with the drag factor F_f:

$$\frac{1}{\sqrt{f}} = F_f \, 2\log\left(\frac{R_e\sqrt{f}}{2.825}\right) \qquad \textbf{Equation 1-36}$$

The drag factor is used to account for bends and fittings in the pipeline and ranges in value from 0.9 to 0.99. Values of the drag factor for specific pipe configurations may be found in the AGA report.[11]

The AGA friction factor for fully turbulent flow is given by the Nikuradse rough pipe law:

$$\frac{1}{\sqrt{f}} = 2\log\left(\frac{3.7D}{\varepsilon}\right) \qquad \textbf{Equation 1-37}$$

An attempt has been made in this section to present the most common equations used in gas pipeline calculations. Many other equations are available for gas pipeline calculations such as the Institute of Gas Technology distribution, Mueller, and Fritzsche equations or the more recent Gersten et al. equation developed through work by GERG (Groupe European de Recherches Gazieres).[12] However, this is by no means an exhaustive list and the reader is encouraged to consult additional resources, with those cited in this paper providing a good starting point.[12,13,14]

Modifications to the Isothermal flow equations: Pipeline practitioners may modify the previously discussed isothermal flow equations in order to more accurately describe a particular system. The compressibility factor (Z_f) is often applied to account for real gas behavior that deviates from the ideal gas equation. Additionally, the previous assumption of a horizontal pipeline is often foregone and elevation changes are accounted for through the inclusion of a potential energy term. Considering that some systems may undergo appreciable elevation changes and modern gas pipelines may operate at much higher pressures than those when the Weymouth and Panhandle equations were developed, the inclusion of the compressibility and potential energy terms may be warranted. For example, the Weymouth equation was originally devised for gas pipelines with operating pressures in the range of 35 to 100 psig, but with the inclusion of the compressibility factor may be extended to systems operating at pressures upward of 1,000 to 3,200 psig.[13]

The compressibility factor is defined as:

$$Z_f = \frac{P'\overline{V}}{RT} \qquad \textbf{Equation 1-38}$$

For an ideal gas $Z_f = 1$ and for real gases Z_f can be greater or less than one depending on the fluid and conditions. Values of the compressibility factor can be determined from a variety of methods.[11,14] However, one must be mindful of the applicable conditions of any particular method for evaluating Z_f.

For the isothermal flow equations the compressibility factor is evaluated at average conditions of the flowing gas. T_{avg} is used for the temperature and the average pressure (P'_{avg}) may be taken as the arithmetic mean of the upstream and downstream pressures for pressure drops less than 0.2 percent. For greater pressure drops (P'_{avg}) may be obtained with the following expression.[11]

$$P'_{avg} = \frac{2}{3}\left[\frac{(P'_1)^3 - (P'_2)^3}{(P'_1)^2 - (P'_2)^2}\right] \qquad \textbf{Equation 1-39}$$

The potential energy term to account for elevation changes can be calculated based on an average gas density:[11]

$$\varphi = 0.0375 \frac{S_g \, \Delta Z \, (P'_{avg})^2}{T_{avg} \, Z_{f,avg}} \qquad \textbf{Equation 1-40}$$

The compressibility factor and potential energy terms are applied to the simplified isothermal equation (Equation 1-29) as follows:

Equation 1-41

$$q'_h = 3.2308 \left(\frac{T_b}{P'_b}\right)\left[\frac{(P'_1)^2 - (P'_2)^2 - \varphi}{f \, L_m \, T_{avg} \, Z_{f,avg} \, S_g}\right]^{0.5} d^{2.5}$$

These modifications may be applied to the other isothermal equations in the same manner.

Principles of Compressible Flow in Pipe

Limiting flow of gases and vapors: The feature not evident in the preceding equations (Equations 1-17 and 1-27 to 1-32 inclusive) is that the mass flow rate of a compressible fluid in a pipe, with a given upstream pressure will approach a certain maximum which it cannot exceed, no matter how much the downstream pressure is reduced. This condition is known as choked flow for compressible fluids.

The maximum velocity of a compressible fluid in a pipe is limited by the speed of sound in the fluid. Since pressure decreases and velocity increases as fluid proceeds downstream in a pipe of uniform cross section, the maximum velocity occurs in the downstream end of the pipe. If the pressure drop is sufficiently high, the exit velocity will reach the speed of sound. Further decrease in the outlet pressure will not be felt upstream because the pressure wave can only travel at the speed of sound in the fluid, and the signal will never translate upstream. The surplus pressure drop obtained by lowering the outlet pressure after the maximum discharge has already been reached takes place beyond the end of the pipe. This pressure is lost in shock waves and turbulence of the jetting fluid. The maximum possible velocity occurs at $M = 1$ and is often termed sonic velocity (v_s), which may be expressed as:

$$v_s = c = \sqrt{kgRT} \qquad \textbf{Equation 1-42}$$

The sonic velocity will occur at the pipe outlet or in a constricted area. The pressure and temperature are those occurring at the point in question. When compressible fluids discharge from the end of a reasonably short pipe of uniform cross section into an area of larger cross section, the flow is usually considered to be adiabatic. Investigation of the complete theoretical analysis of adiabatic flow[15] has led to a basis for establishing correction factors, which may be applied to the Darcy equation for this condition of flow. These correction factors compensate for the changes in fluid properties due to expansion and are identified as Y net expansion factors; given on page A-23.

The Darcy equation, including the Y factor, is:

$$w = 0.525Yd^2 \sqrt{\frac{\Delta P}{KV_1}} \qquad \textbf{Equation 1-43}$$

(Resistance coefficient K is defined on page 2-7)

It should be noted that the value of K in this equation is the total resistance coefficient of the pipeline, including entrance and exit losses when they exist, and losses due to valves and fittings.

The pressure drop, ΔP, in the ratio $\Delta P/P'_1$ which is used for the determination of Y from the charts on page A-23, is the measured difference between the inlet pressure and the pressure in the area of larger cross section. In a system discharging compressible fluids to atmosphere, this ΔP is equal to the inlet gauge pressure, or the difference between absolute inlet pressure and atmospheric pressure. This value of ΔP is also used in Equation 1-43, whenever the Y factor falls within the limits defined by the resistance factor K curves in the charts on page A-23. When the ratio of $\Delta P/P'_1$, using ΔP as defined above, falls beyond the limits of the K curves in the charts, sonic velocity occurs at the point of discharge or at some restriction within the pipe, and the limiting values for Y and ΔP, as determined from the tabulations to the right of the charts on page A-23, must be used in Equation 1-43.

Application of Equation 1-43 and the determination of values for K, Y, and ΔP in the equation is demonstrated in Examples 7-20 through 7-22.

The charts on page A-23 are based upon the general gas laws for perfect gases and, at sonic velocity conditions at the outlet end, will yield accurate results for all gases which can be approximated as a perfect gas. An Example of this type of flow problem is presented in Example 7-20.

This condition of flow is comparable to the flow through nozzles and venturi tubes, covered on page 4-6, and the solutions of such problems are similar.

Simple Compressible Flows: The complete isothermal equation like most compressible flow equations are developed from a generalized model with particular assumptions made. Another typical set of assumptions applied to the generalized model are commonly referred to as the simple compressible flows. Some of which were utilized as the basis for the development of the modified Darcy equation (Equation 1-43) and the net expansion factor. The simple flow analysis considers four driving potentials that affect the nature of the flow; simple area change (isentropic), simple friction (Fanno), simple heat addition or removal (Rayleigh), and simple mass addition or removal. Only one driving potential is considered at a time. For example, simple area change flow is adiabatic and reversible (no friction), with no mass addition or removal. The simple compressible flows are often employed because they adequately model many important real flows with reasonable engineering accuracy.[9] Development of the equations for simple flows is outside the scope of this paper; however the references cited describe them in detail.[9,15]

Software solutions to compressible flow problems: The equations presented in this section can practically be solved by hand. However, there are many compressible flow problems that require more thorough treatment and cannot be reasonably solved without computational techniques. A number of commercial software applications are available for the analysis of such problems. Particular applications typically employ a subset of the common compressible flow techniques. Examples of types of such software packages include those for the gas pipeline industry that implement many of the previously discussed isothermal equations as well as packages that implement a rigorous solution of the simple compressible flows for general applications. It is recommended that one understands the assumptions behind a software program to ensure the applicability to their particular compressible flow problem.

Steam - General Discussion

Substances exist in any one of three phases: solid, liquid, or gas. When outside conditions are varied, they may change from one phase to another.

Water under normal atmospheric conditions exists in the form of a liquid. When a body of water is heated by means of some external medium, the temperature of the water rises and soon small bubbles, which break and form continuously, are noted on the surface. This phenomenon is described as "boiling."

The amount of heat necessary to cause the temperature of the water to rise is expressed in British Thermal Units (Btu), where 1 Btu is the quantity of heat required to raise the temperature of one pound of water from 60 to 61°F. The amount of heat necessary to raise the temperature of a pound of water from 32°F (freezing point) to 212°F (boiling point) is 180.1 Btu. When the pressure does not exceed 50 pounds per square inch absolute, it is usually permissible to assume that each temperature increase of 1°F represents a heat content increase of one Btu per pound, regardless of the temperature of the water.

Assuming the generally accepted reference plane for zero heat content at 32°F, one pound of water at 212°F contains 180.13 Btu. This quantity of heat is called *heat of the liquid or sensible heat* (h_f). In order to change the liquid into a vapor at atmospheric pressure (14.696 psia), 970.17 Btu must be added to each pound of water after the temperature of 212°F is reached. During this transition period, the temperature remains constant. The added quantity of heat is called the *latent heat of evaporation* (h_{fg}). Consequently, the *total heat of the vapor,* (h_g) formed when water boils at atmospheric pressure, is the sum of the two quantities, 180.13 Btu and 970.17 Btu, or 1150.3 Btu per pound.

If water is heated in a closed vessel not completely filled, the pressure will rise after steam begins to form accompanied by an increase in temperature.

Saturated steam is steam in contact with liquid water from which it was generated, at a temperature which is the boiling point of the water and the condensing point of the steam. It may be either "dry" or "wet," depending on the generating conditions. "Dry" saturated steam is steam free from mechanically mixed water particles. "Wet" saturated steam, on the other hand, contains water particles in suspension. Saturated steam at any pressure has a definite temperature.

Superheated steam is steam at any given pressure which is heated to a temperature higher than the temperature of saturated steam at that pressure.

Values of h_f, h_{fg} and h_g are tabulated in Appendix A on pages A-12 to A-16 for saturated steam and saturated water over a pressure range of 0.08859 to 3200.11 psia. Total heat for superheated steam is tabulated on pages A-17 to A-20 from 15 to 15000 psia.

Chapter 2

Flow of Fluids Through Valves and Fittings

The preceding chapter has been devoted to the theory and formulas used in the study of fluid flow in pipes. Since industrial installations usually contain a considerable number of valves and fittings, a knowledge of their resistance to the flow of fluids is necessary to determine the flow characteristics of a complete piping system.

Many texts on hydraulics contain no information on the resistance of valves and fittings to flow, while others present only a limited discussion of the subject. In realization of the need for more complete detailed information on the resistance of valves and fittings to flow, Crane Co. has conducted extensive tests in their Engineering Laboratories and has also sponsored investigations in other laboratories. These tests have been supplemented by a thorough study of all published data on this subject. Appendix A contains data from these many separate tests and the findings have been combined to furnish a basis for calculating the pressure drop through valves and fittings.

Representative resistances to flow of various types of piping components are given in the K Factor Table; see pages A-27 through A-30.

A discussion of the equivalent length and resistance coefficient K, as well as the flow coefficient C_v methods of calculating pressure drop through valves and fittings is presented on pages 2-7 to 2-10.

Types of Valves and Fittings Used in Piping Systems

Valves: The great variety of valve designs precludes any thorough classification.

If valves were classified according to the resistance they offer to flow, those exhibiting a straight-through flow path such as gate, ball, plug, and butterfly valves would fall in the low resistance class, and those having a change in flow path direction such as globe and angle valves would fall in the high resistance class.

For illustrations of some of the most commonly used valve designs, refer to pages 2-18 to 2-20. For line illustrations of typical fittings and pipe bends, as well as valves, see pages A-27 to A-30.

Fittings: Fittings may be classified as branching, reducing, expanding, or deflecting. Such fittings as tees, crosses, side outlet elbows, etc., may be called branching fittings.

Reducing or expanding fittings are those which change the area of the fluid passageway. In this class are reducers and bushings. Deflecting fittings bends, elbows, return bends, etc., are those which change the direction of flow.

Some fittings may be combinations of any of the foregoing general classifications. In addition, there are types such as couplings and unions which offer no appreciable resistance to flow and therefore, need not be considered here.

Pressure Drop Attributed To Valves and Fittings

When a fluid is flowing steadily in a long straight pipe of uniform diameter, the flow pattern, as indicated by the velocity distribution across the pipe diameter, will assume a certain characteristic form. Any impediment in the pipe which changes the direction of the whole stream, or even part of it, will alter the characteristic flow pattern and create turbulence, causing an energy loss greater than that normally accompanying flow in straight pipe. Because valves and fittings in a pipeline disturb the flow pattern, they produce an additional pressure drop.

The loss of pressure produced by a valve (or fitting) consists of:

1. The pressure drop within the valve itself.
2. The pressure drop in the upstream piping in excess of that which would normally occur if there were no valve in the line. This effect is small.
3. The pressure drop in the downstream piping in excess of that which would normally occur if there were no valve in the line. This effect may be comparatively large.

From the experimental point of view it is difficult to measure the three items separately. Their combined effect is the desired quantity however, and this can be accurately measured by well known methods.

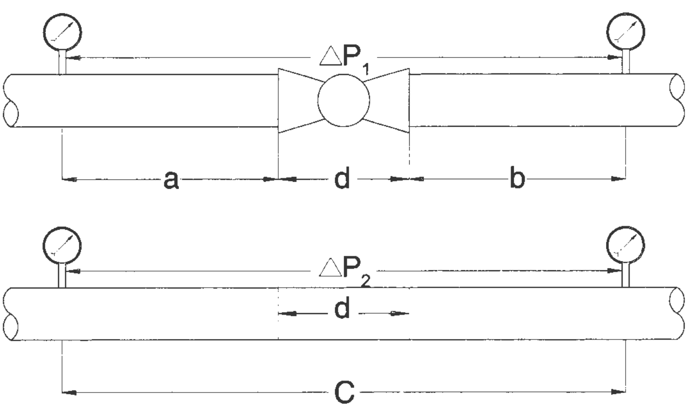


Figure 2-1

Figure 2-1 shows two sections of a pipeline of the same diameter and length. The upper section contains a globe valve. If the pressure drops, ΔP_1 and ΔP_2, were measured between the points indicated, it would be found that ΔP_1 is greater than ΔP_2.

Actually, the loss chargeable to a valve of length "d" is ΔP_1 minus the loss in a section of pipe of length "a + b." The losses, expressed in terms of resistance coefficient K of various valves and fittings as given on pages A-27 to A-30 include the loss due to the length of the valve or fitting.

Crane Flow Tests

Crane Engineering Laboratories have facilities for conducting water, steam, and air flow tests for many sizes and types of valves and fittings. Although a detailed discussion of all the various tests performed is beyond the scope of this paper, a brief description of some of the apparatus will be of interest.

The test piping shown in Figure 2-2 is unique in that 6 inch gate, globe, and angle valves or 90 degree elbows and tees can be tested with either water or steam. The vertical leg of the angle test section permits testing of angle lift check and stop check valves.

Saturated steam at 150 psi is available at flow rates up to 100,000 pounds/hour. The steam is throttled to the desired pressure and its state is determined at the meter as well as upstream and downstream from the test specimen.

For tests on water, a steam-turbine driven pump supplies water at rates up to 1200 gallons per minute through the test piping.

Static pressure differential is measured by means of a manometer connected to piezometer rings upstream and downstream from test position 1 in the angle test section, or test position 2 in the straight test section. The downstream piezometer for the angle test section serves as the upstream piezometer for the straight test section. Measured pressure drop for the pipe alone between piezometer stations is subtracted from the pressure drop through the valve plus pipe to ascertain the pressure drop chargeable to the valve alone.

Results of some of the flow tests conducted in the Crane Engineering Laboratories are plotted in Figures 2-3 to 2-6 shown on the two pages following.

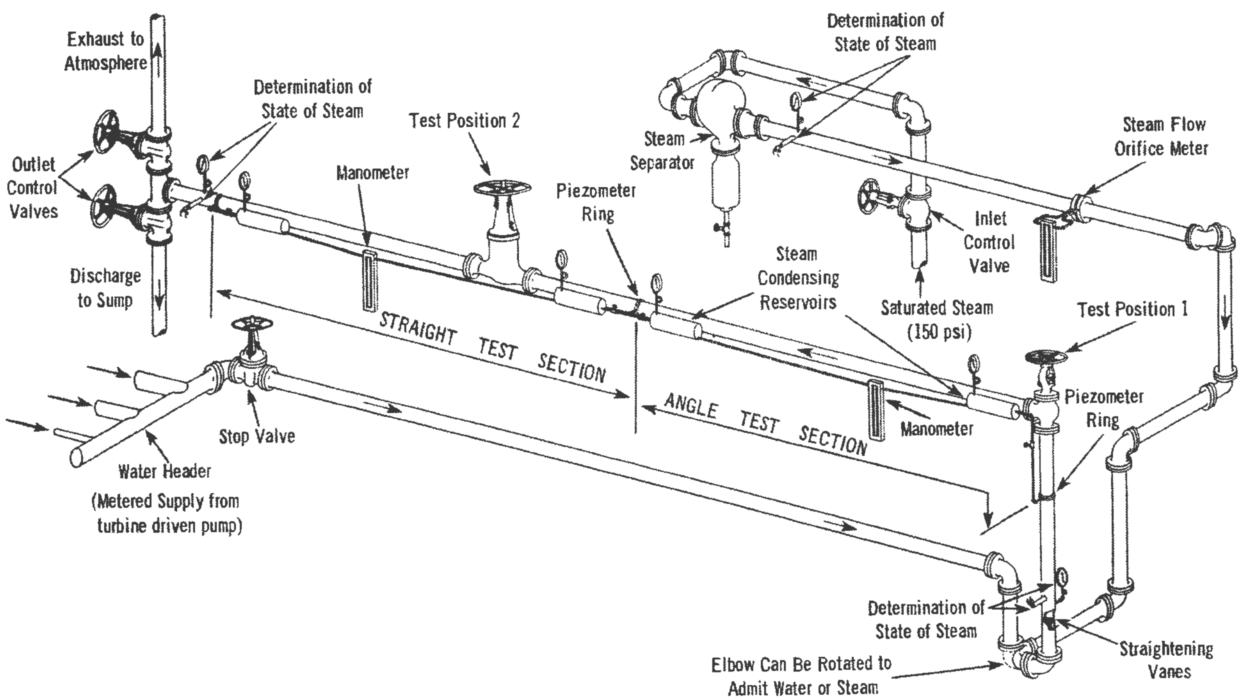


Figure 2-2: Test piping apparatus for measuring the pressure drop through valves and fittings on steam or water lines

Crane Water Flow Tests

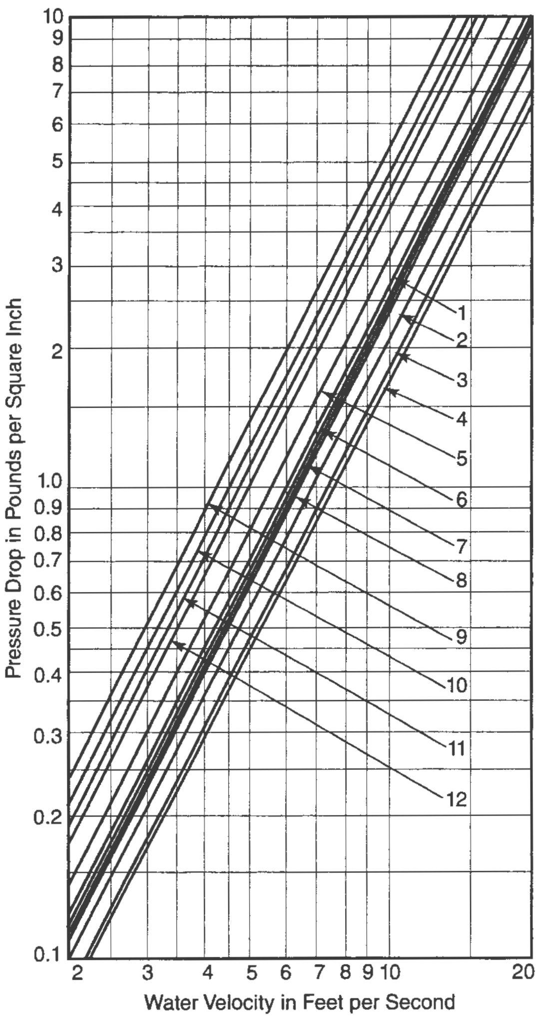


Figure 2-3: Water Flow Test - Curves 1-12

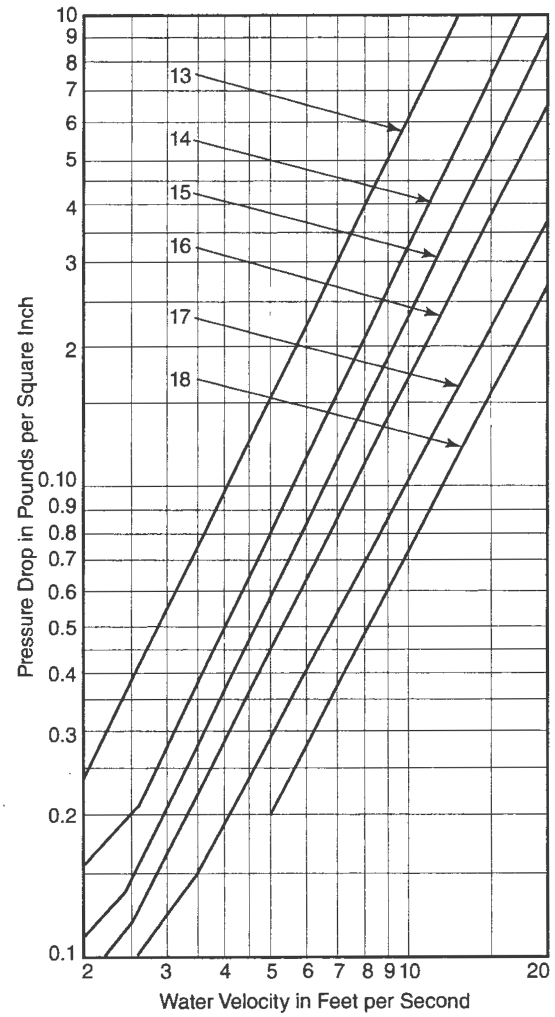


Figure 2-4: Water Flow Test - Curves 13-18

Water Flow Tests - Curves 1 to 18

Fluid	Figure No.	Curve No.	Size, Inches	Valve Type*
Water	Figure 2-3	1 2 3 4	¾ 2 4 6	Class 150 Cast Iron Y-Pattern Globe Valve, Flat Seat
		5 6 7 8	1 ½ 2 2 ½ 3	Class 150 Brass Angle Valve with Composition Disc, Flat Seat
		9 10 11 12	1 ½ 2 2 ½ 3	Class 150 Brass Conventional Globe Valve With Composition Disc, Flat Seat
	Figure 2-4	13 14 15 16 17	⅜ ½ ¾ 1¼ 2	Class 200 Brass Swing Check Valve
		18	6	Class 125 Iron Body Swing Check Valve

*Except for check valves at lower velocities where curves (14 and 17) bend, all valves were tested with disc fully lifted.

Crane Steam Flow Tests

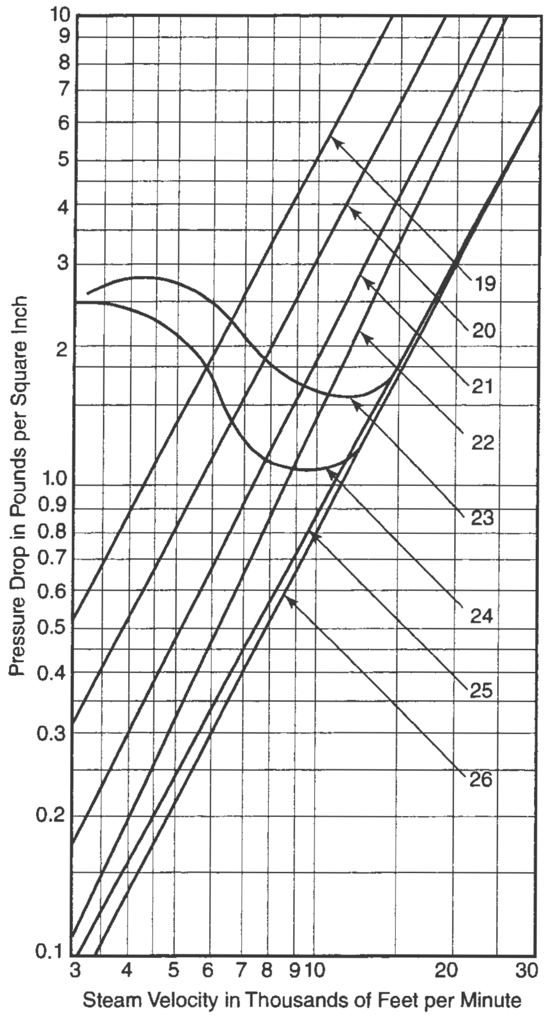


Figure 2-5: Steam Flow Test - Curves 19-26

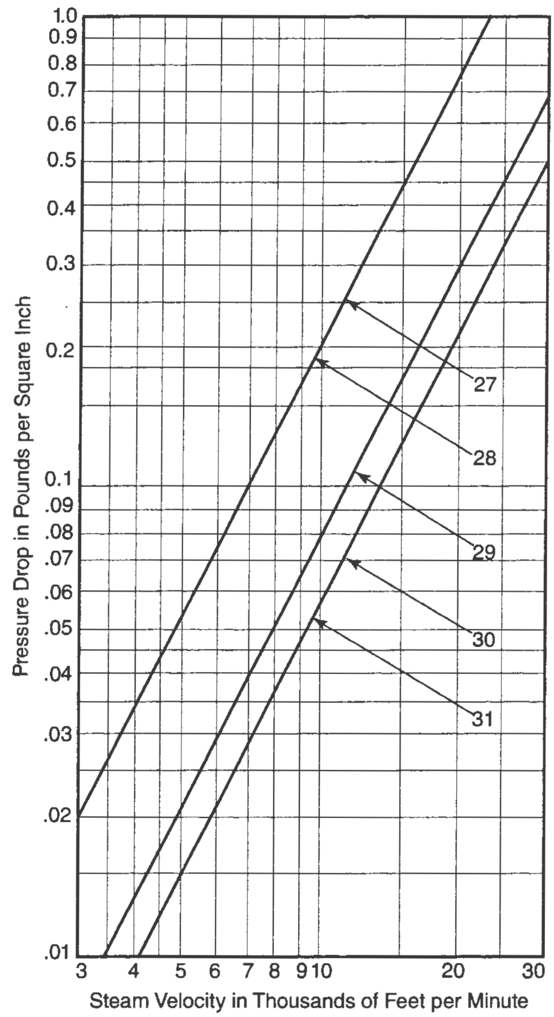


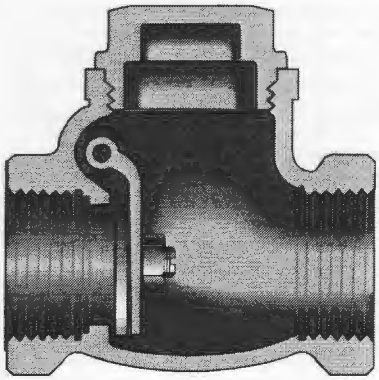
Figure 2-6: Steam Flow Test - Curves 27-31

Steam Flow Tests - Curves 19 to 31

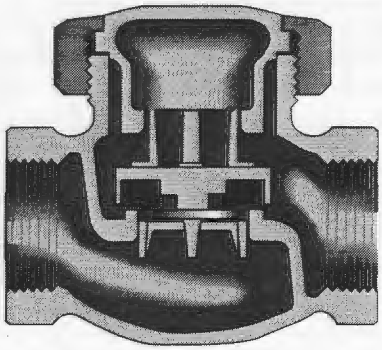
Fluid	Figure No.	Curve No.	Size, Inches	Valve* or Fitting Type
Saturated Steam 50 psi gauge	Figure 2-5	19	2	Class 300 Brass Conventional Globe Valve, Plug Type Seat
		20	6	Class 300 Steel Conventional Globe Valve, Plug Type Seat
		21	6	Class 300 Steel Angle Valve, Plug Type Seat
		22	6	Class 300 Steel Angle Valve, Ball to Cone Seat
		23	6	Class 600 Steel Angle Stop-Check Valve
		24	6	Class 600 Steel Y-Pattern Globe Stop-Check Valve
		26	6	Class 600 Steel Angle Valve
		26	6	Class 600 Steel Y-Pattern Globe Valve
	Figure 2-6	27	2	90° Short Radius Elbow for Use with Schedule 40 Pipe
		28	6	Class 250 Cast Iron Flanged Conventional 90° Elbow
		29	6	Class 600 Steel Gate Valve
		30	6	Class 125 Cast Iron Gate Valve
		31	6	Class 150 Steel Gate Valve

*Except for check valves at lower velocities where curves (23 and 24) bend, all valves were tested with disc fully lifted.

Relationship of Pressure Drop to Velocity of Flow



Swing Check Valve



Lift Check Valve

Figure 2-7 Check Valves

Many experiments have shown that the head loss due to valves and fittings is proportional to a constant power of the velocity. When pressure drop or head loss is plotted against velocity on logarithmic coordinates, the resulting curve is therefore a straight line. In the turbulent flow range, the value of the exponent of velocity has been found to vary from about 1.8 to 2.1 for different designs of valves and fittings. However, for all practical purposes, it can be assumed that the pressure drop or head loss due to the flow of fluids in the turbulent range through valves and fittings varies as the square of the velocity.

This relationship of pressure drop to velocity of flow is valid for check valves, only if there is sufficient flow to hold the disc in a wide open position. The point of deviation of the test curves from a straight line, as illustrated in Figures 2-4 and 2-5, defines the flow conditions necessary to support a check valve disc in the wide open position.

Most of the difficulties encountered with check valves, both lift and swing types, have been found to be due to oversizing which results in noisy operation and premature wear of the moving parts.

Referring again to Figure 2-5, it will be noted that the velocity of 50 psig saturated steam, at the point where the two curves deviate from a straight line, is about 14,000 to 15,000 feet per minute. Lower velocities are not sufficient to lift the disc through its full stroke and hold it in a stable position against the stops, and can actually result in an increase in pressure drop as indicated by the curves. Under these conditions, the disc fluctuates with each minor flow pulsation, causing noisy operation and rapid wear of the contacting moving parts.

The minimum velocity required to lift the disc to the full-open and stable position has been determined by tests for numerous types of check and foot valves, and is given in the K Factor Table (see pages A-27 through A-30). It is expressed in terms of a constant times the square root of the specific volume of the fluid being handled, making it applicable for use with any fluid.

Sizing of check valves in accordance with the specified minimum velocity for full disc lift will often result in valves smaller in size than the pipe in which they are installed; however, the actual pressure drop will be little, if any, higher than that of a full size valve which is used in other than the wide-open position. The advantages are longer valve life and quieter operation. The losses due to sudden or gradual contraction and enlargement which will occur in such installations with bushings, reducing flanges, or tapered reducers can be readily calculated from the data given in the K Factor Table.

Resistance Coefficient K, Equivalent Length L/D, and Flow Coefficient C_v

Hydraulic resistance: Test data for the pressure drop (head loss) across a wide variety of valves and fittings are available from the work of numerous investigators, and this data can be used to characterize the resistance to flow offered by the valve or fitting. The equivalent length, resistance coefficient, and flow coefficient are the three most common methods used to characterize valve and fitting hydraulic performance.

Extensive studies in this field have been conducted by Crane Laboratories. However, due to the time-consuming and costly nature of such testing, it is virtually impossible to obtain test data for every size and type of valve and fitting. It is therefore desirable to provide a means of reliably extrapolating available test information to envelope those items which have not been or cannot readily be tested. However, whenever actual test data is available from a manufacturer, that data should be used in hydraulic calculations.

Resistance Coefficient K, Equivalent Length L/D, and Flow Coefficient C_v

Causes of head loss in valves and fittings: Head loss in a piping system results from a number of system characteristics, which may be categorized as follows:

1. Changes in the direction of the flow path.
2. Obstructions in the flow path.
3. Sudden or gradual changes in the cross-section and shape of the flow path.
4. Friction, which is a function of the surface roughness of the interior surfaces, the inside diameter, and the fluid velocity, density and viscosity.

In the zone of complete turbulence for most valves and fittings, the losses due to friction resulting from the actual length of the flow path are minor compared to those due to one or more of the other three categories listed. Therefore, the hydraulic resistance is considered independent of the friction factor, and hence the Reynolds number, and may be treated as a constant for any given obstruction under flow conditions other than laminar flow. How this resistance varies with laminar flow conditions is discussed later in this section.

Equivalent Length: A common method to characterize the hydraulic resistance of valves and fittings is the equivalent length ratio, or L/D. This method calculates an equivalent length, in pipe diameters of straight pipe, that will cause the same pressure drop as the obstruction under the same flow conditions in the attached pipeline. The head loss in straight pipe is expressed by the Darcy equation,

$$h_L = f \frac{L}{D} \frac{v^2}{2g}$$ **Equation 2-1**

Using the L/D ratio for a valve or fitting and the diameter of the pipe in which the fluid velocity occurs, the equivalent length of pipe is calculated and added to the actual length of straight pipe. The friction factor for the pipe at the calculated Reynolds number is determined and applied to the total length to determine the overall head loss or pressure drop for the pipe, valves, and fittings. The K Factor Table on pages A-27 to A-30 show the equivalent length of clean commercial steel pipe for various valves and fittings as a constant that is multiplied by the completely turbulent friction factor.

Resistance Coefficient: At any point of flow in a piping system, the total fluid energy is given by the Bernoulli theorem as described in Chapter 1, and is the sum of the elevation head, static pressure head, and velocity head. Velocity in a pipe is obtained at the expense of static pressure head and the amount of energy in the form of velocity head, in units of feet of fluid, is:

$$\text{Velocity Head} = \frac{v^2}{2g}$$ **Equation 2-2**

Head loss caused by fluid flow through a valve or fitting also causes a reduction in the static pressure head and is seen as a pressure drop across the device. Since the pressure drop varies proportionally with the square of the velocity in the zone of complete turbulence as discussed earlier, the head loss may be expressed in terms of the velocity head using a dimensionless resistance coefficient K in the equation:

$$h_L = K \frac{v^2}{2g}$$ **Equation 2-3**

The resistance coefficient can be thought of as the number of velocity heads lost due to a valve or fitting and has been shown to be constant for flow in the completely turbulent region. The value of K is an expression of the hydraulic resistance in reference to the diameter of the pipeline in which the velocity occurs.

Comparing the Darcy equation to Equation 2-3, it follows that the hydraulic resistance of a straight pipe can be expressed in terms of a resistance coefficient using the friction factor, pipe length, and pipe diameter by,

$$K = f \frac{L}{D}$$ **Equation 2-4**

Resistance coefficients for pipelines, valves and fittings in series and parallel: Often it is not important to determine the head loss or differential pressure across individual pipelines, valves, or fittings, but instead to determine the overall pressure drop or head loss across a set of components. For components in series, the equivalent total resistance coefficient can be calculated as the sum of the individual resistance of each component:

$$K_{TOTAL} = K_1 + K_2 + K_3 + \ldots K_n$$ **Equation 2-5**

For components in parallel, the inverse of the equivalent total resistance coefficient can be calculated as the sum of the inverses of the individual resistance of each component:

$$\frac{1}{K_{TOTAL}} = \frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3} + \cdots \frac{1}{K_n}$$ **Equation 2-6**

Resistance coefficients for geometrically dissimilar valves and fittings: The resistance coefficient K would theoretically be a constant for all sizes of a given design or line of valves and fittings if all sizes were geometrically similar. However, geometric similarity is seldom, if ever, achieved because the design of valves and fittings is dictated by manufacturing economies, standards, structural strength, and other considerations.

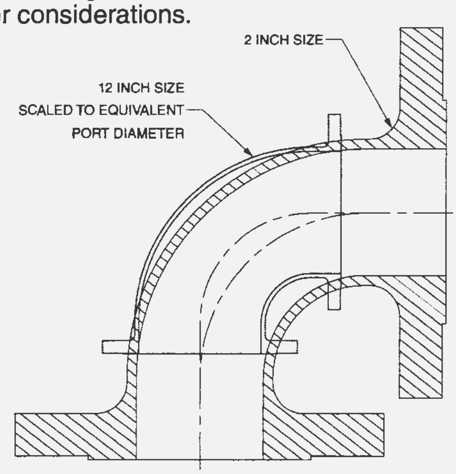


Figure 2-8: Geometric dissimilarity between 2 and 12 inch standard cast iron flanged elbows

An example of geometric dissimilarity is shown in Figure 2-8 where a 12 inch standard elbow has been drawn to scale and over-laid onto a 2 inch standard elbow, so that their port diameters are identical. The flow paths through the two fittings drawn to these scales would also have to be identical to have geometric similarity; in addition, the relative roughness of the surfaces would have to be similar.

Resistance Coefficient K, Equivalent Length L/D, and Flow Coefficient C_v

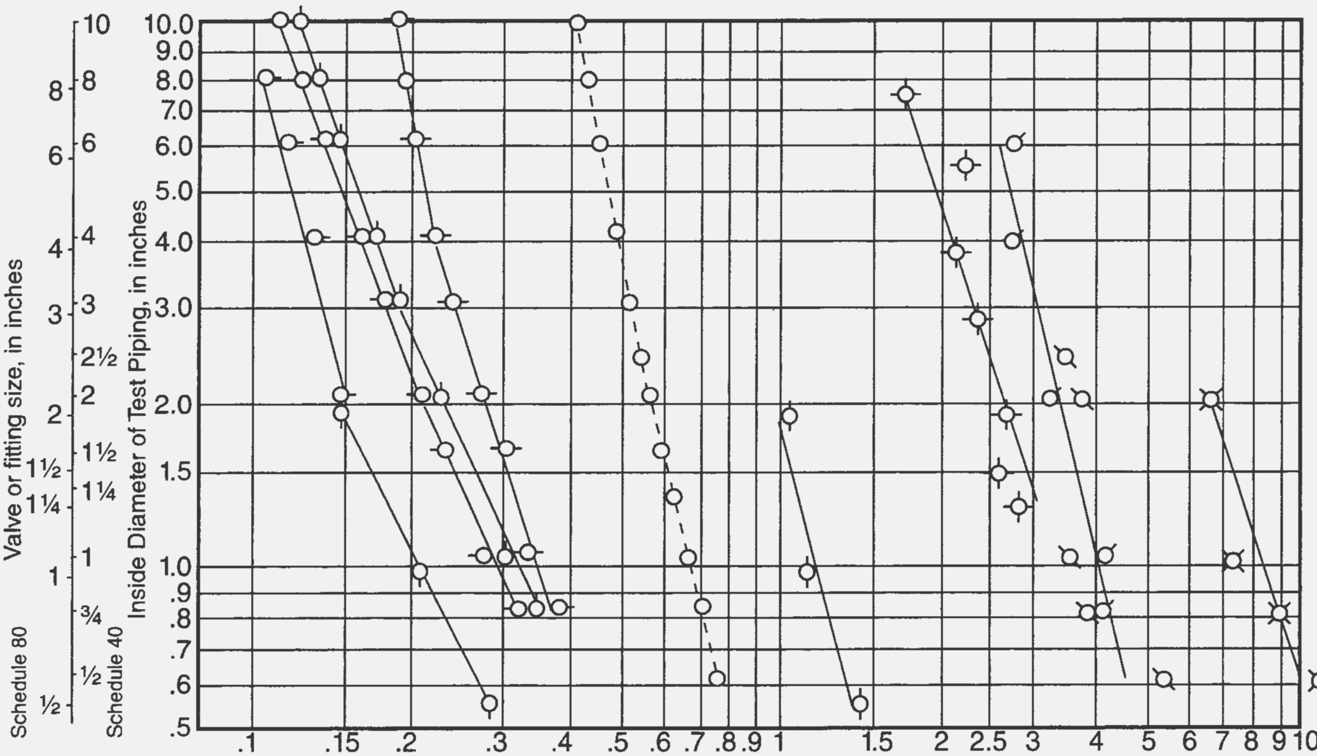


Figure 2-9: Variations of Resistance Coefficient K (= f_T L/D) with Size

Symbol	Product Tested	Authority
○	Schedule 40 Pipe, 30 Diameters Long (K = 30 f_T)*	Moody A.S.M.E. Trans., Nov.-1944[2]
○-	Class 125 Iron Body Wedge Gate Valves	Univ. of Wisc. Exp. Sta. Bull., Vol. 9, No. 1, 1922[16]
⚲	Class 600 Steel Wedge Gate Valves	Crane Tests
-○	90 Degree Pipe Bends, R/D = 2	Pigott A.S.M.E. Trans., 1950[17]
♁	90 Degree Pipe Bends, R/D = 3	Pigott A.S.M.E. Trans., 1950[17]
-○-	90 Degree Pipe Bends,R/D = 1	Pigott A.S.M.E. Trans., 1950[17]
⦶	Class 600 Steel Wedge Gate Valves, Seat Reduced	Crane Tests
-⦶-	Class 300 Steel Venturi Ball-Cage Gate Valves	Crane-Armour Tests
○́	Class 125 Iron Body Y-Pattern Globe Valves	Crane-Armour Tests
○̀	Class 125 Brass Angle Valves, Composition Disc	Crane Tests
¤	Class 125 Brass Globe Valves, Composition Disc	Crane Tests

*f_T = friction factor for flow in the zone of complete turbulence: see page A-27.

Figure 2-9 is based on the analysis of extensive test data from various sources on valves and fittings that are geometrically dissimilar. The resistance coefficients for a number of lines of valves and fittings have been plotted against size, along with the equivalent K for a 30 diameter long straight clean commercial steel pipe with flow conditions in the completely turbulent region, resulting in a constant completely turbulent friction factor. It will be noted that the resistance curves show a definite tendency to follow the same slope as the f_T(L/D) curve for straight clean commercial schedule 40 steel pipe. It appears that the effect of geometric dissimilarity between different sizes of the same line of valves or fittings upon the resistance coefficient is similar to that of relative roughness, or size of pipe, upon friction factor.

Based on the trends shown in Figure 2-9, it can be said that the resistance coefficient, for a given line of valves or fittings, tends to vary with size as does the friction factor for straight clean commercial schedule 40 steel pipe at flow conditions resulting in a constant friction factor, and that the equivalent length ratio L/D tends toward a constant for the various sizes of a given line of valves or fittings at the same flow conditions.

Resistance Coefficient K, Equivalent Length L/D, and Flow Coefficient C_v

This correlation with straight clean commercial steel pipe can be used to standardize the calculation of the resistance coefficient to take into account different sizes of the same valve or fitting, using the completely turbulent friction factor for clean commercial schedule 40 steel pipe as the scaling method. Equation 2-4 can be re-written with K as a function of the turbulent friction factor, f_T:

$$K = f_T \frac{L}{D} \qquad \textbf{Equation 2-7}$$

On the basis of this relationship, the resistance coefficient K for each illustrated type of valve and fitting is presented on pages A-27 through A-30. The resistance coefficient is shown as the product of the completely turbulent friction factor for the desired size of clean commercial schedule 40 steel pipe, and a constant which represents the equivalent length ratio L/D. This equivalent length is valid for all sizes of the valve or fitting type with which it is identified.

The friction factor for clean commercial schedule 40 steel pipe with flow in the zone of complete turbulence (f_T), for nominal sizes from ½ to 36 inches, are tabulated at the beginning of the K Factor Table page A-27 for convenience in converting the algebraic expressions of K to arithmetic quantities. The turbulent friction factor f_T can be calculated with a form of the Colebrook equation with ε = 0.00015 feet using Equation 2-8:

$$f_T = \frac{0.25}{\left[\log\left(\frac{\varepsilon/D}{3.7}\right)\right]^2} \qquad \textbf{Equation 2-8}$$

Geometrically similar fittings: There are some resistances to flow in piping, such as sudden and gradual contractions and enlargements, and pipe entrances and exits, that have geometric similarity between sizes. The resistance coefficients for these items are therefore independent of size, as indicated by the absence of a friction factor in their values given in the K Factor Table.

Adjusting K for pipe schedule: As previously stated, the resistance coefficient is associated with the diameter of pipe in which the velocity in the term $v^2/2g$ occurs. The values in the K Factor Table are associated with the internal diameter of the following pairing of pipe schedule numbers with the various ASME Classes of valves and fittings.

Class 300 and lower	Schedule 40
Class 400 and 600	Schedule 80
Class 900	Schedule 120
Class 1500	Schedule 160
Class 2500 (sizes ½ to 6 inch)	XXS
Class 2500 (sizes 8 inch and up)	Schedule 160

When the resistance coefficient is used in flow Equation 2-3 or any of its equivalent forms (Equations 6-20, 22, 23, 27 or 28), the velocity and internal diameter used in the equation must be based on the dimensions of these schedule numbers regardless of the pipe with which the valve may be installed. For example, if a 4 inch Class 900 valve is installed in a 4 inch schedule 80 pipe, the actual fluid velocity in the schedule 80 pipe will have to be converted into an "equivalent schedule 120 velocity" such that the flow rates would be the same, using the area ratio to calculate the velocity. This "equivalent schedule 120 velocity" and the inside diameter of schedule 120 pipe can then be used with the resistance coefficient from the K Factor Table in the head loss equation.

An alternate procedure which yields identical results is to adjust the K value obtained from the K Factor Table in proportion to the fourth power of the diameter ratio using Equation 2-9, and to use the values of velocity and internal diameter of the actual connecting pipe in the head loss equation.

$$K_a = K_b \left(\frac{d_a}{d_b}\right)^4 \qquad \textbf{Equation 2-9}$$

Subscript "a" defines K and d with reference to the internal diameter of the actual connecting pipe.

Subscript "b" defines K and d with reference to the internal diameter the pipe for which the values of K were established, as given in the foregoing list of pipe schedule numbers.

When a piping system contains more than one size of pipe, valves, or fittings, Equation 2-9 may be used to express all resistances in terms of one size. For this case, subscript "a" relates to the size with reference to which all resistances are to be expressed, and subscript "b" relates to any other size in the system. For sample problem, see Example 7-14.

Flow coefficient (C_v): It has been found convenient in some branches of the valve industry, particularly in connection with control valves, to express the valve capacity and the valve flow characteristics in terms of the flow coefficient C_v. K_v is the metric equivalent of C_v. Equation 3-17 can be used for conversion between them. The application of C_v to control valves is discussed in detail in Chapter 3. The flow coefficient is also used to characterize the hydraulic performance of other components such as strainers, nozzles, and sprinklers.

The flow coefficient is defined as the amount of water flow at 60°F, in gallons per minute, at a pressure drop of one pound per square inch across a component. It can be applied to fluids other than water using specific gravity and calculated with Equation 2-10.

$$C_v = \frac{Q}{\sqrt{\frac{\Delta P}{S}}} \qquad \textbf{Equation 2-10}$$

By substitution of appropriate equivalent units in the Darcy equation, it can be shown that the flow and resistance coefficients are related by:

$$C_v = \frac{29.84 d^2}{\sqrt{K}} \qquad \textbf{Equation 2-11}$$

Also, the quantity in gallons per minute of liquids of low viscosity* that will flow through the component with a given C_v and pressure drop can be determined by rearranging Equation 2-10. Alternatively, given the flow rate and C_v, the pressure drop can then be computed.

Since Equations 2-3 and 2-10 are simply other forms of the Darcy equation, the limitations regarding their use for

*When handling highly viscous liquids determine flow rate or required valve C_v as described in the ANSI/ISA Standard.

compressible flow (explained on page 1-8) apply. Other convenient forms of Equations 2-3 and 2-10 in terms of commonly used units are presented on page 6-4.

Use of flow coefficient for piping and components: The flow coefficient can also be used to characterize the hydraulic performance of any valve, fitting, pipeline, or combination of fixed resistance components in a system. If the flow rate and differential pressure across the components are known, an equivalent C_v can be calculated, as shown in Figure 2-10 for the flow of water at 60°F.

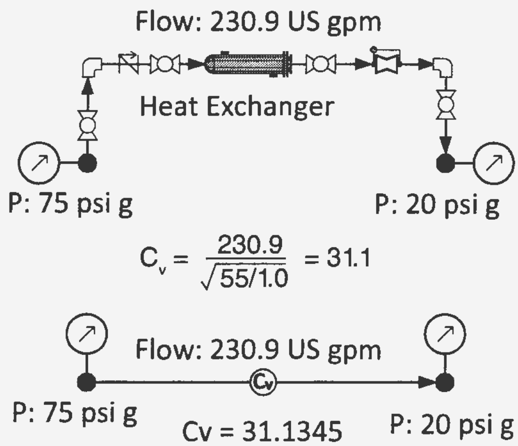


Figure 2-10: Equivalent C_v calculated for fixed resistance piping components with only flow rate and differential pressure known

Using the equivalent C_v, the flow rate at a different pressure drop can be calculated, or the pressure drop can be calculated for a given flow rate, assuming the resistance of all components remains fixed.

Flow coefficients for pipelines, valves and fittings in series and parallel: As with the resistance coefficient, an equivalent total flow coefficient can be calculated to represent the hydraulic performance of multiple piping components in series or parallel. For components in series, the equivalent total flow coefficient can be calculated as follows:

$$\frac{1}{C^2_{v_{TOTAL}}} = \frac{1}{C^2_{v_1}} + \frac{1}{C^2_{v_2}} + \cdots \frac{1}{C^2_{v_n}} \qquad \textbf{Equation 2-12}$$

For components in parallel, the equivalent total flow coefficient can be calculated as the sum of the individual flow coefficients of each component:

$$C_{v_{TOTAL}} = C_{v_1} + C_{v_2} + \ldots C_{v_n} \qquad \textbf{Equation 2-13}$$

Laminar Flow Conditions

Flow will change from laminar to turbulent, typically within a range of Reynolds numbers from 2000 to 4000, defined as the critical zone and illustrated on pages A-25 and A-26. The lower critical Reynolds number of 2000 is usually recognized as the upper limit for the application of Poiseuille's law for laminar flow in straight pipes,

$$h_L = 0.0962\left(\frac{\mu L v}{d^2 \rho}\right) \qquad \textbf{Equation 2-14}$$

which is identical to Equations 2-1 when the value of the friction factor for laminar flow, $f = 64/R_e$, is factored into it. Laminar flow at Reynolds numbers above 2000 is unstable, and in the critical zone and lower range of the transition zone, turbulent mixing and laminar motion may alternate unpredictably.

Equation 2-3 is valid for computing the head loss due to valves and fittings for all conditions of flow, including laminar flow, using the resistance coefficient. If the assumption is made that the resistance coefficient is constant for all flow conditions, K may be obtained from the K Factor Table which uses the turbulent friction factor for clean commercial steel pipe and the equivalent length ratio L/D.

When Equation 2-3 is used to determine the losses in straight pipe, it is necessary to compute the Reynolds number in order to establish the friction factor to be used to determine the value of the resistance coefficient for the pipe in accordance with Equation 2-4. See Examples 7-7 through 7-9.

Adjusting the resistance coefficient for Reynolds number: Recent studies suggest that for flow regimes other than completely turbulent, the frictional forces within a valve or fitting become more influential compared to the changes in direction, cross-sectional shape, or obstructions in the flow passage. This results in an increase in the resistance coefficient as the friction factor increases with decreasing Reynolds number in the transition and laminar regions, as shown in studies presented by Miller and Idelchik.[18,19]

Contraction and Enlargement

The resistance to flow due to sudden enlargements may be expressed by,

$$K_1 = \left(1 - \frac{d_1^2}{d_2^2}\right)^2 \qquad \textbf{Equation 2-15}$$

and the resistance due to sudden contractions, by

$$K_1 = 0.5\left(1 - \frac{d_1^2}{d_2^2}\right) \qquad \textbf{Equation 2-16}$$

Subscripts 1 and 2 define the internal diameters of the small and large pipes respectively.

It is convenient to identify the ratio of diameters of the small to large pipes by the Greek letter β (beta). Using this notation, these equations may be written,

Sudden Enlargement

$$K_1 = (1 - \beta^2)^2 \qquad \textbf{Equation 2-17}$$

Sudden Contraction

$$K_1 = 0.5\,(1 - \beta^2) \qquad \textbf{Equation 2-18}$$

Equation 2-15 is derived from the momentum equation together with the Bernoulli equation. Equation 2-16 uses the derivation of Equation 2-15 together with the continuity equation and a close approximation of the contraction coefficients determined by Julius Weisbach.[20]

The value of the resistance coefficient in terms of the larger pipe is determined by dividing Equations 2-15 and 2-16 by β^4,

$$K_2 = \frac{K_1}{\beta^4} \qquad \textbf{Equation 2-19}$$

The losses due to gradual enlargements in pipes were investigated by A.H. Gibson,[21] and may be expressed as a coefficient, C_e, applied to Equation 2-15. Approximate averages of Gibson's coefficients for different included angles of divergence, θ, are defined by the equations:

If, $\theta \leq 45°$ $\qquad C_e = 2.6 \sin\frac{\theta}{2}$ $\qquad$ **Equation 2-20**

If, $45° < \theta \leq 180°$ $\qquad C_e = 1$ $\qquad$ **Equation 2-21**

The losses due to gradual contractions in pipes were established by the analysis of Crane test data, using the same basis as that of Gibson for gradual enlargements, to provide a contraction coefficient, C_c to be applied to Equation 2-16.

The approximate averages of these coefficients for different included angles of convergence, θ, are defined by the equations:

If, $\theta \leq 45°$ $\qquad C_c = 1.6 \sin\frac{\theta}{2}$ $\qquad$ **Equation 2-22**

If, $45° < \theta \leq 180°$ $\qquad C_c = \sqrt{\sin\frac{\theta}{2}}$ $\qquad$ **Equation 2-23**

The resistance coefficient K for sudden and gradual enlargements and contractions, expressed in terms of the large pipe, is established by combining equations 2-15 to 2-23 inclusive.

Sudden and Gradual Enlargements

Equation 2-24

$$\theta \leq 45° \qquad K_2 = \frac{2.6 \sin\frac{\theta}{2}\,(1 - \beta^2)^2}{\beta^4}$$

$$45° < \theta \leq 180° \qquad K_2 = \frac{(1 - \beta^2)^2}{\beta^4} \qquad \textbf{Equation 2-25}$$

Sudden and Gradual Contractions

Equation 2-26

$$\theta \leq 45° \qquad K_2 = \frac{0.8 \sin\frac{\theta}{2}\,(1 - \beta^2)}{\beta^4}$$

Equation 2-27

$$45° < \theta \leq 180° \qquad K_2 = \frac{0.5\sqrt{\sin\frac{\theta}{2}}\,(1 - \beta^2)}{\beta^4}$$

Valves with Reduced Seats

Valves are often designed with reduced seats, and the transition from seat to valve ends may be either abrupt or gradual. Straight-through types, such as gate and ball valves, so designed with gradual transitions are sometimes referred to as venturi valves. Analysis of tests on such straight-through valves indicates an excellent correlation between test results and calculated values of K based on the summation of Equations 2-19 and 2-24 through 2-27.

Valves which exhibit a change in direction of the flow path, such as globe and angle valves, are classified as high resistance valves. Equations 2-24 through 2-27 for gradual contractions and enlargements cannot be readily applied to those configurations because the angles of convergence and divergence are variable with respect to different planes of reference. The entrance and exit losses for reduced seat globe and angle valves are judged to fall short of those due to sudden expansion and contraction (Equations 2-25 and 2-27 at $\theta = 180°$) if the approaches to seat are gradual. Analysis of available test data indicates that the factor β applied to Equations 2-24 and 2-26 for sudden contraction and enlargement will bring calculated K values for reduced seat globe and angle valves into reasonably close agreement with test results. In the absence of actual test data, the resistance coefficients for reduced seat globe and angle valves may thus be computed as the summation of Equations 2-19 and β times Equations 2-25 and 2-27 at $\theta = 180°$.

The procedure for determining K for reduced seat globe and angle valves is also applicable to throttled globe and angle valves. For this case the value of β must be based upon the square root of the ratio of areas,

$$\beta = \sqrt{\frac{a_1}{a_2}} \qquad \textbf{Equation 2-28}$$

where:

a_1 = the area at the most restricted point in the flow path
a_2 = the internal area of the connecting pipe

Resistance of Bends

Secondary flow: The nature of the flow of liquids in bends has been thoroughly investigated and many interesting facts have been discovered. For example, when a fluid passes around a bend in either viscous or turbulent flow, there is established in the bend a condition known as "secondary flow." This is rotating motion, at right angles to the pipe axis, which is superimposed upon the main motion in the direction of the axis. The friction resistance of the pipe walls and the action of centrifugal force combine to produce this rotation. Figure 2-11 illustrates this phenomenon.

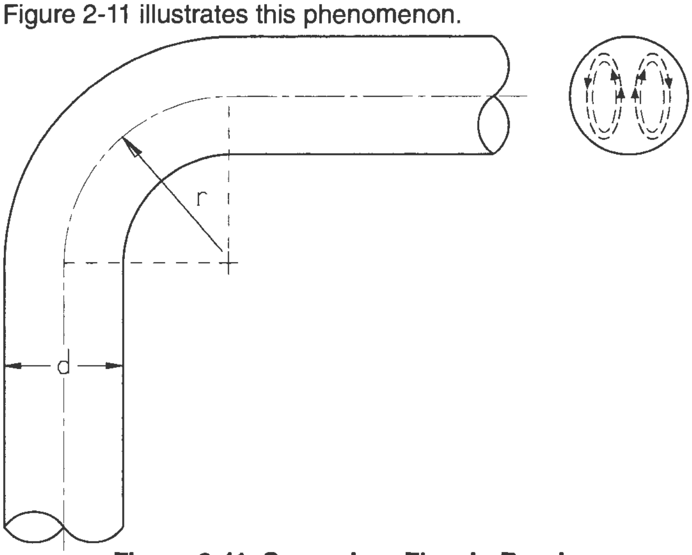


Figure 2-11: Secondary Flow in Bends

Resistance of bends to flow: The resistance or head loss in a bend is conventionally assumed to consist of (1) the loss due to curvature (2) the excess loss in the downstream tangent and (3) the loss due to length, thus:

$$h_t = h_p + h_c + h_L \qquad \textbf{Equation 2-29}$$

where:
h_t = total loss, in feet of fluid
h_p = excess loss in downstream tangent, in feet of fluid
h_c = loss due to curvature, in feet of fluid
h_L = loss in bend due to length, in feet of fluid

if:
$$h_b = h_p + h_c$$

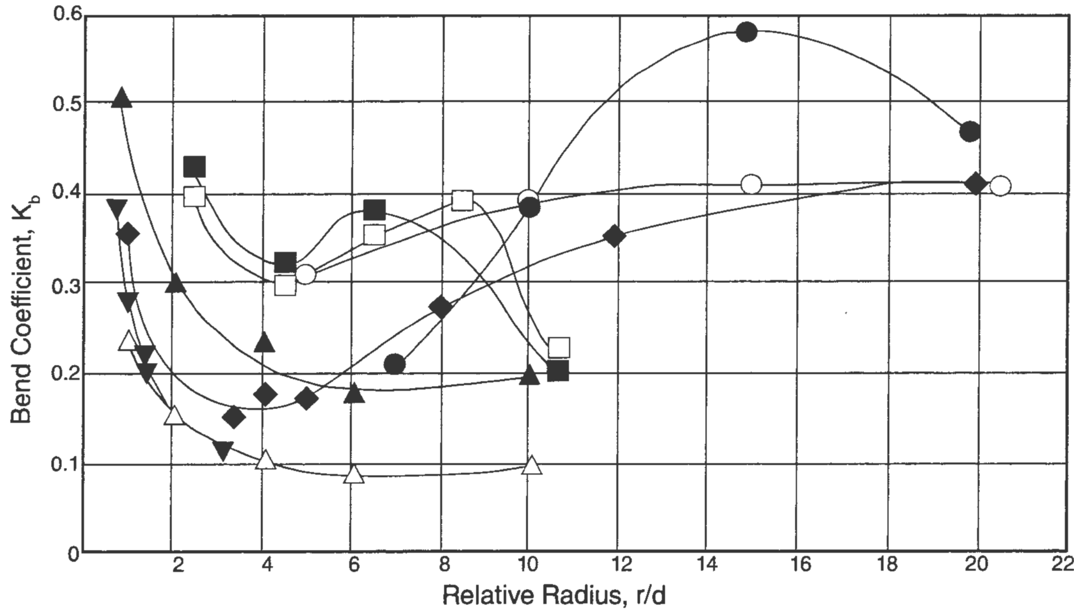
then:
$$h_t = h_b + h_L \qquad \textbf{Equation 2-30}$$

However, the quantity h_b can be expressed as a function of velocity head in the formula:

$$h_b = K_b \frac{v^2}{2g} \qquad \textbf{Equation 2-31}$$

where:
K_b = the bend coefficient

Resistance of Bends



Investigator	Diameter (inch)	Symbol
Balch	3	●
Davis	2	○
Brightmore	3	■
Brightmore	4	□
Hofmann	1.7 (rough pipe)	▲
Hofmann	1.7 (smooth pipe)	△
Vogel	6, 8 and 10	▼
Beij	4	◆

Figure 2-12: Bend Coefficients Found by Various Investigators[22]
From "Pressure Losses for Fluid Flow in 90° Pipe Bends" by K.H. Beij.
Courtesy of Journal of Research of National Bureau of Standards.

The relationship between K_b and r/d (relative radius*) is not well defined, as can be observed by reference to Figure 2-12 (taken from the work of Beij).[22] The curves in this chart indicate that K_b has a minimum value when r/d is between 3 and 5.

Values of K for 90 degree bends with various bend ratios (r/d) are listed on page A-30. The values (also based on the work of Beij) represent average conditions of flow in 90 degree bends.

The loss due to continuous bends greater than 90 degrees, such as pipe coils or expansion bends, is less than the summation of losses in the total number of 90 degree bends contained in the coil, considered separately, because the loss h_p in Equation 2-29 occurs only once in the coil.

The loss due to length in terms of K is equal to the developed length of the bend, in pipe diameters, multiplied by the friction factor f_T as previously described and as tabulated on page A-27.

$$K_{length} = 0.5\, f_T\, \pi\, (r/d) \qquad \textbf{Equation 2-32}$$

In the absence of experimental data, it is assumed that $h_p = h_c$ in Equation 2-29. On this basis, the total value of K for a pipe coil or expansion bend made up of continuous 90 degree bends can be determined by multiplying the number (n) of 90 degree bends less one contained in the coil by the value of K due to length, plus one-half of the value of K due to bend resistance, and adding the value of K for one 90 degree bend (page A-30).

Equation 2-33

$$K_B = (n - 1)(0.25\, f_T\, \pi\, r/d + 0.5\, K_1) + K_1$$

Subscript 1 defines the value of K (see page A-30) for one 90 degree bend.

Example:
A 2 inch Schedule 40 pipe coil contains five complete turns, i.e., twenty (n) 90 degree bends. The relative radius (r/d) of the bends is 16, and the resistance coefficient K_1 of one 90 degree bend is $42 f_T$ (42 x .019 = .80) per page A-30.

Find the total resistance coefficient (K_B) for the coil.

$$K_B = (20 - 1)\ (0.25 \times 0.019\pi \times 16 + 0.5 \times 0.8) + 0.8 = 13$$

Resistance of miter bends: The equivalent length of miter bends, based on the work of H. Kirchbach,[23] is also shown on page A-30.

*The relative radius of a bend is the ratio of the radius of the bend axis to the internal diameter of the pipe. Both dimensions must be in the same units.

Hydraulic Resistance of Tees and Wyes

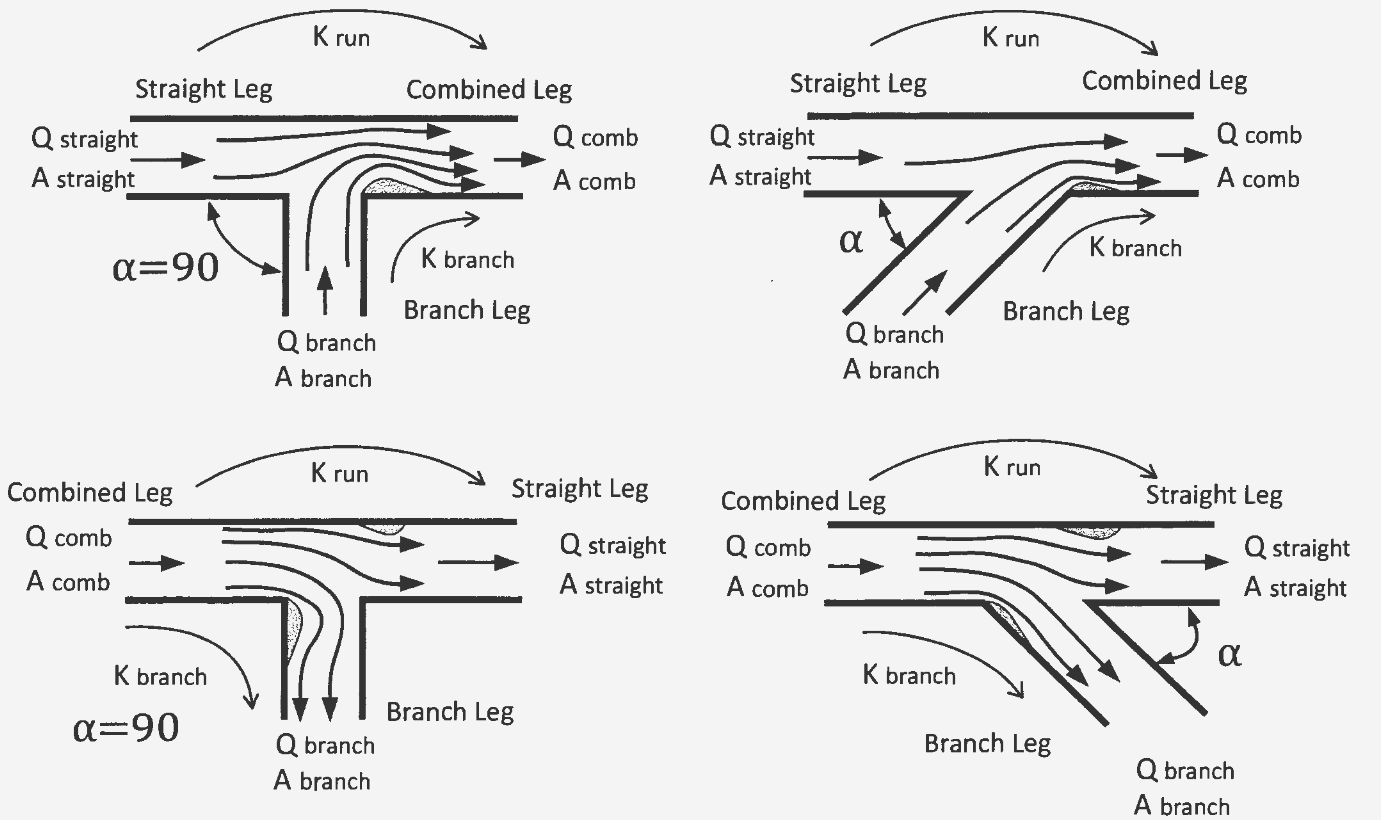


Figure 2-13: Converging flow (top) and diverging flow (bottom) through tees and wyes

Tees and wyes are employed in piping systems to either combine the flow of two streams or divide the flow of a single stream. The resistance imposed by these fittings is dependent on the variations in geometry and flow conditions in each flow path. For convention, the three legs that constitute a tee or wye are referred to as the combined, straight, and branch, as shown in Figure 2-13. The resistance across the two flow paths can be characterized using two distinct resistance coefficients, one which represents the resistance across the straight and combined legs (K_{run}), and one which represents the resistance across the branch and combined legs (K_{branch}). These resistances can be expressed in terms of the fluid velocity in any leg, but for the purposes of this paper, the resistances are expressed in terms of, and should be applied to, the fluid velocity in the combined leg.

The method used in previous versions of this paper treated K_{run} and K_{branch} as dependent only on the fitting size without regard for the effect of geometry or division of flow in the different paths. Further research has shown that the resistance coefficients depend on the cross sectional area ratios of the legs, the angle between the legs, the ratio of the flow rates, and whether the flows are converging or diverging.[18,19]

Extensive evaluation of the methods in these two references is presented here in a concise format for ease of use for the cases in which the straight leg and combined leg areas are equal. General equations using constants are given to calculate the resistance coefficients using the area and flow ratio between the branch and combined leg as well as the angle between the branch and straight legs. The constants used in the equations are tabulated and depend on the flow and area ratios.

Resistance coefficients for standard tees and wyes in which all the channels have equal cross sectional area (i.e., area ratio is equal to 1.0) are also graphically represented. Configurations for tees and wyes not represented by these equations or graphs can be found in other sources.[18,19]

It is convenient to express the area ratio in terms of the leg diameters with the use of the diameter ratio (β), such that for the branch area ratio can be defined with Equation 2-34:

$$\frac{A_{branch}}{A_{comb}} = \frac{d^2_{branch}}{d^2_{comb}} = \left(\frac{d_{branch}}{d_{comb}}\right)^2 = \beta^2_{branch} \qquad \textbf{Equation 2-34}$$

It should be noted that in some cases, the resistance coefficient may be negative, indicating that instead of a head loss there is a head gain as energy is added to the flow stream as a result of passing through the tee or wye. This occurs due to the slower moving fluid accelerating to the velocity of the total combined flow.

Hydraulic Resistance of Tees and Wyes

Converging flow: For converging flow in a tee or wye, Equation 2-35 applies for all area ratios, flow ratios, and branch angles shown in Table 2-1 for both the branch and run resistance coefficients, except for K_{run} for a 90 degree tee.

Constants for Equation 2-35 are shown in Table 2-1 and 2-2. Use the fluid velocity in the combined leg to calculate the head loss and pressure drop across both flow paths in the tee or wye.

To calculate K_{run} for 90 degree tees for any area ratio, use Equation 2-36. Graphs for K_{branch} and K_{run} for converging flow with an area ratio of one can be seen in Figures 2-14 and 2-15 respectively. See also Example 7-35.

$$K_{branch} \text{ or } K_{run} = C\left[1 + D\left(\frac{Q_{branch}}{Q_{comb}}\frac{1}{\beta^2_{branch}}\right)^2 - E\left(1 - \frac{Q_{branch}}{Q_{comb}}\right)^2 - F\frac{1}{\beta^2_{branch}}\left(\frac{Q_{branch}}{Q_{comb}}\right)^2\right] \qquad \textbf{Equation 2-35}$$

Table 2-1: Constants for Equation 2-35

	K_{branch}				K_{run}			
Angle	C	D	E	F	C	D	E	F
30	See Table 2-2	1	2	1.74	1	0	1	1.74
45		1	2	1.41	1	0	1	1.41
60		1	2	1	1	0	1	1
90		1	2	0	Use Equation 2-36			

$$K_{run} \cong 1.55\left(\frac{Q_{branch}}{Q_{comb}}\right) - \left(\frac{Q_{branch}}{Q_{comb}}\right)^2 \qquad \textbf{Equation 2-36}$$

Table 2-2: Values of C for Equation 2-35

		Q_{branch} / Q_{comb}	
		≤ 0.35	**> 0.35**
β^2_{branch}	**≤ 0.35**	C = 1	
	> 0.35	$C = 0.9\left(1 - \frac{Q_{branch}}{Q_{comb}}\right)$	C = 0.55

Diverging flow: For diverging flow in a tee or wye, Equation 2-37 applies for calculating the branch resistance coefficient of all area ratios, flow ratios, and branch angles shown in Table 2-3. Constants for Equation 2-37 are given in Table 2-3 and 2-4. Use the fluid velocity in the combined leg to calculate the head loss and pressure drop across both flow paths in the tee or wye.

Equation 2-37

$$K_{branch} = G\left[1 + H\left(\frac{Q_{branch}}{Q_{comb}}\frac{1}{\beta^2_{branch}}\right)^2 - J\left(\frac{Q_{branch}}{Q_{comb}}\frac{1}{\beta^2_{branch}}\right)\cos\alpha\right]$$

To calculate K_{run} for diverging flow in tees and wyes, use Equation 2-38 with the constants provided in Table 2-5. Graphs for K_{branch} and K_{run} for diverging flow with an area ratio of one can be seen in Figures 2-16 and 2-17 respectively. See also Example 7-36.

$$K_{run} = M\left(\frac{Q_{branch}}{Q_{comb}}\right)^2 \qquad \textbf{Equation 2-38}$$

Table 2-3: Constants for K_{branch} in Equation 2-37

Angle (α)	G	H	J
0 - 60°	Table 2-4	1	2
α = 90° at $\beta_{branch} \leq 2/3$	1	1	2
α = 90° at β_{branch} = 1*	$G = 1 + 0.3\left(\frac{Q_{branch}}{Q_{comb}}\right)^2$	0.3	0

* up to $\frac{Q_{branch}}{Q_{comb}}\frac{1}{\beta^2_{branch}} \cong 2$

Table 2-4: Values of G for Equation 2-37

		Q_{branch} / Q_{comb}	
		≤ 0.6	**> 0.6**
β^2_{branch}	**≤ 0.35**	$G = 1.1 - 0.7\frac{Q_{branch}}{Q_{comb}}$	G = 0.85
	> 0.35	$G = 1.0 - 0.6\frac{Q_{branch}}{Q_{comb}}$	G = 0.6
		≤ 0.4	**> 0.4**
		Q_{branch} / Q_{comb}	

Table 2-5: Values of M for Equation 2-38

		Q_{branch} / Q_{comb}	
		≤ 0.5	**> 0.5**
β^2_{branch}	**≤ 0.4**	M = 0.4	
	> 0.4	$M = 2\left(2\frac{Q_{branch}}{Q_{comb}} - 1\right)$	$M = 0.3\left(2\frac{Q_{branch}}{Q_{comb}} - 1\right)$

Hydraulic Resistance of Tees and Wyes

Graphical representation of K_{run} and K_{branch}: The previous equations for calculating the resistance coefficients for the branch and straight paths for converging and diverging flow in tees and wyes are applicable for the angles, area ratios, and flow ratios within the stated limitations. Figures 2-14 to 2-17 graphically show the resistance coefficients for the branch and run paths for tees and wyes with all leg diameters equal and are provided for convenience.

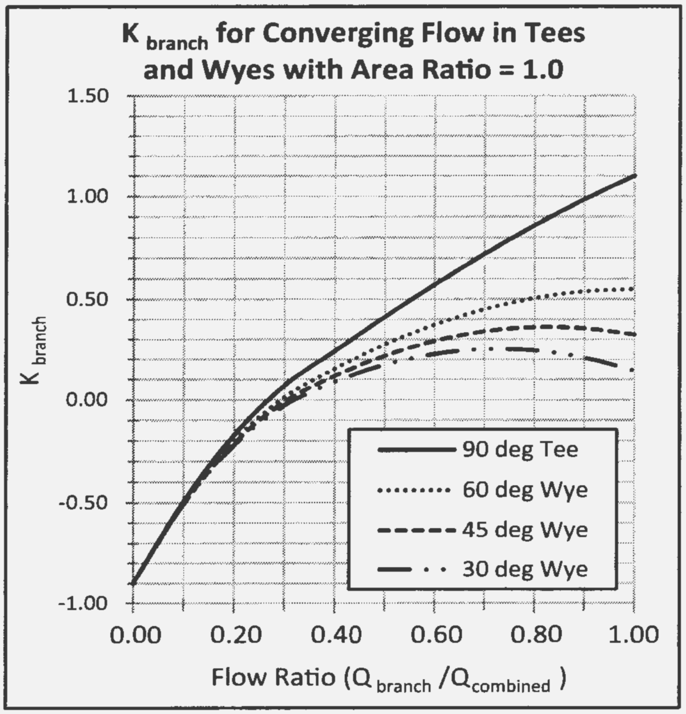


Figure 2-14:
K_{branch} for converging flow in tees and wyes

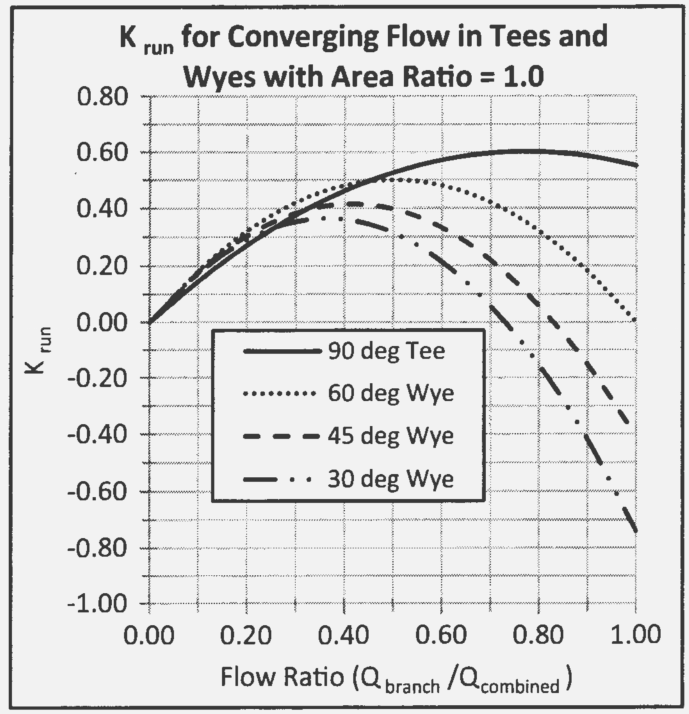


Figure 2-15:
K_{run} for converging flow in tees and wyes

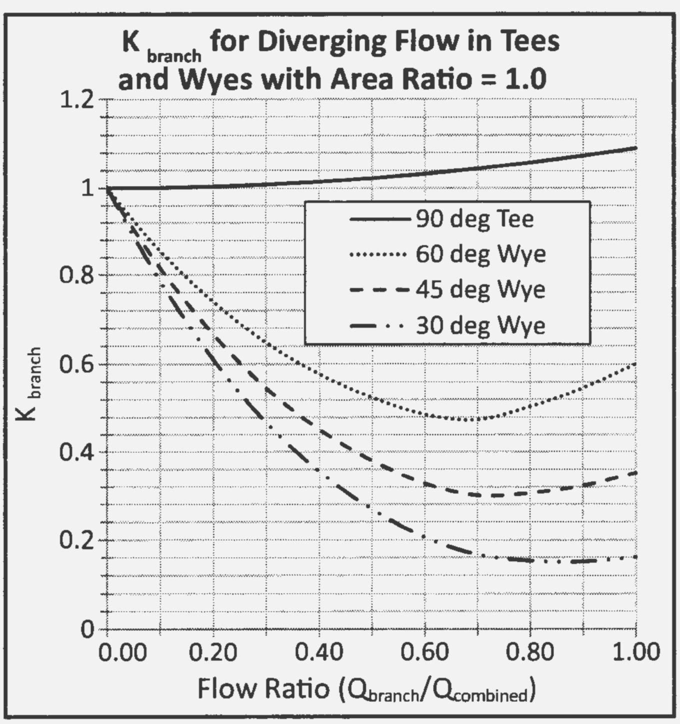


Figure 2-16:
K_{branch} for diverging flow in tees and wyes

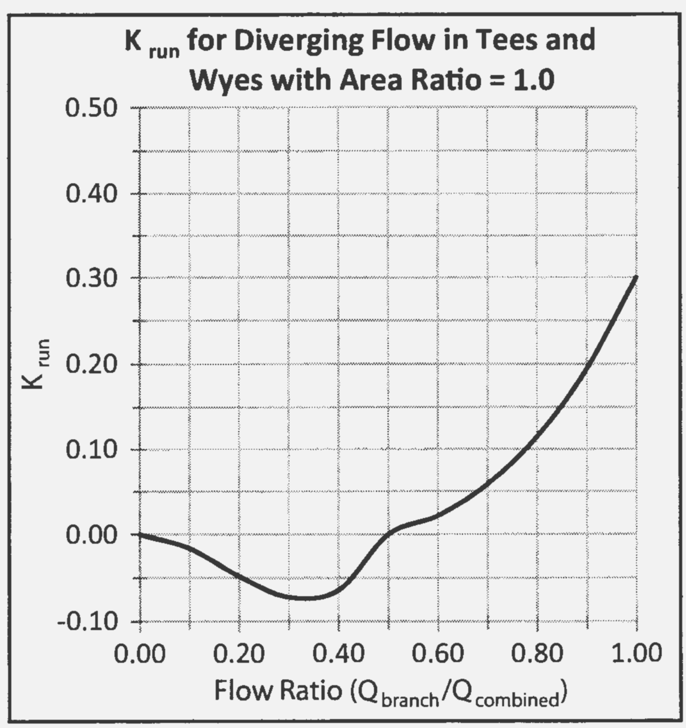


Figure 2-17:
K_{run} for diverging flow in tees and wyes of all angles
(Note: the inflection at about 0.5 flow ratio is due to the use of different equations in the calculation of the constant for Equation 2-38).

Discharge of Fluids Through Valves, Fittings and Pipe

Liquid flow: To determine the flow of liquid through pipe, the Darcy formula is used. Equation 1-16 (page 1-6) has been converted to more convenient terms in Chapter 6 and has been rewritten as Equation 6-22. The form of Equation 6-22, which is most applicable to liquid flow is written in terms of the flow rate in gallons per minute.

$$h_L = \frac{0.002593\ KQ^2}{d^4}$$ **Equation 2-39**

Solving for Q, the equation can be rewritten,

$$Q = \sqrt{\frac{h_L d^4}{0.002593\ K}} = 19.64 d^2 \sqrt{\frac{h_L}{K}}$$

Equation 2-39 can be employed for valves, fittings, and pipe where K would be the sum of all the resistances in the piping system, including entrance and exit losses when they exist. An Example of a problem of this type is shown in Example 7-19.

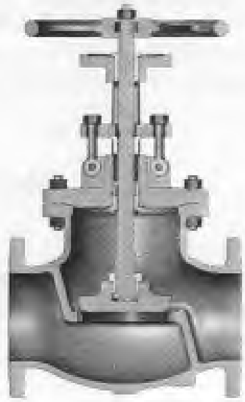
Compressible flow: When a compressible fluid flows from a piping system into an area of larger cross section that of the pipe, as in the case of discharge to atmosphere, a modified form of the Darcy formula, Equation 1-43 developed on page 1-11, is used.

$$w = 0.525\ Y\ d^2 \sqrt{\frac{\Delta P}{K\bar{V}_1}}$$ **Equation 2-40**

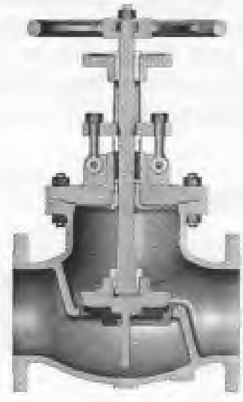
The determination of values of K, Y, and ΔP in this equation is described on page 1-11 and is illustrated in the Examples 7-20 through 7-22.

Types of Valves

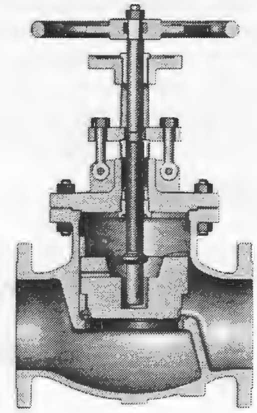
For K Factors see pages A-28 through A-30.



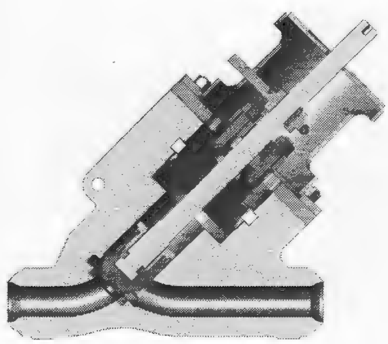
Conventional Globe Valve



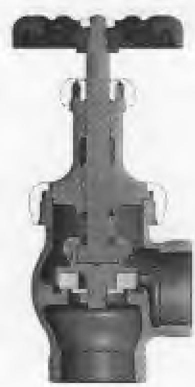
Conventional Globe Valve with Disc Guide



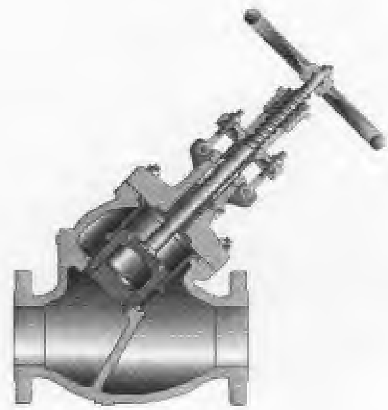
Globe Stop-Check Valve



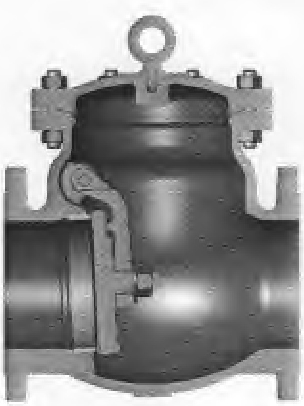
Y-Pattern Globe Valve with Stem 45 degrees from Run



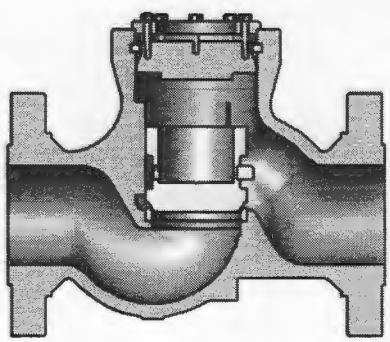
Conventional Angle Valve



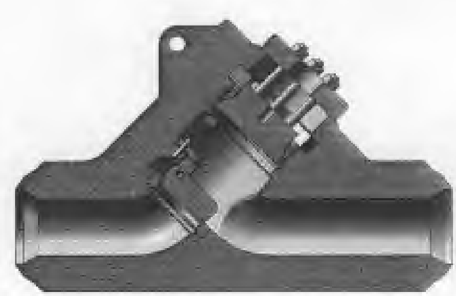
Angle Stop-Check Valve



Conventional Swing Check Valve



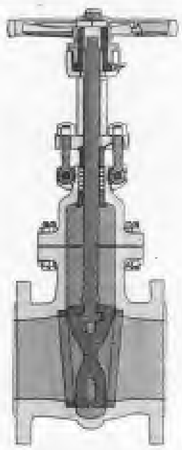
Globe Type Lift Check Valve



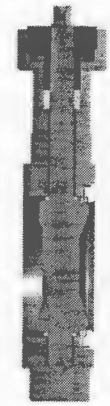
Tilting Disc Check Valve

Types of Valves

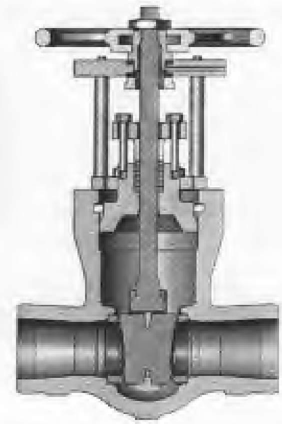
For K Factors see pages A-28 through A-30.



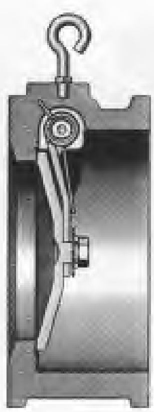
Wedge Gate Valve (Bolted Bonnet)



High Performance Butterfly Valve

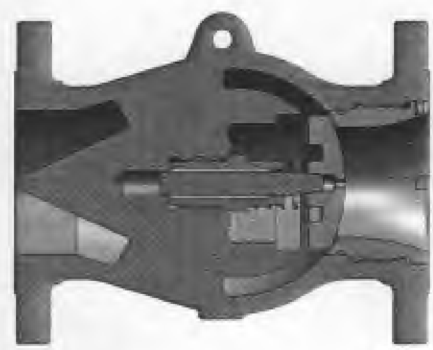


Flexible Wedge Gate Valve (Pressure Seal Bonnet)

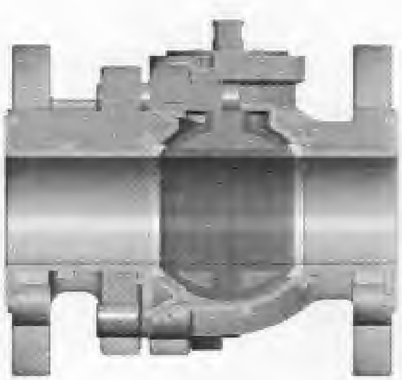


Swing Check Valve

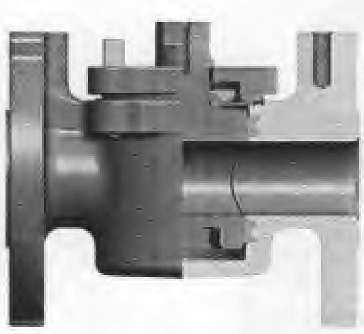
Dual Plate Check Valve



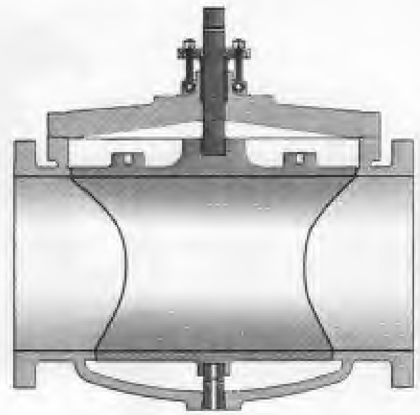
Nozzle (Venturi) Check Valve



Ball Valve



Plug Valve

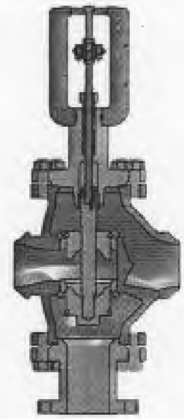


Lift Plug Valve

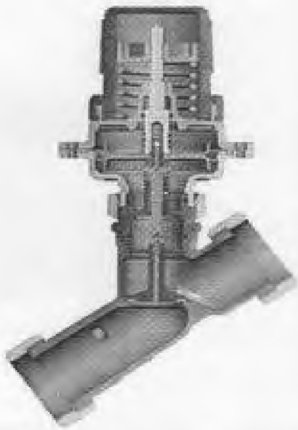
Types of Valves

For K Factors see pages A-28 through A-30.

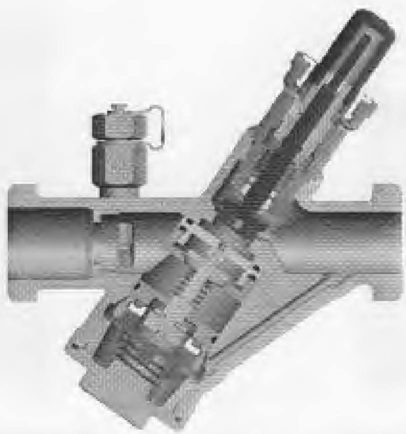
Safety Relief Valve



3 - Way Valve

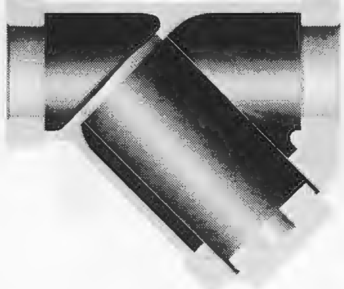


Differential Pressure Control Valve



Pressure Independent Control Valve

Industrial Diaphragm Valve



Strainer

Chapter 3

Regulating Flow with Control Valves

Control valves are uniquely engineered components used to vary the amount of fluid energy (head loss) dissipated across the valve in the form of heat, noise, and vibration. This is done in order to control the system flow rate, pressure, temperature, chemical composition, or some other system parameter such as a tank level. To vary the head loss of the control valve, the resistance of the valve is changed by changing the cross-sectional area of the valve's flow passage by adjusting the position of the valve disc (or plug) with relation to the seat. The movement of the disc can be done by an externally powered actuator; with a self-contained regulator that uses the energy of the fluid itself; or by manual actuation by an operator.

There are many types of control valves with unique designs for use in a wide range of applications. Linear motion valves move the valve stem linearly to change the shape of the flow area. These include globe, gate, diaphragm, and pinch valves. Ball, butterfly, and plug valves use rotary motion to rotate the valve stem to change the shape of the flow area.

Control Valves

Components: The valve body and bonnet contain the internal trim. The trim is considered the components that come into contact with the fluid passing through the valve and includes the valve stem, seat, disc (or plug), and cage (if installed), shown in Figure 3-1. The actuator is a pneumatic, hydraulic, electro-mechanical, or manual device attached to the valve stem that provides the force to move the stem in order to open, close, or throttle the position of the valve.

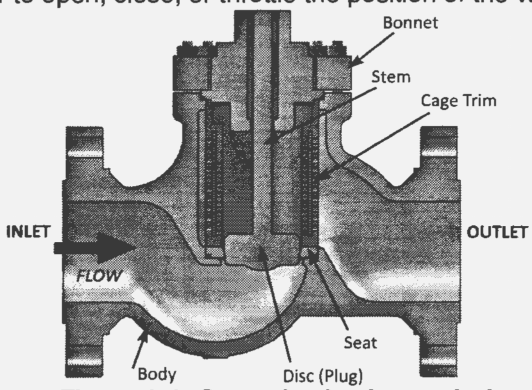


Figure 3-1: Control valve internal trim

The design of the trim determines the valve characteristics, or how the valve will perform with regard to its capacity as a function of the valve position. Valve trim can also be designed to reduce noise levels and prevent or minimize cavitation in the valve.

Inherent Characteristic Curve: The performance of a control valve is defined by its inherent characteristic curve, which is a plot of the valve position vs. flow coefficient (C_V) or percent of maximum C_V. The inherent characteristic curve is determined by measuring the flow rate (in gpm) of 60°F water at various positions of valve travel with a fixed differential pressure across the valve (typically 1 psid) and calculating the valve C_V at each position using Equation 3-1.

$$C_v = \frac{Q}{F_P\sqrt{\frac{P'_1 - P'_2}{S}}} \qquad \textbf{Equation 3-1}$$

The most common characteristic curves are the quick opening, linear, and equal percentage curves (Figure 3-2). With different trim designs, manufactures also make valves with modified linear, modified equal percentage, parabolic, or square root characteristic curves.

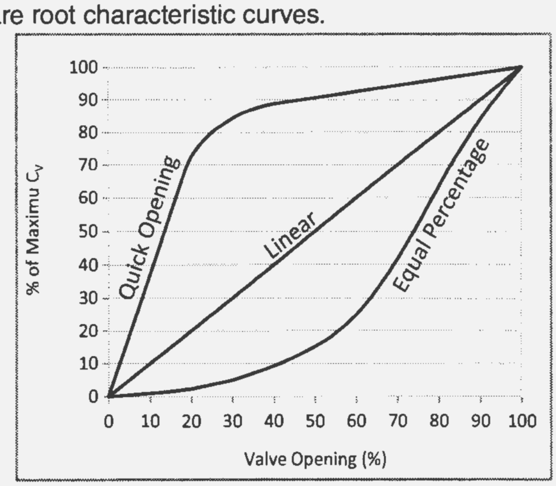


Figure 3-2: Common inherent characteristic curves

Installed characteristic curve: The installed characteristic curve is a plot of the valve position versus the percent of maximum flow rate for a valve installed in an actual piping system. The flow coefficient at each position for an installed valve will not change, but the installed curve will differ from the inherent curve because the differential pressure across the valve will change with a change in valve position. How the curve shifts is determined by the shape of the pump curve and the amount of static and dynamic head in the system. In general, the installed curve is shifted up and to the left from the inherent curve.[24] In the system shown in Figure 3-3 below, the flow control valve (FCV) has a maximum C_v of 225. The graph in Figure 3-4 shows how the installed characteristic curve is shifted if the valve had a linear or equal percentage characteristic.

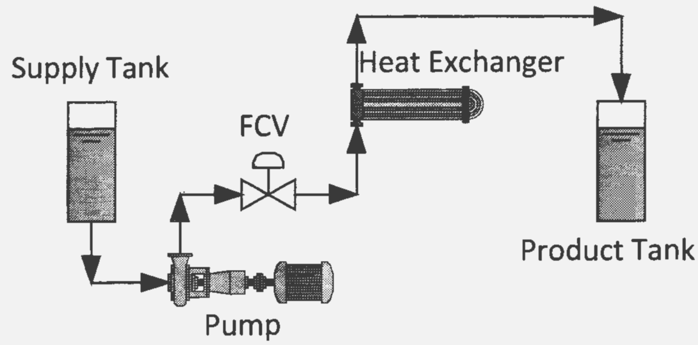


Figure 3-3: Typical piping system with tanks, pump, flow control valve, and heat exchanger

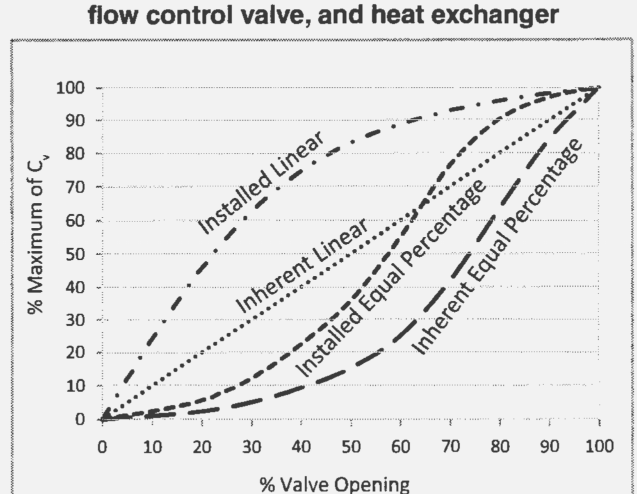


Figure 3-4: Shift of inherent characteristic curves

Pressure, velocity, and energy profiles: A graph of a generalized profile for pressure, velocity, and fluid energy is shown in Figure 3-5. Fluid velocity increases from the valve inlet to a maximum at the vena contracta due to the reduction in the area of the flow passage. Static pressure decreases as pressure head is converted into velocity head according to the Bernoulli theorem. Velocity decreases from the vena contracta to the valve outlet as the area of the flow passage increases, resulting in some pressure recovery in this region.

Control Valves

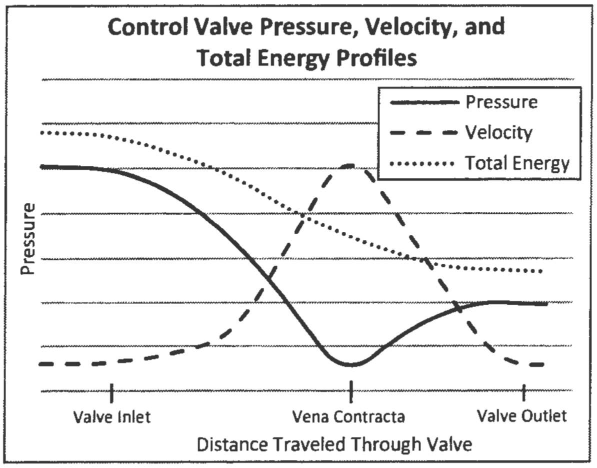


Figure 3-5: Pressure, velocity, and energy profiles in a control valve

The valve's liquid pressure recovery factor (F_L) is a measure of the pressure recovered relative to the valve pressure drop from the inlet to the vena contracta and is given by Equation 3-2.[25]

$$F_L = \sqrt{\frac{P'_1 - P'_2}{P'_1 - P'_{VC}}} \qquad \textbf{Equation 3-2}$$

Because the pressure at the vena contracta is not easily measured, F_L is determined by the manufacturer by testing and is typically included in the valve data table along with the C_V values. If F_L is not readily available from the manufacturer, typical values for various types of valves can be obtained from the ANSI/ISA-75.01.01 standard.[26]

Cavitation, choked flow, and flashing: For a set valve position with the flow of an incompressible liquid in which the static pressure at the vena contracta remains above the fluid's vapor pressure, a decrease in the downstream pressure will result in an increase in the flow rate through the valve as determined by the C_V equation. As the downstream pressure is reduced the flow rate increases and the static pressure at the vena contracta will decrease.

If the static pressure within the valve drops below the liquid vapor pressure, vapor bubbles will form and then collapse as they move into a region of higher pressure as the fluid velocity decreases. This process is called cavitation and can cause severe erosion and damage to the valve if the vapor bubble collapses on the surface of the valve seat, disc, or other internal surface. Cavitation also results in increased noise levels and vibration of the attached piping and supports. In addition, because the vapor bubbles occupy more volume than the same mass of liquid, the flow rate through the valve begins to be restricted and the flow rate starts to deviate from that predicted by the C_v equation, as shown in Figure 3-6.

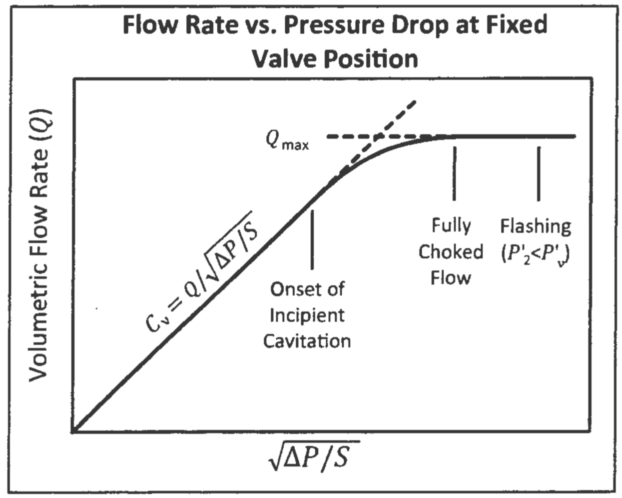


Figure 3-6: Graph of control valve flow rate and pressure drop

Incipient cavitation occurs when the static pressure at the vena contracta drops only slightly below the vapor pressure. Very small vapor bubbles are formed, the noise may be barely audible, and very little damage to the valve may result. The severity of cavitation increases as the downstream pressure is reduced and the fluid velocity increases, resulting in the liquid changing phase to vapor bubbles. When all of the liquid at the vena contracta changes phase to a vapor, a further drop in downstream pressure will not result in an increase in flow rate and the flow is considered fully choked.

If the downstream pressure does not rise above the vapor pressure, a condition called flashing is present which can severely damage valve internals and cause damage to the downstream piping.

Control Valve Sizing and Selection[26]

The process of properly sizing and selecting a control valve involves specifying the flow rate requirements, system pressures, and fluid properties. The valve flow coefficient is calculated using an appropriate equation for the application and then used to select a valve from a manufacturer that meets the size of the calculated flow coefficient. This is often an iterative process that results in adjustments to the calculated flow coefficient to take into account other variables as the proper valve size is determined.

The equations for incompressible flow are used for Newtonian fluids and should not be used for non-Newtonian fluids, mixtures, slurries, or liquid-solid conveyance systems. The equations for compressible fluids use correction factors to take into account compressibility effects and are for single phase gases and vapors only.

Sizing for Incompressible Flow: For non-choked, turbulent flow of an incompressible fluid with or without fittings attached to the valve inlet and/or outlet, calculate C_V using Equation 3-3:

$$C_v = \frac{Q}{F_P\sqrt{\frac{P'_1 - P'_2}{S}}} \qquad \textbf{Equation 3-3}$$

F_P is the piping geometry factor that takes into account the reduced flow capacity due to the head loss across fittings such as tees, elbows, or reducers attached two nominal pipe diameters (2D) upstream or six nominal pipe diameters (6D) downstream of the valve. F_P is calculated using the sum of the resistance coefficients of the fittings with Equation 3-4:

$$F_P = \frac{1}{\sqrt{1 + \frac{\Sigma K}{890}\left(\frac{C_v}{d^2_{nom}}\right)^2}} \qquad \textbf{Equation 3-4}$$

Where:

$$\Sigma K = K_1 + K_2 + K_{B1} - K_{B2}$$

K_1 = upstream fittings resistance coefficient
K_2 = downstream fittings resistance coefficient
K_{B1} = inlet Bernoulli coefficient = $1-(d_{nom}/d_1)^4$
K_{B2} = outlet Bernoulli coefficient = $1-(d_{nom}/d_2)^4$
d_{nom} = nominal valve size (in)
d = internal pipe diameter (in) (1=upstream, 2=downstream)
C_V = flow coefficient of assumed valve size at 100% open

If a smaller control valve than the line size can be selected, inlet and outlet reducers will need to be installed. For short-length commercially available concentric reducers, the resistance coefficients can be approximated as:*

$$K^{inlet}_{reducer} = 0.5\left[1 - \left(\frac{d_{nom}}{d_1}\right)^2\right]^2 \qquad \textbf{Equation 3-5}$$

$$K^{outlet}_{reducer} = 1.0\left[1 - \left(\frac{d_{nom}}{d_2}\right)^2\right]^2 \qquad \textbf{Equation 3-6}$$

If the inlet pipe is the same size as the outlet pipe, the Bernoulli coefficients K_{B1} and K_{B2} cancel out, therefore:

$$\Sigma K = 1.5\left[1 - \left(\frac{d_{nom}}{d}\right)^2\right]^2 \qquad \textbf{Equation 3-7}$$

The first iteration of the C_V calculation is done assuming the valve size is the same as the line size so no reducers are attached to the control valve.

If the valve will be installed close to an elbow or tee, F_P must be calculated with these fittings, otherwise F_P = 1.0 for the first iteration of the C_V calculation. If the initial calculated C_V allows the selection of a control valve smaller than the line size, F_P and C_V are recalculated to ensure the selected valve will still meet the service requirements.

Once an initial C_V is calculated and a preliminary control valve selected, the valve needs to be checked for the possibility of choked flow conditions by calculating the maximum flow rate at which choking occurs using Equation 3-8 or by calculating the maximum differential pressure at choked conditions using Equation 3-11 or 3-12.

$$Q_{max} = F_L C_V\sqrt{\frac{P'_1 - F_F P'_v}{S}} \qquad \textbf{Equation 3-8}$$

Where F_F is the liquid critical pressure ratio factor calculated using Equation 3-9.

$$F_F = 0.96 - 0.28\sqrt{\frac{P'_v}{P'_c}} \qquad \textbf{Equation 3-9}$$

If there are fittings attached, F_L in Equation 3-8 is replaced with F_{LP}/F_P. The combined liquid pressure recovery and piping geometry factor (F_{LP}) combines F_L and F_P into one factor, and should be determined by testing by the valve manufacturer. Otherwise, F_{LP} can be calculated with Equation 3-10, based on the sum of the resistances of the fittings upstream of the valve inlet, including the upstream Bernoulli coefficient, K_{B1}.

$$F_{LP} = \frac{F_L}{\sqrt{1 + F_L^2\frac{\Sigma K_i}{890}\left(\frac{C_v}{d^2_{nom}}\right)^2}} \qquad \textbf{Equation 3-10}$$

Where: $\Sigma K_i = K_1 + K_{B1}$

If Q_{max} is less than the flow rate used to calculate C_V, the flow will be choked at the valve position corresponding to the flow coefficient.

Another way to determine if the flow will be choked is to calculate the maximum differential pressure at which choked flow will occur, using Equation 3-11 for valves without fittings or Equation 3-12 for valves with fittings.

*For use only with control valves per ANSI/ISA 75.01.01, for reducers in pipelines see page 2-11.

Control Valve Sizing and Selection

$$\Delta P_{max} = F_L^2 (P'_1 - F_F P'_v) \qquad \text{Equation 3-11}$$

$$\Delta P_{max} = \left(\frac{F_{LP}}{F_P}\right)^2 (P'_1 - F_F P'_v) \qquad \text{Equation 3-12}$$

F_F is calculated using Equation 3-9.

If ΔP_{max} is less than the actual differential pressure across the valve, choked flow conditions will occur.

Calculations illustrating control valve sizing and choked flow conditions can be seen in Examples 7-27 and 7-28.

For laminar or transitional flow through a control valve, the sizing equation is modified to include the use of the valve Reynolds number factor (F_R), which is calculated iteratively. The ANSI/ISA-75.01.01 standard should be consulted for this calculation.

Sizing for Compressible Flow: For compressible fluid flow, the sizing equations are adjusted to take into account the compressibility of the fluid by calculating an expansion factor, Y, using Equation 3-13. The expansion factor takes into account the change in fluid density from the valve inlet to the vena contracta, as well as the change in the area of the vena contracta as the valve differential pressure is varied.

$$Y = 1 - \frac{x}{3F_K x_T} \qquad \text{Equation 3-13}$$

Where:
x = pressure drop ratio = $\Delta P / P'_1$
F_K = ratio of specific heats factor = k/1.4
(k = ratio of specific heats = c_p/c_v)
x_T = critical pressure drop ratio factor for a valve without fittings

The critical pressure drop ratio corresponds to the pressure ratio at which choked flow occurs. With a compressible fluid, choked flow occurs when the velocity at the vena contracta reaches sonic velocity and a further drop in downstream pressure will not result in an increase in flow.

The pressure drop ratio factor, x_T, is determined by air testing and should be included in the valve data table provided by the valve manufacturer. Typical values of x_T for various types of valves can be obtained from the ANSI/ISA-75.01.01 standard. The expansion factor varies from 1.0 at low pressure drop ratios to a minimum value of 0.667 at choked flow conditions.

If the valve is installed with fittings, x_T in Equation 3-13 is replaced with x_{TP} calculated with Equation 3-14.

$$x_{TP} = \frac{x_T / F_p^2}{1 + \left(\frac{x_T K_i}{1000}\right)\left(\frac{C_v}{d^2_{nom}}\right)^2} \qquad \text{Equation 3-14}$$

Where:
d_{nom} = assumed nominal valve size
C_V = valve flow coefficient at 100% open
F_P = piping geometry factor (Equation 3-4)
K_i = $K_1 + K_{B1}$

There are several forms of the valve sizing equation that can be used for compressible fluids based on the known fluid properties and flow rate units. Equation 3-15 is used if mass flow rate is specified and Equation 3-16 is used if volumetric flow rate is specified. If the valve is installed without fittings, F_P = 1.0.

$$C_v = \frac{W}{63.3\, F_P Y \sqrt{x P'_1 \rho_1}} \qquad \text{Equation 3-15}$$

$$C_v = \frac{q'_h}{1360\, F_P P'_1 Y \sqrt{\frac{x}{S_g T_1 Z_f}}} \qquad \text{Equation 3-16}$$

The compressibility factor (Z_f) used in Equation 3-16 takes into account the behavior of a real gas compared to an ideal gas.

A control valve is selected that has a flow coefficient close to the calculated C_V at the desired operating position. If a valve is selected that requires the installation of additional fittings, the calculations are repeated to ensure proper sizing and to determine if choked flow conditions can occur.

Conversion of C_V to K_V: The calculated flow coefficient C_V can be converted to the flow coefficient K_V, which is used by European valve manufacturers, using Equation 3-17.

$$K_V = 0.865\, C_V \qquad \text{Equation 3-17}$$

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Chapter 4

Measuring Flow with Differential Pressure Meters

The primary function of flow meters is to monitor, measure, or record the rate of fluid flow. Rate of flow can be determined by measuring the fluid's velocity, change in pressure, or directly measuring the volumetric or mass flow rate. There are a wide range of different types of flow meters. Some of these, arranged according to measurement technique, are listed in Table 4-1.

Differential Pressure	Velocity	Positive Displacement (Volumetric)	Mass
Orifice Plate	Turbine	Reciprocating Piston	Coriolis
Flow Nozzle	Vortex Shedding	Oval Gear	Thermal
Venturi Meter	Swirl	Nutating Disk	
Flow Tube	Electromagnetic (Mag meter)	Rotary Vane	
Pitot Tube	Ultrasonic (Doppler)		
Elbow Tap	Ultrasonic (Transit Time)		
Target	Coanda Effect		
Variable Area (Rotameter)	Momentum Exchange		

Table 4-1: Types of flow meters

The calculation of flow rate by measuring the differential pressure across a restriction-type flow meter is the most commonly used measurement technique in industrial applications. These devices follow Bernoulli's principles and the associated flow rates and pressure drops have been well documented over the years. Many of the other flow meters which directly measure velocity, volumetric flow rate or mass flow rate, rely on the use of calibrated electronics to transmit these measurements, and as such are not globally standardized. Standards have been developed by organizations including ASME MFC-3M[27] which detail the flow rate, pressure drop, and discharge coefficient equations for concentric orifice plates, flow nozzles, and venturi meters. For this reason, we will concentrate on these types of differential pressure flow meters in this chapter.

Differential Pressure Flow Meters

Orifice Plate: An orifice meter consists of a thin flat plate with a circular hole drilled in it (Figure 4-1). For most applications, the orifice hole is concentric and aligned with the center line of the pipe.

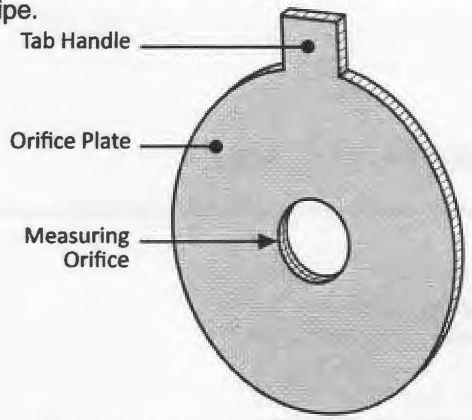


Figure 4-1: Concentric orifice plate

The general shape of the plate is such that the upstream and downstream faces of the plate are flat and parallel (Figure 4-2).

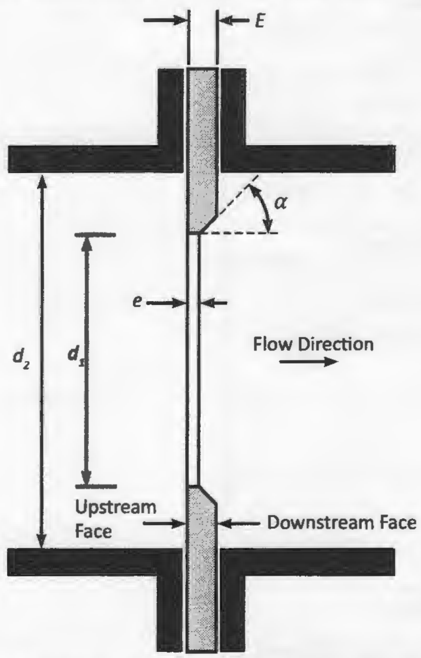


Figure 4-2: Orifice plate geometry

The diameter d_1 should be greater than or equal to 0.5 inch, and the Beta ratio ($\beta = d_1/d_2$) should be between 0.10 and 0.75. The upstream edge of the bore should be sharp and at an angle of 90° from the upstream surface. If the thickness at the bore (*e*) is less than the plate thickness (E), then the plate should be beveled on the downstream side at an angle α of 45°.

The orifice plate is typically installed in a pipeline between two flanges and pressure taps are placed both upstream and downstream to measure the differential pressure. There are several different tap arrangements which can be used (Figure 4-3).

Flange taps: Taps are located 1 inch upstream from the upstream face and 1 inch downstream from the downstream face of the orifice plate.

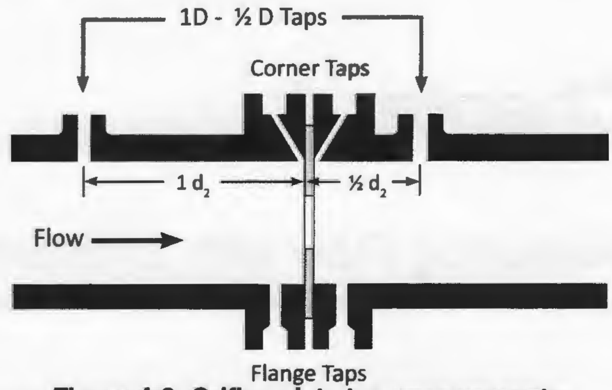


Figure 4-3: Orifice plate tap arrangements

1 D – ½ D taps: Taps are located one pipe diameter upstream from the upstream face of the orifice plate, and one half pipe diameter downstream from the downstream face of the orifice plate.

Corner taps: These are often used on smaller orifice plates with space restrictions. These taps are flush with the walls of the orifice plate.

Limits of Use: The formulas for standard orifice plates in this chapter can be applied under the following geometry and flow conditions:

1. For orifice plates with corner taps or 1D – ½ D taps:
   a. $d_1 \geq 0.5$ inch
   b. 2 inch $\leq d_2 \leq$ 40 inch
   c. $0.10 \leq \beta \leq 0.75$
   d. $R_e \geq 5000$ for $0.10 \leq \beta \leq 0.56$
   e. $R_e \geq 16{,}000\beta^2$ for $\beta > 0.56$
2. For orifice plates with flange taps:
   a. $d_1 \geq 0.5$ inch
   b. 2 inch $\leq d_2 \leq$ 40 inch
   c. $0.10 \leq \beta \leq 0.75$
   d. $R_e \geq 5000$ and $R_e \geq 4{,}318\beta^2 d_2$
3. For gases, $0.80 < (P'_2/P'_1) < 1.00$
4. $dP \leq 36.31$ psid

The performance of orifice plates is sensitive to flow conditions in the pipeline. Therefore it is recommended that the orifice be installed in a straight length of pipe which is free of obstructions, valves, and fittings. The length of straight pipe necessary is dependent upon the types of fittings, the beta ratio of the orifice, and the presence of any flow conditioning devices. A table of straight lengths between orifice plates and fittings can be found in the ASME MFC-3M standard.[27]

Flow Nozzle: A flow nozzle is a circular device similar in function to an orifice. It consists of an upstream face, a convergent section, a cylindrical throat, and a plain end. Flow nozzles come in three basic types:

1. Long radius nozzle
2. ISA 1932 nozzle
3. Venturi nozzle

Long radius nozzles: There are two varieties of long radius nozzle: high beta ratio and low beta ratio nozzles (Figure 4-4).

The convergent section of a long radius nozzle follows the shape of a quarter ellipse. From Figure 4-4, the convergent section of the high β nozzle follows the shape of a quarter

Differential Pressure Flow Meters

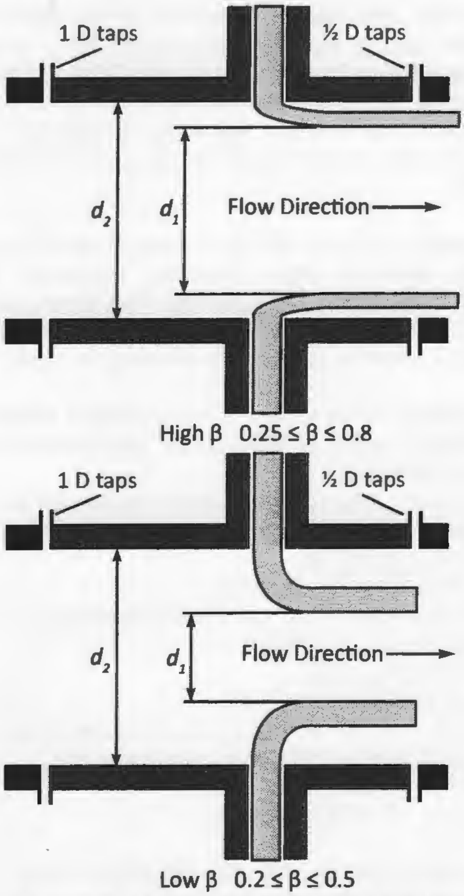


Figure 4-4: Long radius flow nozzles

ellipse with a large major axis diameter. The low β nozzle follows the shape of a quarter ellipse with a smaller major axis diameter.

The standard tap locations for long radius nozzles place the upstream tap at one pipe diameter from the plane of the inlet face of the nozzle, and the downstream tap at one half pipe diameter from the plane of the inlet face of the nozzle.

<u>ISA 1932 nozzles:</u> The ISA 1932 nozzle is similar in shape to the long radius nozzle with the exception of the convergent section, which has a rounded profile as opposed to elliptical (Figure 4-5).

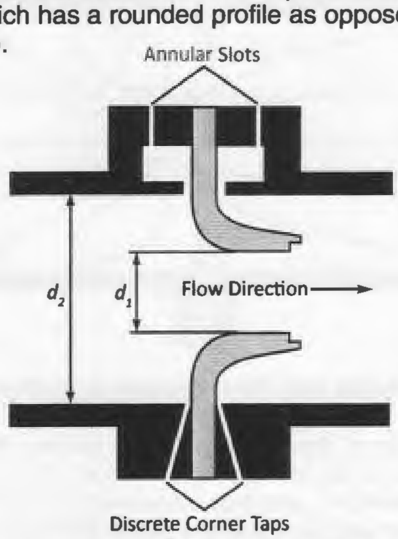


Figure 4-5: ISA 1932 flow nozzle

There is also an optional recess around the inside circumference of the nozzle outlet which is designed to prevent damage to the edge.

Corner taps are used with the ISA 1932 flow nozzle. These corner taps can either be single discrete taps, or annular slots as shown in Figure 4-5.

<u>Venturi nozzles:</u> The venturi nozzle consists of a convergent section with a rounded profile (exactly like the ISA 1932 nozzle), a cylindrical throat, and a divergent section (Figure 4-6).

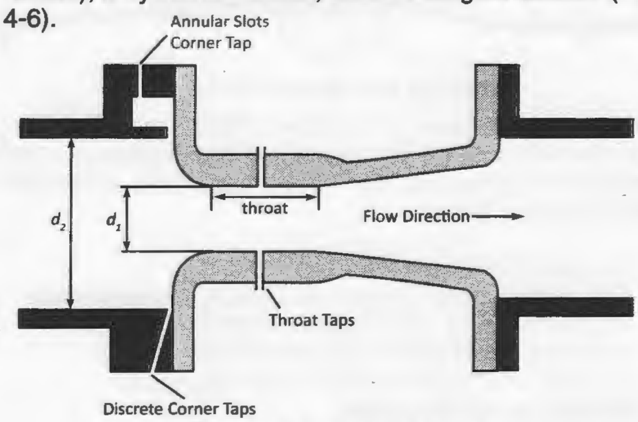


Figure 4-6: Venturi nozzle

The venturi nozzle design delivers a lower permanent pressure loss than either the long radius nozzle or ISA 1932 nozzle. The upstream pressure taps for a venturi nozzle are corner taps. They can be either single discrete taps, or annular slots. The downstream pressure taps are located in the throat section, and are referred to as the throat pressure taps.

Limits of Use: The formulas for standard flow nozzles in this chapter can be applied under the following geometry and flow conditions:

1. For long radius nozzles:
   a. 2 inch $\leq d_2 \leq$ 25 inch
   b. $0.20 \leq \beta \leq 0.80$
   c. $1 \times 10^4 \leq R_e \leq 1 \times 10^7$
   d. $\varepsilon/d_2 \leq 3.2 \times 10^{-4}$
2. For ISA 1932 nozzles:
   a. 2 inch $\leq d_2 \leq$ 20 inch
   b. $0.30 \leq \beta \leq 0.80$
   c. $7 \times 10^4 \leq R_e \leq 1 \times 10^7$ for $0.30 \leq \beta \leq 0.44$
   d. $2 \times 10^4 \leq R_e \leq 1 \times 10^7$ for $0.44 \leq \beta \leq 0.80$
3. For venturi nozzles:
   a. 2.5 inch $\leq d_2 \leq$ 20 inch
   b. $d_1 \geq$ 2 inch
   c. $0.316 \leq \beta \leq 0.775$
   d. $1.5 \times 10^5 \leq R_e \leq 2 \times 10^6$

Flow nozzles are dimensionally more stable than orifice plates, and as such can handle high temperature and high velocity service applications. Flow nozzles can also handle higher flow capacities; however, like orifice plates, they are sensitive to flow conditions. Therefore it is recommended that they be installed in a straight length of pipe which is free of obstructions, valves, and fittings. A table of straight lengths between nozzles and fittings can be found in the ASME MFC-3M standard.[27]

Differential Pressure Flow Meters

Venturi Meter: A venturi meter consists of a cylindrical entrance section, a tapered conical convergent section, a short section of straight pipe called the throat, and a tapered conical divergent outlet section (Figure 4-7). All sections are concentric with the center line of the pipe.

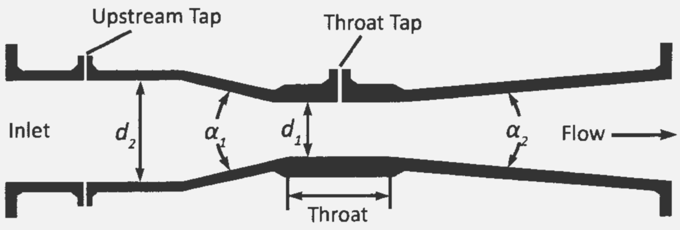


Figure 4-7: Venturi meter

There are three types of standard ASME venturi meters, each of which is defined by its method of manufacture of the convergent section and the intersection of the convergent section and the throat.

1. "As-Cast" Convergent Section.
   This venturi meter is made by casting in a sand mold or other method. The throat is machined and the junctions between the cylinders and cones are rounded. This type is used in pipe diameters between 4 and 48 inches.

2. Machined Convergent Section.
   This venturi meter has the convergent section machined as well as the cylindrical entrance and throat. The junctions between the cylinders and cones are rounded. This type is used in pipe diameters between 2 and 10 inches.

3. Rough-Welded Convergent Section.
   This venturi meter is normally fabricated by welding. This type is used in pipe diameters between 4 and 48 inches.

For all venturi, the entrance section is the same diameter as the pipe, and at least one pipe diameter in length. The convergent section has an angle α_1 of 21°± 1° and a length approximately equal to 2.7 times the quantity of d_2 - d_1. The throat section has a length equal to its diameter (d_1). The divergent section can have an angle α_2 anywhere between 7° and 15°.

The upstream pressure tap for a venturi meter is generally placed at one-half pipe diameter upstream from the beginning of the convergent section. The downstream tap is generally placed at one-half the length of the throat diameter downstream from the end of the convergent section.

Limits of Use: The formulas for standard venturi meters in this chapter can be applied under the following geometry and flow conditions:

1. For venturi tubes with "As-Cast" convergent section:
   a. 4 inch $\leq d_2 \leq$ 48 inch
   b. $0.30 \leq \beta \leq 0.75$
   c. $2.0 \times 10^5 \leq R_e \leq 6 \times 10^6$
2. For venturi tubes with machined convergent section:
   a. 2 inch $\leq d_2 \leq$ 10 inch
   b. $0.30 \leq \beta \leq 0.75$
   c. $2.0 \times 10^5 \leq R_e \leq 2 \times 10^6$
3. For venturi tubes with rough-welded convergent section:
   a. 4 inch $\leq d_2 \leq$ 48 inch
   b. $0.30 \leq \beta \leq 0.75$
   c. $2.0 \times 10^5 \leq R_e \leq 6 \times 10^6$

Venturi meters can handle very high flow rates, and have high pressure and energy recovery rates. Unrecovered pressure rarely exceeds 10% of the total measured differential pressure.

Liquid Flow Through Orifices, Nozzles, and Venturi

Orifices, nozzles and venturi are used principally to meter rate of flow. Orifices are also used to restrict flow or to reduce pressure, and are commonly referred to as flow restricting or balancing orifices. For liquid flow, several orifices are sometimes used to reduce pressure in steps so as to avoid cavitation.

Fluid accelerates as it passes through the restriction. The energy for this acceleration is provided by the fluid's static pressure. The fluid velocity increases and the static pressure decreases until the point of the vena contracta as shown for the orifice plate in Figure 4-8. Fluid velocity then slows and recovers some of the static pressure per the Bernoulli theorem.

Meter Differential Pressure (dP): The difference between the absolute pressures at the upstream and downstream taps (P'_1 - P'_2) is referred to as the differential pressure, dP, or ΔP.

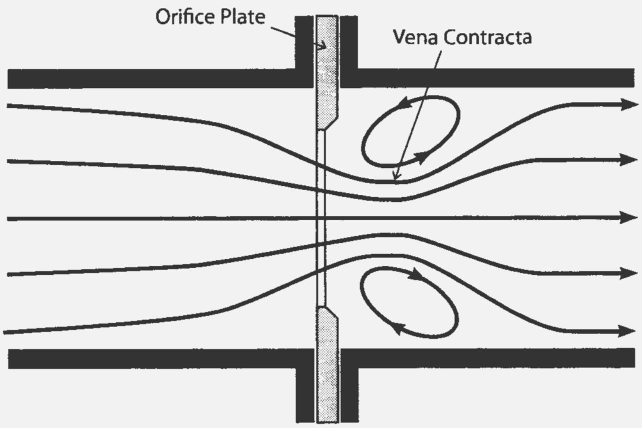


Figure 4-8: Flow through an orifice

Pressure Loss (NRPD): The permanent pressure loss or non-recoverable pressure drop (NRPD) is the difference in static pressure between the pressure measured on the upstream side of the primary device before, the influence of the approach impact pressure (approximately one pipe diameter upstream), and that measured on the downstream side of the primary device where the static pressure recovery can be considered completed (approximately six pipe

Liquid Flow Through Orifices, Nozzles, and Venturi

diameter downstream). For orifice plates, ISA 1932 nozzles and long radius nozzles, the NRPD can be approximated as:

$$NRPD = \Delta P\left[\frac{\sqrt{1-\beta^4(1-C_d^2)}-C_d\beta^2}{\sqrt{1-\beta^4(1-C_d^2)}+C_d\beta^2}\right] \qquad \textbf{Equation 4-1}$$

For venturi nozzles and tubes, when the divergent angle is not greater than 15 degrees, an approximate value of the NRPD can be accepted as being between 5% and 20% of the metered dP plus the differential pressure of the same length of straight pipe as the venturi nozzle or tube, depending on construction and flow conditions of the meter.

For orifice plates, ISA 1932 nozzles, and long radius nozzles, the pressure loss coefficient K can be calculated as follows:

$$K = \left[\frac{\sqrt{1-\beta^4(1-C_d^2)}}{C_d\beta^2} - 1\right]^2 \qquad \textbf{Equation 4-2}$$

The velocity, volumetric flow rate, and mass flow rate of the fluid can be calculated from the differential pressure since they are proportional to the square root of the pressure drop. The rate of flow of any fluid through an orifice, nozzle or venturi meter, neglecting the velocity of approach, may be expressed by:

$$q = C_d A\sqrt{2gh_L} \qquad \textbf{Equation 4-3}$$

Velocity of approach may have considerable effect on the quantity discharged through an orifice, nozzle or venturi. The factor correcting for velocity of approach, $\frac{1}{\sqrt{1-\beta^4}}$ may be incorporated in Equation 4-3 as follows:

$$q = \frac{C_d A}{\sqrt{1-\beta^4}}\sqrt{2gh_L} \qquad \textbf{Equation 4-4}$$

The flow coefficient C is related to the discharge coefficient C_d through the following equation:

$$C = \frac{C_d}{\sqrt{1-\beta^4}} \qquad \textbf{Equation 4-5}$$

Use of the flow coefficient C eliminates the necessity for calculating the velocity of approach, and Equation 4-3 may now be written:

$$q = CA\sqrt{2gh_L} = CA\sqrt{\frac{2g(144)\Delta P}{\rho}} \qquad \textbf{Equation 4-6}$$

Flow coefficient C values may be taken from the charts on page A-21. For primary devices discharging incompressible fluids to atmosphere, Equation 4-6 may be used if h_L or ΔP is taken as the upstream head or gauge pressure.

Discharge Coefficients C_d: The discharge coefficient is a dimensionless value which relates the actual flow rate to the theoretical flow rate through a primary device. It is dependent on the type of device and the tap arrangements. Reynolds number values are in reference to the upstream pipe.

Orifice plates: The discharge coefficient for orifice plates is given by the Reader-Harris/Gallagher (1998) equation:

$$C_d = 0.5961 + 0.0261\beta^2 - 0.216\beta^8 + 0.000521\left(\frac{10^6\beta}{R_e}\right)^{0.7} +$$

$$(0.0188 + 0.0063J)\beta^{3.5}\left(\frac{10^6}{R_e}\right)^{0.3} + (0.043 + 0.080e^{-10L_1} -$$

$$0.123e^{-7L_1})(1 - 0.11J)\frac{\beta^4}{1-\beta^4} - 0.031\,(M'_2 - 0.8M'^{1.1}_2)\beta^{1.3}$$

Equation 4-7a

When $d_2 < 2.8$ inch, the following term is added to equation 4-7a:

$$+\,0.011(0.75 - \beta)(2.8 - d_2) \qquad \textbf{Equation 4-7b}$$

where:

β = d_1/d_2 = diameter ratio

R_e = pipe Reynolds number

L_1 = ratio of the distance of the upstream tap from the upstream face of the plate and the pipe diameter

L'_2 = ratio of the distance of the downstream tap from the downstream face of the plate, and the pipe diameter (L'_2 denotes the reference of the downstream spacing from the downstream face, while L_2 would denote the reference of the downstream spacing from the upstream face)

$$M'_2 = \frac{2L'_2}{1-\beta}$$

$$J = \frac{19000\beta}{R_e}$$

The tap arrangements for the orifice plate must be in accordance with ASME MFC-3M specifications for corner taps, D and ½ D taps, and flange taps. The values of L_1 and L'_2 to be used in this equation are as follows:

a) for Corner taps: $L_1 = L'_2 = 0$
b) for D and ½ D taps: $L_1 = 1$, $L'_2 = 0.47$
c) for Flange taps: $L_1 = L'_2 = 1/d_2$

Tables of discharge coefficients for all three tap arrangements can be found in the ASME MFC-3M standard.[27]

Flow nozzles: The discharge coefficients for flow nozzles can be calculated for D and ½ D tap arrangements which are in accordance with ASME MFC-3M.[27]

ISA 1932 nozzles: **Equation 4-8**

$$C_d = 0.9900 - 0.2262\beta^{4.1} - (0.00175\beta^2 - 0.0033\beta^{4.15})\left(\frac{10^6}{R_e}\right)^{1.15}$$

Long radius nozzles:

$$C_d = 0.9965 - 0.00653\beta^{0.5}\left(\frac{10^6}{R_e}\right)^{0.5} \qquad \textbf{Equation 4-9}$$

Venturi nozzles:

$$C_d = 0.9858 - 0.196\beta^{4.5} \qquad \textbf{Equation 4-10}$$

Tables of discharge coefficients for these three types of nozzles can be found in the ASME MFC-3M standard.[27]

Venturi meters: The discharge coefficients for venturi meters are dependent upon the method of manufacture. When manufactured in accordance with ASME MFC-3M specifications, the discharge coefficients are a constant.

"As-Cast" convergent section: $C_d = 0.984$ **Equation 4-11**
Machined convergent section: $C_d = 0.995$ **4-12**
Rough-welded convergent section: $C_d = 0.985$ **4-13**

Compressible Flow Through Orifices, Nozzles, and Venturi

Examples illustrating flow meter calculations can be seen in Examples 7-23, 7-24, and 7-29 through 7-31.

Flow of gases and vapors: The flow of compressible fluids through orifices, nozzles and venturi can be expressed by the same equation used for liquids except the net expansibility factor must be included.

$$q = YCA\sqrt{\frac{2g(144)\Delta P}{\rho}}$$ **Equation 4-14**

Expansibility Factors (Y): The expansibility factor Y is a function of:

1. The specific heat ratio, k.
2. The ratio β of orifice or throat diameter to inlet diameter.
3. Ratio of downstream to upstream absolute pressures.

Orifice plates: The formula for calculating the expansibility factor for standard orifice plates is as follows:

Equation 4-15

$$Y = 1 - (0.351 + 0.256\beta^4 + 0.93\beta^8)\left[1 - \left(\frac{P'_2}{P'_1}\right)^{\frac{1}{k}}\right]$$

Flow nozzles and venturi meters: The formula for calculating the expansibility factor for flow nozzles and venturi meters is as follows:

Equation 4-16

$$Y = \left\{\left[\frac{k\left(\frac{P'_2}{P'_1}\right)^{\frac{2}{k}}}{k-1}\right]\left[\frac{1-\beta^4}{1-\beta^4\left(\frac{P'_2}{P'_1}\right)^{\frac{2}{k}}}\right]\left[\frac{1-\left(\frac{P'_2}{P'_1}\right)^{\frac{(k-1)}{k}}}{1-\left(\frac{P'_2}{P'_1}\right)}\right]\right\}^{0.5}$$

The expansibility factor has been experimentally determined on the basis of air with a specific heat ratio of approximately 1.4, and steam with a specific heat ratio of approximately 1.3. Tables of expansibility factors for orifices, flow nozzles and venturi meters can be found in the ASME MFC-3M[27] standard. The data is also plotted on page A-22 of this reference.

Values of k for some of the common vapors and gases are given on pages A-9 and A-10. The specific heat ratio k may vary slightly for different pressures and temperatures but for most practical problems the values given will provide reasonably accurate results.

Equation 4-14 may be used for orifices discharging compressible fluids to atmosphere by using:

1. Flow coefficient C given on page A-21 in the Reynolds number range where C is a constant for the given diameter ratio β.
2. Expansibility factor Y per page A-22.
3. Differential pressure ΔP, equal to the inlet gauge pressure.

This also applies to nozzles discharging compressible fluids to atmosphere only if the absolute inlet pressure is less than the absolute atmospheric pressure divided by the critical pressure ratio r_c; this is discussed in the next section. When the absolute inlet pressure is greater than this amount, flow through nozzles should be calculated as outlined in the next section.

Maximum flow of compressible fluids in a nozzle: A smoothly convergent nozzle has the property of being able to deliver a compressible fluid up to the velocity of sound in its minimum cross section or throat, providing the available pressure drop is sufficiently high. Sonic velocity is the maximum velocity that may be attained in the throat of a nozzle (supersonic velocity is attained in a gradually divergent section following the convergent nozzle, when sonic velocity exists in the throat).

The critical pressure ratio is the largest ratio of downstream pressure to upstream pressure capable of producing sonic velocity. Values of critical pressure ratio r_c which depend upon the ratio of nozzle diameter to upstream diameter as well as the specific heat ratio k are given on page A-22.

Flow through nozzles and venturi meters is limited by critical pressure ratio and minimum values of Y to be used in Equation 4-14 for this condition, are indicated on page A-22 by the termination of the curves at $\rho'_2 / \rho'_1 - r_c$.

Equation 4-14 may be used for discharge of compressible fluids through a nozzle to atmosphere, or to a downstream pressure lower than indicated by the critical pressure ratio r_c by using values of:

Y	minimum per page A-22
C	page A-21
ΔP	$P'_1(1 - r_c)$; r_c per page A-22
ρ	weight density at upstream condition

Flow through short tubes: Since complete experimental data for the discharge of fluids to atmosphere through short tubes (L/D is less than, or equal to, 2.5 pipe diameters) are not available, it is suggested that reasonably accurate approximations may be obtained by using Equations 4-6 and 4-14, with values of C somewhere between those for orifices and nozzles, depending upon entrance conditions.

If the entrance is well rounded, C values would tend to approach those for nozzles, whereas short tubes with square entrances would have characteristics similar to those for square edged orifices.

Chapter 5

Pumping Fluid Through Piping Systems

Pumps are mechanical devices that add hydraulic energy to a fluid to cause flow through piping systems, by increasing the fluid pressure at the pump discharge. There are a wide variety of pump designs to accomplish this task; but they fall into two general categories, kinetic and positive displacement. As centrifugal pumps, a type of kinetic, are the most common in industry, these will be the focus of this chapter.

Even among the centrifugal pumps there is a vast array of pump types; submersible, end suction, split case and column pumps for example. The varying designs handle different fluids, pressures, flows (capacities) and other system conditions. The selection of a pump should take into account all of these factors. The head and capacity information is typically presented by the manufacturer in the form of a pump performance curve. This curve represents performance characteristics over the operating range of the pump.

The pump curve, affinity rules, Net Positive Suction Head available and operating costs can be used to properly select a pump and evaluate the system performance.

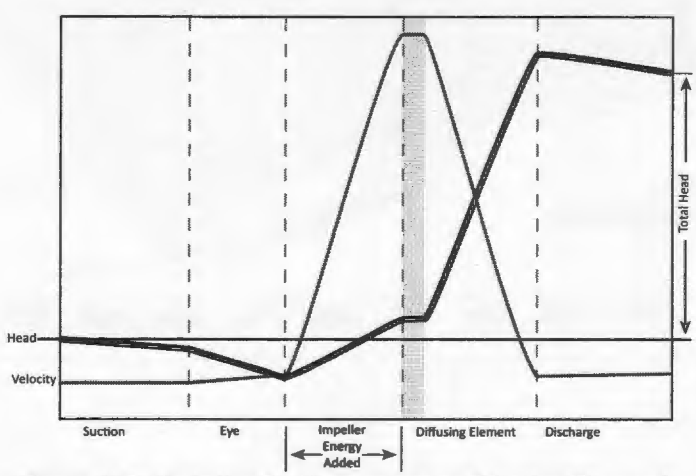


Figure 5-1: Variation in the energy and hydraulic grade lines as fluid pass through a pump[1]

Centrifugal Pump Operation

Centrifugal Pump Operation: When fluid approaches the pump suction, pressure drops as the fluid begins to experience centrifugal forces and changes direction from axial to radial flow. The pressure is further reduced to a minimum at the eye of the impeller. The low pressure area brings more fluid into the pump suction. As the fluid travels between the impeller vanes, centrifugal forces accelerate the fluid and energy is added in the form of velocity and pressure head. Fluid velocity reaches a maximum at the tips of the impeller. At this point, pump geometry and operating conditions play a substantial role in the pressure and velocity profiles. The kinetic energy of the decelerating fluid is then converted to potential energy in the form of pressure head. In some pump designs this is preceded by a period of constant head and velocity as shown in the gray area of figure 5-1. The high pressure fluid then exits the discharge of the pump to the piping system. This conversion of energy is described by the Bernoulli Equation and can be seen approximated in Figure 5-1.

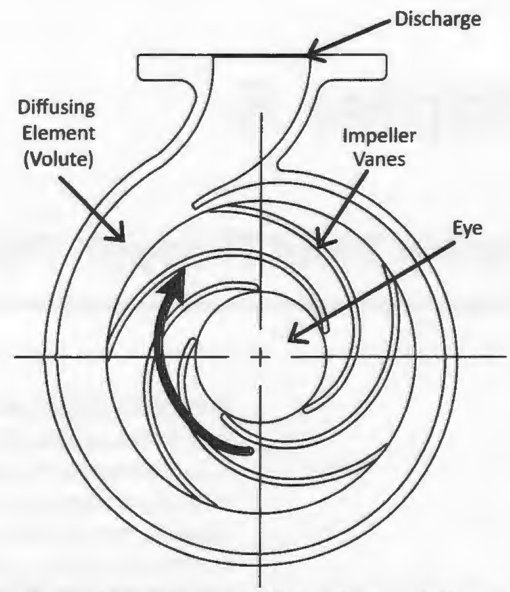


Figure 5-2: Typical cross section of a centrifugal pump

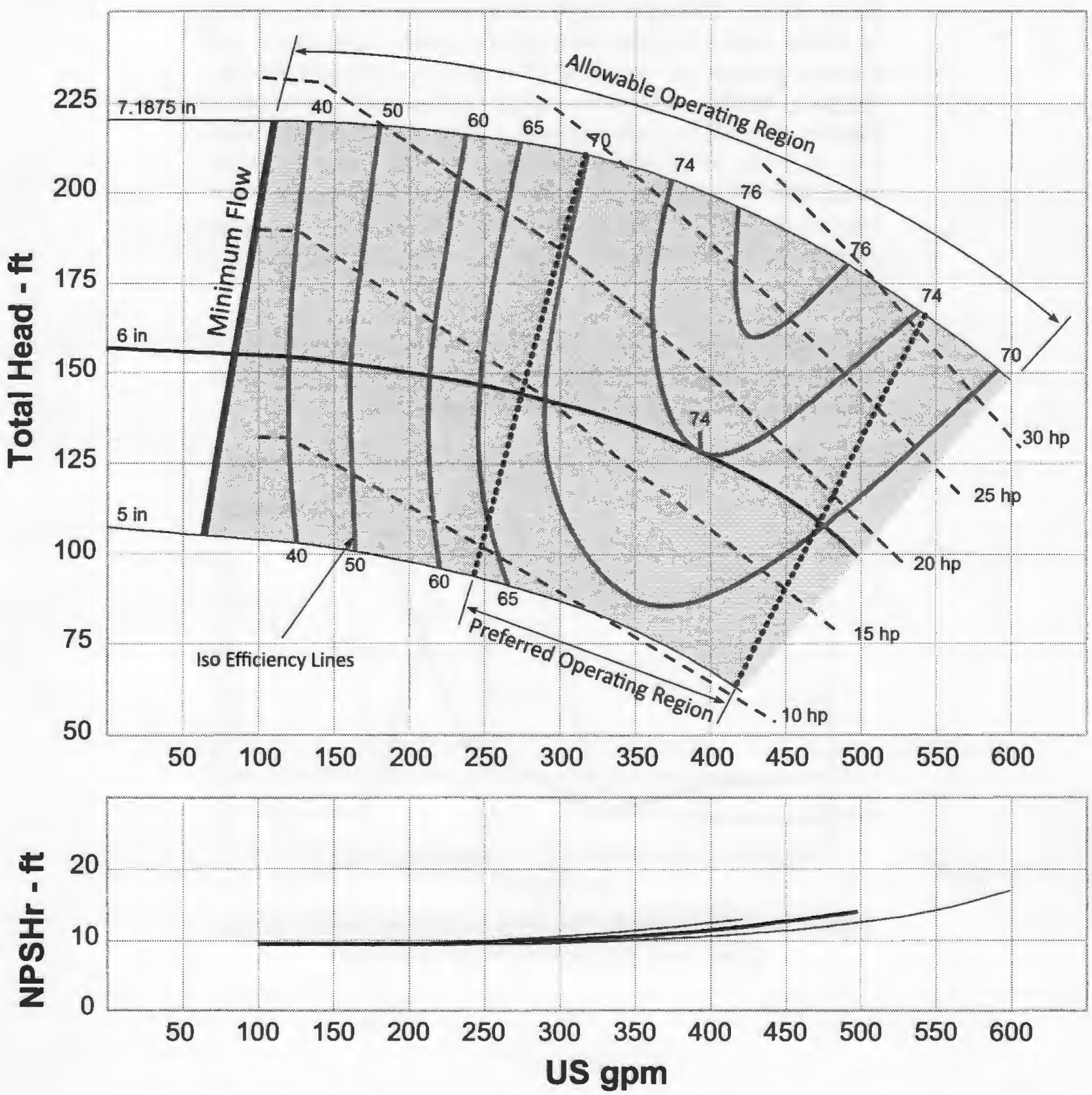


Figure 5-3: Pump curve showing efficiency over a range of impeller sizes

Centrifugal Pump Sizing and Selection

Pump Curve: The pump curve is developed by testing the pump according to industry standards[31] and smoothing the resulting head and flow rate data into a curve. The manufacturer may provide a performance curve for a single impeller size and speed, or multiple curves for a range of impeller sizes or speeds. Elements found on the pump curve include:

Total Head: The energy content of the liquid, imparted by the pump, expressed in feet of liquid.

Pump Efficiency: The ratio of the energy supplied to the liquid to the energy delivered from the pump shaft.

Shutoff Head: The head generated at the condition of zero flow where no liquid is flowing through the pump, but the pump is primed and running.

Minimum Flow: The lowest flow rate at which the manufacturer recommends the pump be operated.

Allowable Operating Region (AOR): The range of flow rates recommended by the pump manufacturer in which the service life of the pump is not seriously reduced by continuous operation.

Best Efficiency Point (BEP): The flow rate on the pump curve where the efficiency of the pump is at its maximum. Operating near this point will minimize pump wear.

Preferred Operating Region (POR): A region around the BEP on the pump curve, defined by the user, to ensure reliable and efficient operation.

Maximum Flow Rate: The end of the manufacturer's curve for the pump, commonly referred to as "run out."

Net Positive Suction Head required (NPSHr): The amount of suction head above the vapor pressure needed to avoid more than 3% loss in total head due to cavitation at a specific capacity.

NPSHa: The Net Positive Suction Head available (NPSHa) is the head provided by the piping system to the pump suction. It is influenced by the configuration of the system and the properties of the fluid. The NPSHa should be calculated to ensure that it exceeds the (NPSHr) provided by the manufacturer to prevent cavitation in the pump.

$$\text{NPSHa} = \frac{144}{\rho}(P'_t - P'_v) + (Z_t - Z_s) - h_L \qquad \textbf{Equation 5-1}$$

NPSH Optimization: The variables of equation 5-1 can be optimized within a piping system to increase NPSHa at the pump suction.

Pump Location: Lowering the pump suction in relation to the tank will increase NPSHa.

Pump Suction Piping: Minimizing suction pipeline head loss will increase NPSHa. This head loss will be a factor of pipe size, pipe roughness and any components installed in the pipeline. This head loss can be calculated using the methods, such as Darcy's formula, outlined in Chapter 1. In addition, as flow increases through the suction pipeline, head loss will increase, effectively reducing the NPSHa. For most pumps, NPSHr will increase with flow rate.

Fluid Properties: Fluid properties such as vapor pressure, density and viscosity vary with temperature. The net effect of a change in fluid temperature on NPSHa should be evaluated.

Supply Tank: An increase in supply tank pressure, elevation or liquid level will increase the NPSHa.

Atmospheric Pressure: As the pressures in Equation 5-1 are absolute, and those read at a tank typically reference gauge, a decrease in atmospheric pressure will reduce the NPSHa.

Viscosity Corrections: Most published pump curves reflect the performance of the pump with water as the operating fluid. A more viscous fluid will lead to an increase in required power and a reduction in flow rate, head and efficiency. Pump performance should be corrected for viscosity to obtain the most accurate representation of operation. There are published methods available to predict the effects of viscosity on pump performance.[29] Pump selection software is also available that will perform these viscosity corrections as part of the selection process.[30]

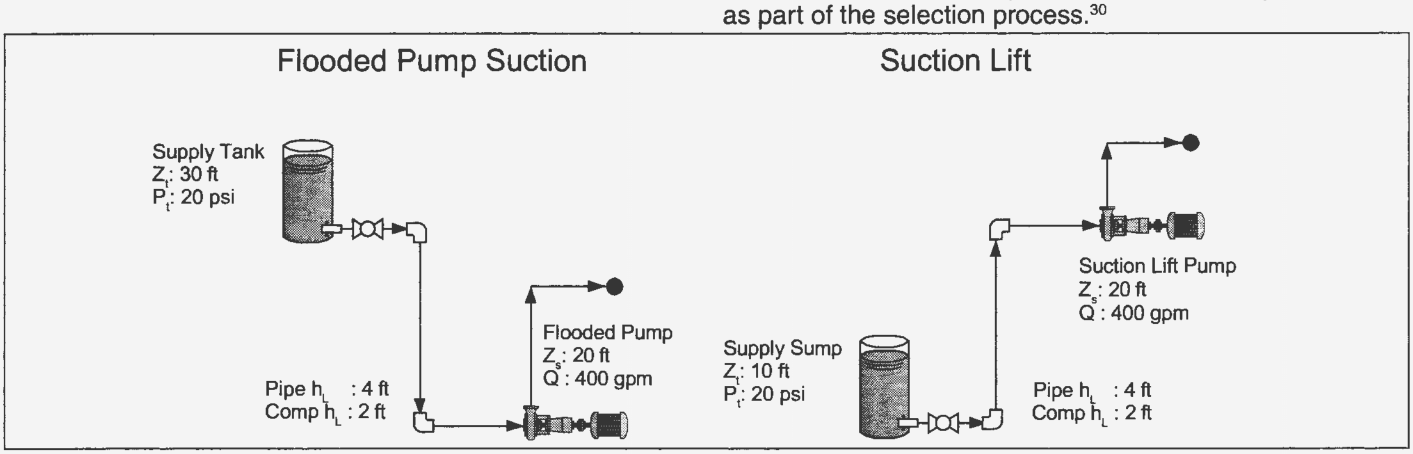


Figure 5-4: NPSHa parameters for flooded suction and suction lift

Centrifugal Pump Sizing and Selection

Pump Affinity Rules: The affinity rules predict the pump performance for a given change in impeller speed or diameter.

Changes in Impeller Speed: When a pump's rotational speed (N) is changed, the head (H), capacity (Q), and power (P) for a point on the pump curve vary according to the pump affinity rules.

Flow rate:

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2} \qquad \textbf{Equation 5-2}$$

Head:

$$\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2 \qquad \textbf{Equation 5-3}$$

Power:

$$\frac{P_1}{P_2} = \left(\frac{N_1}{N_2}\right)^3 \qquad \textbf{Equation 5-4}$$

Changes in Impeller Diameter: Trimming an impeller changes the vane angle, vane thickness and impeller clearance. These changes will impact pump performance but are not accounted for by the affinity rules. As a result, the affinity rules should be used only for small changes (typically <5%) in impeller diameters, as increased inaccuracies may occur with larger changes. Interpolation between two known impeller diameters on the pump curve typically provides more accurate results.

Capacity:

$$\frac{Q_1}{Q_2} = \frac{D_1}{D_2} \qquad \textbf{Equation 5-5}$$

Head:

$$\frac{H_1}{H_2} = \left(\frac{D_1}{D_2}\right)^2 \qquad \textbf{Equation 5-6}$$

Power:

$$\frac{P_1}{P_2} = \left(\frac{D_1}{D_2}\right)^3 \qquad \textbf{Equation 5-7}$$

Pump Power Calculations: Pump horsepower can be used to appropriately size a motor for the pump and calculate operating costs based on pump and motor efficiencies.

Brake (shaft) Horsepower:

$$bhp = \frac{QH\rho}{247{,}000\,\eta_p} \qquad \textbf{Equation 5-8}$$

Electrical Horsepower:

$$ehp = \frac{bhp}{\eta_m\,\eta_{vsd}} \qquad \textbf{Equation 5-9}$$

Operating Cost: **Equation 5-10**

$$OC = \frac{0.7457QH\rho}{247{,}000\eta_p\eta_m\eta_{vsd}}(\text{Operating time})(\$/\text{kWh})$$

Calculations illustrating NPSHa, pump affinity rules, and power/operating cost can be seen in Examples 7-32 through 7-34.

Pump Selection: The process of pump selection can be broken down into a series of distinct steps. There are many software packages available to facilitate the pump selection process. [31]

Determine Pump Capacity: This is the flow rate that is desired from the pump, usually in gallons per minute.

Determine Head Requirements: The pump must overcome the static and dynamic head losses of the system. These losses can be an estimate based on general system conditions or hand calculated using the Darcy equation. For more complex systems, hydraulic analysis software may be warranted.

Find NPSHa: This can be calculated by hand using the equations found in this chapter.

Select the Pump: Typically, a selection chart is consulted to create a short list of pumps for evaluation, the curves are then individually considered to find the best fit.

Correct for Fluid Density and Viscosity: Both of these will impact the shape of the pump performance curve and need to be adjusted for the fluid being pumped.

Find the Pump Horsepower: Curves for horsepower may be included on the published pump curve, if not it can be calculated using the equations in this chapter.

Positive Displacement Pumps

Positive Displacement (P.D.) Pumps: P.D. pumps add energy to a fluid by the direct application of force to one or more movable volumes of liquid. This energy is added in a periodic (not continuous) fashion. There are two main categories of P.D. pump. Reciprocating P.D. pumps use reciprocal motion (e.g., pistons or diaphragms) to directly displace a volume of fluid. Rotary pumps employ a variety of designs (e.g., peristaltic, screw and gear pumps) to displace fluid through the direct application of rotary motion.

P.D. Pump Application: The fact that P.D. pumps add energy by direct force on a volume makes them a suitable choice for certain applications. They impart little shear force to the fluid, making them suitable for high viscosity fluids, low shear requirements and the pumping of fragile solids. By directly moving a volume of fluid they can meet high pressure/low flow, and precise fluid delivery requirements as well as efficient pumping of two-phase fluids.

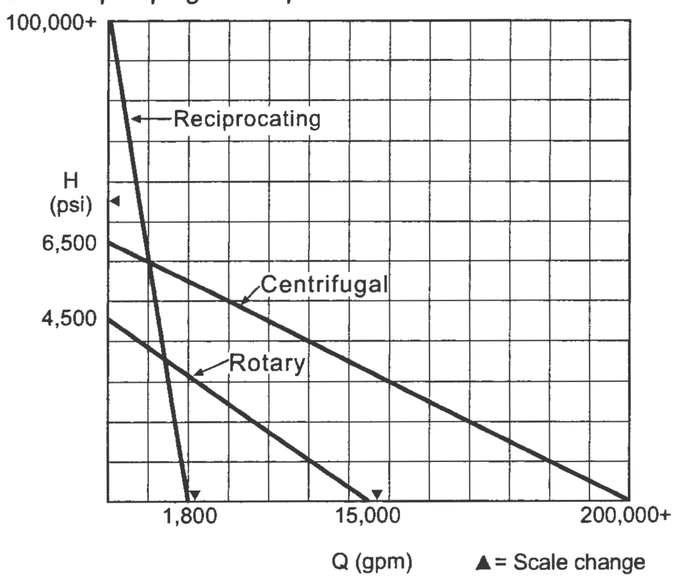


Figure 5-5: Head versus flow for centrifugal, rotary and reciprocating pumps[32]

Pump Range: Centrifugal pumps and the two main types of P.D. pumps are applicable over different ranges of head and flow. Figure 5-5 shows typical effective ranges for these pumps.

P.D. Pump Curve: Positive displacement pump curves are not limited to a flow vs. head relationship. Flow vs. speed and flow vs. discharge graphs are also commonly used. With the exception of slip, capacity in a P.D. pump varies directly with speed, independent of head, as shown in Figure 5-6. Positive displacement pumps typically exhibit slip, which is fluid leakage from the high pressure side to the low pressure side of the pump. At higher pressures and/or lower viscosities, this will result in an increasing loss of capacity through the pump.

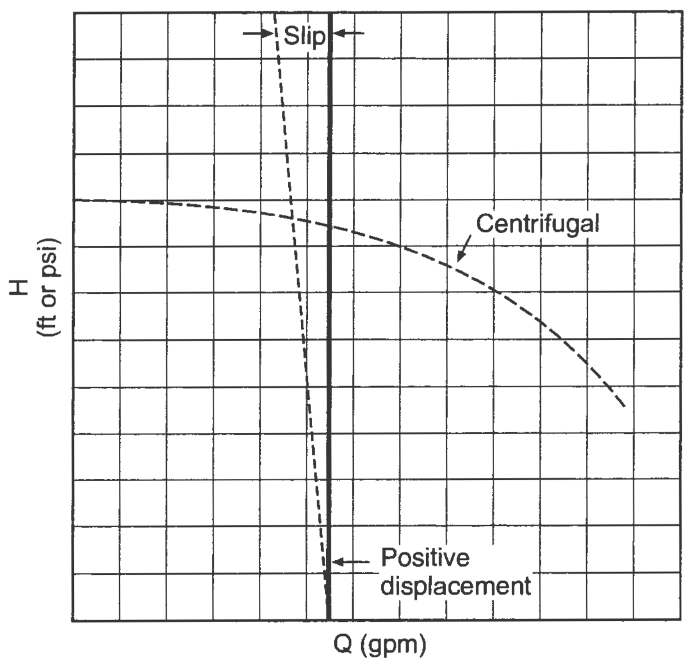
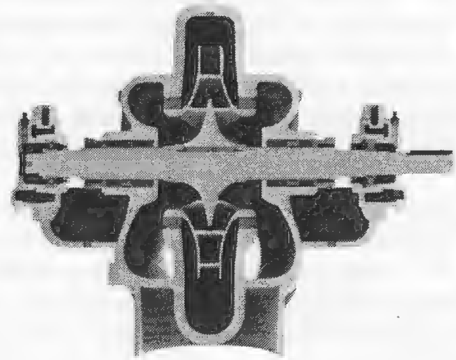


Figure 5-6: Typical head capacity relationships for centrifugal and P.D. pumps[32]

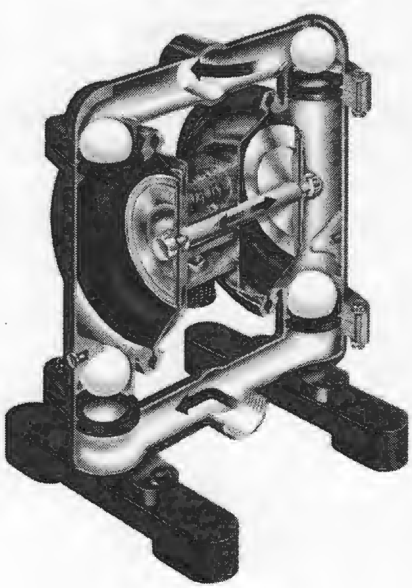
Types of Pumps

End Suction Pump

Submersible, Solids Handling Pump



Split Case Pump



Air Operated Double Diaphragm Pump

Column Sump Pump

Peristaltic Pump

Chapter 6

Formulas for Flow

Only basic formulas needed for the presentation of the theory of fluid flow through valves, fittings, and pipe were presented in the first five chapters of this paper. In the summary of formulas given in this chapter, the basic formulas are rewritten in terms of units which are most commonly used. This summary provides the user with an equation which will enable him to arrive at a solution to his problem with a minimum conversion of units.

Formulas from the chapters on control valves, pumps, and flow meters have also been included.

Nomographs used in previous versions of this technical paper have been removed due to the out-dated nature of their use and the availability of calculators, computers, computer software and web-based tools. To support users of this technical paper, many of the calculations previously done using nomographs have been moved to a suite of online tools located at www.flowoffluids.com.

www. This symbol = online calculators are available at www.flowoffluids.com.

Summary of Formulas

Basic Conversions: To eliminate needless duplication, formulas have been written only in terms of base units. The conversions given below can be substituted into any of the formulas in this paper whenever necessary.

$$q = vA = 0.001736\pi\, vd^2 = 0.005454vd^2$$

$$Q = 448.8q = 0.7792\pi\, vd^2 = 0.1247\frac{W}{\rho}$$

$$W = 3600w = 3600\rho q = 8.021\rho Q = \rho_a q'_h S_g$$

$$\rho = \frac{1}{V}$$

$$\upsilon = \frac{\mu}{\rho'} = \frac{\mu}{S} = \frac{62.364\mu}{\rho}$$

$$\Delta P = \frac{\rho h_L}{144}$$

Bernoulli's Theorem: **Equation 6-1**

$$H = Z + 144\frac{P}{\rho} + \frac{v^2}{2g}$$

$$Z_1 + 144\frac{P_1}{\rho_1} + \frac{v_1^2}{2g} = Z_2 + 144\frac{P_2}{\rho_2} + \frac{v_2^2}{2g} + h_L$$

Mean velocity of flow in pipe: **Equation 6-2**
(Continuity Equation)

$$v = \frac{w}{\rho A} = 0.16\frac{W}{\rho\pi d^2} = 0.05093\frac{W}{\rho d^2}$$

$$v = \frac{q}{A} = 1.283\frac{Q}{\pi d^2} = 0.4085\frac{Q}{d^2}$$

Head loss and pressure drop for incompressible flow in straight pipe: Pressure loss due to flow is the same in a sloping, vertical, or horizontal pipe. The pressure drop due to the difference in head is discussed elsewhere and must be considered in the pressure drop calculations.

Reynolds number of flow in pipe: **Equation 6-3**

$$R_e = \frac{Dv\rho}{\mu_e} = 124.0\frac{dv\rho}{\mu}$$

$$R_e = 71430\frac{q\rho}{\pi\mu d} = 22740\frac{q\rho}{d\mu} = 50.66\frac{Q\rho}{d\mu}$$

$$R_e = 19.84\frac{dW\rho}{\rho\pi d^2\mu} = 6.315\frac{W}{d\mu} = 22740\frac{w}{d\mu} = 0.4821\frac{q'_h S_g}{d\mu}$$

$$R_e = \frac{Dv}{\upsilon'} = \frac{dv}{12\upsilon'} = 7742\frac{dv}{\upsilon}$$

Laminar Friction Factor: **Equation 6-4**

$$f = \frac{64}{R_e} = 0.5161\frac{\mu}{dv\rho}$$

Turbulent Friction Factor:
For turbulent flow there is a variety of methods. The Moody Diagram offers a graphical representation of empirical data that provides an efficient method for use when performing hand calculations. The Colebrook White equation is an implicit, iterative solution that offers the best correlation to the Moody Diagram.

Colebrook Implicit Equation: **Equation 6-5**

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon}{3.7D} + \frac{2.51}{R_e\sqrt{f}}\right)$$

There are many explicit approximations of the Colebrook White equation that are in use. The Serghide equation offers a complex, but highly accurate direct approximation. The Swamee-Jaine is much simpler, but does not work for the full range of the Moody Diagram, nor does it correlate as well as the Serghide.

Serghide Explicit Equation: **Equation 6-6**

$$A = -2\log\left[\frac{\left(\frac{\varepsilon}{D}\right)}{3.7} + \frac{12}{R_e}\right]$$

$$B = -2\log\left[\frac{\left(\frac{\varepsilon}{D}\right)}{3.7} + \frac{2.51A}{R_e}\right]$$

$$C = -2\log\left[\frac{\left(\frac{\varepsilon}{D}\right)}{3.7} + \frac{2.51B}{R_e}\right]$$

$$f = \left[A - \frac{(B - A)^2}{C - 2B + A}\right]^{-2}$$

Swamee-Jain: The Swamee-Jain is valid for the following ranges of Reynolds number and relative roughness:

$5000 < Re < 3 \times 10^8, 10^{-6} < \frac{\varepsilon}{D} < 0.01$ **Equation 6-7**

$$f = \frac{0.25}{\left[\log\left(\frac{\varepsilon}{3.7D} + \frac{5.74}{R_e^{0.9}}\right)\right]^2}$$

Summary of Formulas

Head loss due to friction in straight pipes (Darcy):

Equation 6-8

$$h_L = f\frac{Lv^2}{D2g} = 6\frac{fLv^2}{dg} = 0.1865\frac{fLv^2}{d}$$

$$h_L = 9.882\frac{fLQ^2}{\pi^2 d^5 g} = 0.03112\frac{fLQ^2}{d^5}$$

$$h_L = 0.1536\frac{fLW^2}{\rho^2\pi^2 d^5 g} = 4.837\times 10^{-4}\left(\frac{fLW^2}{\rho^2 d^5}\right)$$

$$\Delta P = \frac{\rho}{144}\left(\frac{fLv^2}{D2g}\right) = 0.04167\frac{fL\rho v^2}{dg} = 0.001295\frac{fL\rho v^2}{d}$$

$$\Delta P = 0.06862\frac{fL\rho Q^2}{\pi^2 d^5 g} = 2.161\times 10^{-4}\left(\frac{fL\rho Q^2}{d^5}\right)$$

$$\Delta P = 0.0010667\frac{fLW^2}{\pi^2\rho d^5 g} = 3.3591\times 10^{-6}\left(\frac{fLW^2}{\rho d^5}\right)$$

Hazen-Williams formula for flow of water[4]: This formula is only appropriate for fully turbulent flow of fluids that are similar to 60°F water.

Equation 6-9

$$\Delta P_{per\_foot} = 4.52\frac{Q^{1.85}}{C^{1.85} d^{4.87}}$$

$$\Delta P = 4.52\frac{LQ^{1.85}}{C^{1.85} d^{4.87}}$$

$$h_L = 10.435\frac{LQ^{1.85}}{C^{1.85} d^{4.87}}$$

Pipe or Tube	Hazen-Williams C Value
Unlined cast or ductile iron	100
Galvanized steel	120
Plastic	150
Cement lined cast or ductile iron	140
Copper tube or stainless steel	150

Limitations of the Darcy Formula: The Darcy equation may be used without restriction for the flow of water, oil, and other liquids in pipe. However, when extreme velocities occurring in pipe cause the downstream pressure to fall to the vapor pressure of the liquid, cavitation occurs and calculated flow rates are inaccurate.

The Darcy equation may be used for gases and vapors with the following restrictions:

1. When ΔP is less than ~10% of P'_1, use ρ or $\bar{V}$ based on either the upstream or downstream conditions.

2. When ΔP is greater than ~10% of P'_1 but less than ~40% of P'_1, use the average ρ or $\bar{V}$ based on the upstream and downstream conditions, or use Equations 6-28.

3. When ΔP is greater than ~40% of P'_1, use the equations given on this page for compressible flow, or use Equations 6-28 (for theory, see page 1-11).

Isothermal Compressible Flow Equations:

The following isothermal compressible flow equations include the potential energy term (φ) and compressibility factor (Z_f) modifications. If elevation changes are neglected, then $\varphi = 0$ and if deviations from the ideal gas equation are neglected, then $Z_{f,avg} = 1$.

Equation 6-10

$$\varphi = 0.0375\frac{S_g\,\Delta Z\,(P'_{avg})^2}{T_{avg} Z_{f,avg}}$$

$$P'_{avg} = \frac{2}{3}\left[\frac{(P'_1)^3 - (P'_2)^3}{(P'_1)^2 - (P'_2)^2}\right]$$

$Z_{f,avg}$ is evaluated at P'_{avg} and T_{avg} and values for a specific gas may be obtained from various equations of state, charts, or other correlations.[11,14]

Simplified isothermal equation for long pipelines:

Equation 6-11

$$q'_h = 3.2308\left(\frac{T_b}{P'_b}\right)\left[\frac{(P'_1)^2 - (P'_2)^2 - \varphi}{f\,L_m T_{avg} Z_{f,avg} S_g}\right]^{0.5} d^{2.5}$$

Weymouth Equation (fully turbulent flow):

Equation 6-12

$$q'_h = 18.062\left(\frac{T_b}{P'_b}\right)\left[\frac{(P'_1)^2 - (P'_2)^2 - \varphi}{L_m T_{avg} Z_{f,avg} S_g}\right]^{0.5} d^{2.667}$$

Panhandle A Equation (partially turbulent flow):

Equation 6-13

$$q'_h = 18.161\,E\left(\frac{T_b}{P'_b}\right)^{1.0788}\left[\frac{(P'_1)^2 - (P'_2)^2 - \varphi}{L_m T_{avg} Z_{f,avg} S_g^{0.8539}}\right]^{0.5394} d^{2.6182}$$

Summary of Formulas

Panhandle B Equation (fully turbulent flow):

Equation 6-14

$$q'_h = 30.708\,E\left(\frac{T_b}{P'_b}\right)^{1.02}\left[\frac{(P'_1)^2-(P'_2)^2-\varphi}{L_m T_{avg} Z_{f,avg} S_g^{0.961}}\right]^{0.510} d^{2.53}$$

E is the efficiency factor for the Panhandle A and B equations.

E = 1.00 for brand new pipe without any bends, elbows, valves, and change of pipe diameter or elevation

E = 0.95 for very good operating conditions

E = 0.92 for average operating conditions

E = 0.85 for unusually unfavorable operating conditions

AGA Equation (partially turbulent flow):

Equation 6-15

$$\frac{1}{\sqrt{f}} = F_f\ 2\log\left(\frac{R_e\sqrt{f}}{2.825}\right)$$

$$q'_h = 3.2308\,E\left(\frac{T_b}{P'_b}\right)\left[\frac{(P'_1)^2-(P'_2)^2-\varphi}{L_m T_{avg} Z_{f,avg} S_g}\right]^{0.5}\left[F_f\ 2\log\left(\frac{R_e\sqrt{f}}{2.825}\right)\right] d^{2.5}$$

The drag factor (F_f) is used to account for additional resistances such as bends and fittings and ranges in value from 0.90 to 0.99. Specific values may be obtained from the AGA report.[11]

AGA Equation (fully turbulent flow): **Equation 6-16**

$$\frac{1}{\sqrt{f}} = 2\log\left(\frac{3.7D}{\varepsilon}\right)$$

$$q'_h = 3.2308\left(\frac{T_b}{P'_b}\right)\left[\frac{(P'_1)^2-(P'_2)^2-\varphi}{L_m T_{avg} Z_{f,avg} S_g}\right]^{0.5}\left[2\log\left(\frac{3.7D}{\varepsilon}\right)\right] d^{2.5}$$

Speed of sound and Mach number: The maximum possible velocity of a compressible fluid in a pipe is equivalent to the speed of sound in the fluid; for a perfect gas this is expressed as: **Equation 6-17**

$$v_s = c = \sqrt{kgRT}$$

$$v_s = c = \sqrt{kg144\,P'\,\overline{V}} = 68.067\sqrt{k\,P'\,\overline{V}}$$

The Mach number is a dimensionless ratio of the velocity of fluid to the speed of sound in fluid at local conditions and is expressed as: **Equation 6-18**

$$M = \frac{v}{c}$$

Head loss and pressure drop through valves and fittings: Head loss through valves and fittings is generally given in terms of resistance coefficient K which indicates static head loss through a valve in terms of "velocity head" or, equivalent length in pipe diameters L/D that will cause the same head loss as the valve.

From Darcy's Formula, head loss through a pipe is: **Equation 6-19**

$$h_L = f\frac{L}{D}\frac{v^2}{2g}$$

and head loss through a valve is: **Equation 6-20**

$$h_L = K\frac{v^2}{2g}$$

Therefore: **Equation 6-21**

$$K = f\frac{L}{D}$$

To eliminate needless duplication of formulas, the following are given in terms of K:

Equation 6-22

$$h_L = K\frac{v^2}{2g} = \frac{0.8236KQ^2}{\pi^2 d^4 g} = 0.002593\frac{KQ^2}{d^4}$$

$$h_L = K\frac{0.0128W^2}{\pi^2 d^4 \rho^2 g} = 4.031\times10^{-5}\frac{KW^2}{\rho^2 d^4}$$

$$\Delta P = \frac{\rho}{144}\left(K\frac{v^2}{2g}\right) = \frac{0.005719K\rho Q^2}{\pi^2 d^4 g} = 1.801\times10^{-5}\frac{K\rho Q^2}{d^4}$$

$$\Delta P = \frac{8.889\times10^{-5}KW^2}{\pi^2 d^4 \rho g} = 2.799\times10^{-7}\frac{KW^2}{\rho d^4}$$

For compressible flow with h_L or ΔP greater that approximately 10% of inlet absolute pressure, the denominator should be multiplied by Y^2. For values of Y, see page A-23.

Pressure drop and flow of liquids of low viscosity using flow coefficient: **Equation 6-23**

$$P = S\left(\frac{Q}{C_v}\right)^2 = \frac{\rho}{62.364}\left(\frac{Q}{C_v}\right)^2$$

$$Q = C_v\sqrt{\frac{\Delta P}{S}} = 7.897\,C_v\sqrt{\frac{\Delta P}{\rho}}$$

$$C_v = Q\sqrt{\frac{S}{\Delta P}} = 1.266Q\sqrt{\frac{\rho}{\Delta P}} = \frac{29.84d^2}{\sqrt{K}}$$

$$K = \frac{890.3d^4}{C_v^2}$$

Summary of Formulas

Resistance and Flow coefficients, K and C_v, in series and parallel: **Equation 6-24**

Series:

$$K_{Total} = K_1 + K_2 + K_3 + \dots K_n$$

$$\frac{1}{C_{vTotal}^2} = \frac{1}{C_{v1}^2} + \frac{1}{C_{v2}^2} + \dots \frac{1}{C_{vn}^2}$$

Parallel: **Equation 6-25**

$$\frac{1}{K_{Total}} = \frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3} \dots \frac{1}{K_n}$$

$$C_{vTotal} = C_{v1} + C_{v2} + \dots C_{vn}$$

Changes in resistance coefficient, K, required to compensate for different pipe I.D.: **Equation 6-26**

$$K_a = K_b\left(\frac{d_a}{d_b}\right)^4$$

Subscript a refers to the pipe in which the valve will be installed. Subscript b refers to the pipe for which the resistance coefficient K was established.

Representative Resistance Coefficients K for Various Valves and Fittings: The methods and equations for sudden and gradual enlargements and contractions are described on page A-27. The methods for resistance in tees and wyes can be found on page 2-14 to 2-16. Resistance coefficients for valves, elbows, bends, entrances and exits can be found on pages A-28 to A-30.

Discharge of fluid through valves, fittings and pipe; Darcy's formula:
Values of Y are shown on page A-23. For K, Y and ΔP determination, see Examples on page 7-13 and 7-14.

Liquid flow: **Equation 6-27**

$$q = 0.04375\, d^2 \sqrt{\frac{h_L}{K}} = 0.525\, d^2 \sqrt{\frac{\Delta P}{K\rho}}$$

$$Q = 19.64\, d^2 \sqrt{\frac{h_L}{K}} = 235.6\, d^2 \sqrt{\frac{\Delta P}{K\rho}}$$

$$W = 157.5\, \rho d^2 \sqrt{\frac{h_L}{K}} = 1890\, d^2 \sqrt{\frac{\Delta P \rho}{K}}$$

Compressible flow: **Equation 6-28**

$$q'_h = 40700\, Yd^2 \sqrt{\frac{\Delta P\, P'_1}{K\, T_1 S_g}}$$

$$q'_h = 24700\, \frac{Yd^2}{S_g} \sqrt{\frac{\Delta P\, \rho_1}{K}}$$

$$W = 1890\, Yd^2 \sqrt{\frac{\Delta P}{K\, \overline{V}_1}}$$

Flow through orifices, nozzles and venturi:
Values for C can be found from the charts on page A-21 or calculated using C_d. Values for C_d can be found with the methods outlined on pages 4-5 and 4-6. Values for Y can be found on page A-22 or calculated with the methods on page 4-6.

Equation 6-29

$$\beta = \frac{d_1}{d_2}$$

Equation 6-30

$$C = \frac{C_d}{\sqrt{1-\beta^4}}$$

Liquid: **Equation 6-31**

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$$q = CA\sqrt{2gh_L} = CA\sqrt{\frac{2g(144)\Delta P}{\rho}}$$

$$q = 0.04375 d_1^2 C\sqrt{h_L} = 0.525 d_1^2 C\sqrt{\frac{\Delta P}{\rho}}$$

$$Q = 19.64 d_1^2 C\sqrt{h_L} = 235.6 d_1^2 C\sqrt{\frac{\Delta P}{\rho}}$$

$$W = 157.5 d_1^2 C\sqrt{h_L \rho^2} = 1890 d_1^2 C\sqrt{\Delta P \rho}$$

Compressible flow: **Equation 6-32**

www.

$$q = YCA\sqrt{2gh_L} = YCA\sqrt{\frac{2g(144)\Delta P}{\rho}}$$

$$q'_h = 40700 Yd_1^2 C\sqrt{\frac{\Delta P P'_1}{T_1 S_g}} = 24700\frac{Yd_1^2 C}{S_g}\sqrt{\Delta P \rho_1}$$

$$W = 1890 Yd_1^2 C\sqrt{\Delta P \rho_1}$$

Non-Recoverable Pressure Drop (NRPD):
Applicable for ISA1932 and long radius nozzles and orfices. **Equation 6-33**

$$NRPD = \Delta P\left[\frac{\sqrt{1-\beta^4\left(1-C_d^2\right)} - C_d\beta^2}{\sqrt{1-\beta^4\left(1-C_d^2\right)} + C_d\beta^2}\right]$$

Summary of Formulas

Control valve sizing equations:

Incompressible fluids: **Equation 6-34**

$$C_V = \frac{Q}{F_p \sqrt{\frac{P'_1 - P'_2}{S}}}$$

$$F_p = \frac{1}{\sqrt{1 + \frac{\Sigma K}{890}\left(\frac{C_v}{d^2_{nom}}\right)^2}}$$

$$\Sigma K = K_1 + K_2 + K_{B1} - K_{B2}$$

$$K_B = 1 - \left(\frac{d_{nom}}{d}\right)^4$$

For short length concentric reducers: **Equation 6-35**

$$K^{inlet}_{reducer} = 0.5\left[1 - \left(\frac{d_{nom}}{d_1}\right)^2\right]^2$$

$$K^{outlet}_{reducer} = 1.0\left[1 - \left(\frac{d_{nom}}{d_2}\right)^2\right]^2$$

For reducers with $d_1 = d_2 = d$: **Equation 6-36**

$$\Sigma K = 1.5\left[1 - \left(\frac{d_{nom}}{d}\right)^2\right]^2$$

Choked flow conditions for incompressible flow: **Equation 6-37**

$$Q_{max} = F_L C_V \sqrt{\frac{P'_1 - F_F P'_v}{S}}$$ without fittings

$$\Delta P_{max} = F_L^2 (P'_1 - F_F P'_v)$$ without fittings

$$Q_{max} = \left(\frac{F_{LP}}{F_P}\right) C_V \sqrt{\frac{P'_1 - F_F P'_v}{S}}$$ with fittings

$$\Delta P_{max} = \left(\frac{F_{LP}}{F_P}\right)^2 (P'_1 - F_F P'_v)$$ with fittings

$$F_F = 0.96 - 0.28\sqrt{\frac{P'_v}{P'_c}}$$

$$F_{LP} = \frac{F_L}{\sqrt{1 + F_L^2 \frac{\Sigma K_i}{890}\left(\frac{C_v}{d^2_{nom}}\right)^2}}$$

$$\Sigma K_i = K_1 + K_{B1}$$

Compressible fluids: **Equation 6-38**

$$C_v = \frac{W}{63.3 F_p Y \sqrt{x P'_1 \rho_1}}$$

$$C_v = \frac{q'_h}{1360 F_p P'_1 Y \sqrt{\frac{x}{S_g T_1 Z_f}}}$$

$$Y = 1 - \frac{x}{3F_k x_T}$$ without fittings

$$Y = 1 - \frac{x}{3F_k x_{TP}}$$ with fittings

$$x = \frac{\Delta P}{P'_1}$$

$$x_{TP} = \frac{\frac{x_T}{F_p^2}}{1 + \left(\frac{x_T K_i}{1000}\right)\left(\frac{C_v}{d^2_{nom}}\right)^2}$$ with fittings

$$F_k = \frac{k}{1.4}$$

Choked flow conditions for compressible flow occur when: **Equation 6-39**

$$x \geq F_k x_T$$ without fittings

$$x \geq F_k x_{TP}$$ with fittings

Conversion of C_v to K_v: **Equation 6-40**

$$K_v = 0.865 C_v$$

Summary of Formulas

Pump Performance Equations:

Net positive suction head available. **Equation 6-41**

$$NPSH_a = \frac{144}{\rho}\left(P'_t - P'_v\right) + \left(Z_t - Z_s\right) - h_L$$

Pump Affinity Rules:

Change in impeller speed: **Equation 6-42**

Flow rate: $$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$$

Head: $$\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2$$

Power: $$\frac{P_1}{P_2} = \left(\frac{N_1}{N_2}\right)^3$$

Change in impeller diameter: **Equation 6-43**

Capacity : $$\frac{Q_1}{Q_2} = \frac{D_1}{D_2}$$

Head: $$\frac{H_1}{H_2} = \left(\frac{D_1}{D_2}\right)^2$$

Power: $$\frac{P_1}{P_2} = \left(\frac{D_1}{D_2}\right)^3$$

Pump Power Calculations:

Brake (shaft) Horsepower: **Equation 6-44**

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$$bhp = \frac{QH\rho}{247000\eta_p}$$

Electrical Horsepower: **Equation 6-45**

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$$ehp = \frac{bhp}{\eta_m \, \eta_{vsd}}$$

Operating Cost: **Equation 6-46**

$$OC = \frac{0.7457QH\rho}{247000\eta_p \, \eta_m \, \eta_{vsd}} \text{(operating time in hours)}\left(\frac{\$}{kWh}\right)$$

Specific gravity of liquids:

Any Liquid: **Equation 6-47**

$$S = \frac{\rho \text{ (any liquid at 60°F, Unless otherwise specified)}}{\rho\text{(Water at 60°F)}}$$

Oils: **Equation 6-48**

$$S\left(\frac{60°F}{60°F}\right) = \frac{141.5}{131.5 + \text{Deg API}}$$

Liquids lighter than water: **Equation 6-49**

$$S\left(\frac{60°F}{60°F}\right) = \frac{140}{130 + \text{Deg Baume}}$$

Liquids heavier than water: **Equation 6-50**

$$S\left(\frac{60°F}{60°F}\right) = \frac{145}{145 - \text{Deg Baume}}$$

Specific gravity of gases: **Equation 6-51**

$$S_g = \frac{R\text{ (air)}}{R\text{ (gas)}} = \frac{53.35}{R\text{ (gas)}}$$

$$S_g = \frac{M_r\text{ (gas)}}{M_r\text{ (air)}} = \frac{M_r\text{ (gas)}}{28.966}$$

Ideal Gas Equation: **Equation 6-52**

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$$p' \; V_a = w_a \, R \, T$$

$$\rho = \frac{w_a}{V_a} = \frac{p'}{RT} = \frac{144P'}{RT}$$

$$R = \frac{\overline{R}}{M_r} = \frac{1545.35}{M_r} = \frac{144P'}{\rho T}$$

$$p' \; V_a = n_a \, \overline{R} \; T = n_a \, 1545.35 \, T = \frac{w_a}{M_r} \, 1545.35 \, T$$

$$\rho = \frac{w_a}{V_a} = \frac{p' M_r}{1545.35T} = \frac{P' M_r}{10.73 \, T} = \frac{2.70P' Sg}{T}$$

where: $n_a = \frac{w_a}{M_r}$ = number of moles of a gas.

Equivalent Hydraulic Diameter Relationship* **Equation 6-53**

$$R_H = \frac{\text{Cross Sectional Area}}{\text{Wetted Perimeter}}$$

Equivalent diameter relationship

$D_H = 4 \, R_H$

$d_H = 48 \, R_H$

*See page 1-4 for limitations.

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Chapter 7

Examples of Flow Problems

Theory and answers to questions regarding proper application of formulas to flow problems can be presented to good advantage by the solution of practical problems. A few flow problems, both simple and complex, are presented in this chapter.

Many of the examples given in this chapter employ the basic formulas of Chapters 1 through 5; these formulas were rewritten in more commonly used terms for Chapter 6.

The controversial subject regarding the selection of a formula most applicable to the flow of gas through long pipelines is analyzed in Chapter 1. It is shown that the three commonly used formulas are basically identical, the only difference being in the selection of friction factors. A comparison of results obtained, using the three formulas, is presented in this chapter.

An original method has been developed for the solution of problems involving the discharge of compressible fluids from pipe systems. Illustrative examples applying this method demonstrate the simplicity of handling these, heretofore complex, problems.

Reynolds Number and Friction Factor For Pipe Other Than Steel: The example below shows the procedure in obtaining the Reynolds number and friction factor for smooth pipe (plastic). The same procedure applies for any pipe other than steel such as concrete, wood stave, riveted steel, etc. For relative roughness of these and other piping materials, see page A-24.

Example 7-1 Smooth Pipe (Plastic)

Given: Water at 80°F is flowing through 70 feet of 2" standard wall plastic pipe (smooth wall) at a rate of 50 gallons per minute.

Find: The Reynolds number and friction factor.

Solution:

1. $R_e = \dfrac{50.66 Q \rho}{d \mu}$ page 6-2
2. $\rho = 62.212$ page A-7
3. $d = 2.067$ page B-13
4. $\mu = 0.85$ page A-3
5. $R_e = \dfrac{50.66 \times 50 \times 62.212}{2.067 \times 0.85}$ $R_e = 89690$ or 8.969×10^4
6. $f = 0.0182$ page A-25

Determination of Valve Resistance
in L, L/D, K and Flow Coefficient C_v

Example 7-2

L, L/D, and K from C_v for Conventional Type Valves

Given: A 6" Class 125 iron Y-pattern globe valve has a flow coefficient, C_v, of 600.

Find: Resistance coefficient K and equivalent lengths L/D and L for flow in zone of complete turbulence.

Solution:

1. K, L/D, and L should be given in terms of 6" Schedule 40 pipe; see page 2-9.

2. $K = \dfrac{890.3\,d^4}{C_V^{\,2}}$ page 6-4

3. $d = 6.065$ $d^4 = 1352.8$ page B-14

   $D = 0.5054$

4. $K = \dfrac{890.3 \times 1352.8}{600^2} = 3.35$ based on 6" Sched. 40 pipe

5. $\dfrac{L}{D} = \dfrac{K}{f}$ page 6-4

6. $f = 0.015$ for 6.065" I.D. pipe in fully turbulent flow range; A-26

7. $\dfrac{L}{D} = \dfrac{K}{f} = \dfrac{3.35}{0.015} = 223$

8. $L = \left(\dfrac{L}{D}\right) D = 223 \times 0.5054 = 113$

Example 7-3

L, L/D, K, and C_v for Conventional Type Valves

Given: A 4" Class 600 steel conventional angle valve with full area seat.

Find: Resistance coefficient K, flow coefficient C_v, and equivalent lengths L/D and L for flow in zone of complete turbulence.

Solution:

1. K, L/D, and L should be given in terms of 4" Schedule 80 pipe; see page 2-9.

2. $K = 150\, f_T$ page A-28

   $C_V = \dfrac{29.84\,d^2}{\sqrt{K}}$ page 6-4

   $K = f\,\dfrac{L}{D}$ or $\dfrac{L}{D} = \dfrac{K}{f_T}$ page 6-4

(subscript "T"- refers to flow in zone of complete turbulence)

3. $d = 3.826$ page B-14

   $f_T = 0.0165$ page A-26

4. $K = 150 \times 0.0165 = 2.475$ based on 4" Sched. 80 pipe

5. $C_V = \dfrac{29.84 \times 3.826}{\sqrt{2.475}} = 277.4$

6. $\dfrac{L}{D} = \dfrac{2.475}{0.0165} = 150$

7. $L = \dfrac{150 \times 3.826}{12} = 47.8$

Example 7-4 Venturi Type Valves

Given: A 6 x 4" Class 600 steel gate valve with inlet and outlet ports conically tapered from back of body rings to valve ends. Face-to-face dimension is 22" and back of seat ring to back of seat ring is about 6".

Find: K_2 for any flow condition, and L/D and L for flow in zone of complete turbulence.

Solution:

1. K_2, L/D, and L should be given in terms of 6" Sched. 80 pipe; see page 2-9.

2. $K_1 = 8\, f_T$ page A-28

   $K_2 = \dfrac{K_1 + \sin\left(\dfrac{\theta}{2}\right)\left[0.8\left(1-\beta^2\right) + 2.6\left(1-\beta^2\right)^2\right]}{\beta^4}$ page A-27

   $K = f\,\dfrac{L}{D}$ or $\dfrac{L}{D} = \dfrac{K}{f_T}$ page 6-4

   $\beta = \dfrac{d_1}{d_2}$ page A-27

3. $d_1 = 4.00$ valve seat bore

   $d_2 = 5.761$ 6" Sched. 80 pipe; page B-14

   $f_T = 0.015$ for 6" size; page A-26

4. $\beta = \dfrac{4.00}{5.761} = 0.69$ $\beta^2 = 0.48$ $\beta^4 = 0.23$

   $\tan\left(\dfrac{\theta}{2}\right) = \dfrac{0.5(5.761 - 4.00)}{0.5(22 - 6)}$

   $\tan\left(\dfrac{\theta}{2}\right) = 0.110 = \sin\left(\dfrac{\theta}{2}\right)$ approximately

5. $K_2 = \dfrac{8 \times 0.015 + 0.110\left(0.8 \times 0.52 + 2.6 \times 0.52^2\right)}{0.23}$

   $K_2 = 1.06$

6. $\dfrac{L}{D} = \dfrac{1.06}{0.015} = 70$ diameters 6" Sched. 80 pipe

7. $L = \dfrac{70 \times 5.761}{12} = 34$ feet of 6" Sched. 80 pipe

Check Valves
Determination of Size

Example 7-5 Lift Check Valves

Given: A globe type lift check valve with a wing-guided disc is required in a 3" Schedule 40 horizontal pipe carrying 70°F water at the rate of 80 gallons per minute.

Find: The proper size check valve and the pressure drop. The valve should be sized so that the disc is fully lifted at the specified flow; see page 2-6 for discussion.

Solution:

1. $v_{min} = 40\sqrt{\bar{V}}$ page A-28

$v = \dfrac{0.4085Q}{d^2}$ page 6-2

$\Delta P = \dfrac{1.801\times10^{-5}\,K\rho Q^2}{d^4}$ page 6-4

$K_1 = 600\, f_T$ page A-28

$K_2 = \dfrac{K_1 + \beta\left[0.5\left(1-\beta^2\right) + \left(1-\beta^2\right)^2\right]}{\beta^4}$ page A-28

$\beta = \dfrac{d_1}{d_2}$ page A-27

2. $d_1 = 2.469$ for 2 ½" Sched. 40 pipe; page B-13

$d_2 = 3.068$ for 3" Sched. 40 pipe; page B-13

$\bar{V} = 0.01605$ 70°F water; page A-7

$\rho = 62.298$ 70°F water; page A-7

$f_T = 0.018$ for 2 ½" size; page A-27

$f_T = 0.017$ for 3" size; page A-27

3. $v_{min} = 40\sqrt{0.01605} = 5.1$

$v = \dfrac{0.4085\times80}{3.068^2} = 3.47$ for 3" valve

In as much as v is less than v_{min}, a 3" valve will be too large. Try a 2 ½" size.

$v = \dfrac{0.4085\times80}{2.469^2} = 5.36$ for a 2 ½" valve

Based on above, a 2 ½" valve installed in 3" Schedule 40 pipe with reducers is advisable.

4. $\beta = \dfrac{2.469}{3.068} = 0.80 \qquad \beta^2 = 0.64 \qquad \beta^4 = 0.41$

5. $K_2 = \dfrac{600\times0.018 + 0.8\left[0.5(1-0.64) + (1-0.64)^2\right]}{0.41}$

$K_2 = 27$

6. $\Delta P = \dfrac{1.801\times10^{-5}\times27\times62.298\times80^2}{3.068^4} = 2.2$

Reduced Port Valves
Velocity and Rate of Discharge

Example 7-6 Reduced Port Ball Valve

Given: Water at 60°F discharges from a tank with 22-feet average head to atmosphere through:

200 feet - 3" Schedule 40 pipe

6 - 3" standard 90° threaded elbows

1 - 3" flanged ball valve having a 2 ⅜ diameter seat, 16° conical inlet, and 30° conical outlet end. Sharp-edged entrance is flush with the inside of the tank.

Find: Velocity of flow in the pipe and rate of discharge in gallons per minute.

Solution:

1. $h_L = K\dfrac{v^2}{2g}$ or $v = \sqrt{\dfrac{2g\,h_L}{K}}$ page 6-4

$v = 0.4085\left(\dfrac{Q}{d^2}\right)$ or $Q = 2.448vd^2$ page 6-2

$R_e = 50.66\dfrac{Q\rho}{d\mu}$

2. $K = 0.5$ entrance; page A-30

$K = 1.0$ exit; page A-30

$f_T = 0.017 \qquad \rho = 62.364 \qquad \mu = 1.1$ page A-27, A-3, A-7

3. For K (ball valve), page A-29 indicates use of Formula 5. However, when inlet and outlet angles (θ) differ, Formula 5 must be expanded to:

$$K_2 = \frac{K_1 + 0.8\sin\left(\frac{\theta}{2}\right)\left(1-\beta^2\right) + 2.6\sin\left(\frac{\theta}{2}\right)\left(1-\beta^2\right)^2}{\beta^4}$$

4. $\beta = \dfrac{d_1}{d_2} = \dfrac{2.375}{3.068} = 0.77$ page A-27

5. $\sin\left(\dfrac{\theta}{2}\right) = \sin\left(8^\circ\right) = 0.14$ valve inlet

6. $\sin\left(\dfrac{\theta}{2}\right) = \sin\left(15^\circ\right) = 0.26$ valve outlet

7. $K_2 = \dfrac{3\times0.017 + 0.8\times0.14\left(1-0.77^2\right)}{0.77^4} + \dfrac{2.6\times0.26\left(1-0.77^2\right)^2}{0.77^4} = 0.59$ valve

$K = 6\times30\,f_T = 180\times0.017 = 3.06$ 6 elbows; page A-30

$K = f\dfrac{L}{D} = \dfrac{0.017\times200\times12}{3.068} = 13.30$ pipe; page 6-4

8. Then, for entire system (entrance, pipe, ball valve, six elbows, and exit),

$K = 0.5 + 13.30 + 0.59 + 3.06 + 1.0 = 18.45$

9. $v = \sqrt{(64.4\times22)/18.45} = 8.76$

$Q = 2.448\times8.76\times3.068^2 = 201.8$

10. Calculate Reynolds number to verify that friction factor of 0.017 (zone of complete turbulence) is correct for flow condition $R_e = 50.66\dfrac{201.8\times62.364}{3.068\times1.1} = 1.89\times10^5$

11. Enter chart on page A-26 at $R_e = 1.89\times10^5$

Note *f* for 3" pipe is less than 0.02. Therefore, flow is in the transition zone (slightly less than fully turbulent) but the difference is small enough to forego any correction of K for the pipe.

Laminar Flow in Valves, Fittings, and Pipe

In flow problems where viscosity is high, calculate the Reynolds Number to determine whether the flow is laminar or turbulent.

Example 7-7

Given: S.A.E. 10W Oil at 60°F flows through the system described in Example 7-6 at the same differential head.

Find: The velocity in the pipe and rate of flow in gallons per minute.

Solution:

1. $h_L = \frac{K v^2}{2 g}$ page 6-4

$v = \sqrt{\frac{2 g h_L}{K}}$

$v = 0.4085 \frac{Q}{d^2}$ page 6-2

$Q = 2.448 v d^2$

$R_e = 124 \frac{d v \rho}{\mu}$ page 6-2

$f = \frac{64}{R_e}$ pipe, laminar flow; page 6-2

$K = f \frac{L}{D}$ pipe; page 6-4

2. $K_2 = 0.59$ valve; Example 7-6

$K = 3.06$ 6 elbows; Example 7-6

$K = 0.5$ entrance; Example 7-6

$K = 1.0$ exit; Example 7-6

$\rho = 54.5$ page A-8

$\mu = 75$ page A-3

$h_L = 22$ Example 7-6

3. *Assume laminar flow with v = 5.

$R_e = \frac{124 \times 3.068 \times 5 \times 54.5}{75} = 1382.2$

$f = \frac{64}{1382.2} = 0.046$ pipe

$K = \frac{0.046 \times 200 \times 12}{3.068} = 36$ pipe

$K = 36 + 0.59 + 3.06 + 0.5 + 1.0$

$K = 41.15$ entire system

4. $v = \sqrt{\frac{64.4 \times 22}{41.15}} = 5.87$

5. $Q = 2.448 \times 5.87 \times 3.068^2 = 135.3$

*Note: This problem has two unknowns and, therefore, requires a trial-and-error solution. Two or three trial assumptions will usually bring the solution and final assumption into agreement within desired limits.

Example 7-8

Given: S.A.E. 50 Oil at 100°F is flowing at the rate of 600 barrels per hour through 200 feet of 8" Schedule 40 pipe, in which an 8" conventional globe valve with full area seat is installed.

Find: The pressure drop due to flow through the pipe and valve.

Solution:

1. $\Delta P = \frac{1.801 \times 10^{-5} K \rho Q^2}{d^4}$ page 6-4

$R_e = \frac{50.66\ Q\rho}{d\mu}$ page 6-2

$K_1 = 340\ f_T$ valve; page A-28

$K = f \frac{L}{D}$ pipe; page 6-4

$f = \frac{64}{R_e}$ pipe

2. $S = 0.893$ at 60°F page A-8

$S = 0.875$ at 100°F page A-8

$d = 7.981$ 8" Sched. 40 pipe; page B-14

$\mu = 470$ page A-3

$f_T = 0.014$ page A-27

$Q = \frac{600\ \text{bbl}}{\text{hr}} \Big| \frac{42\ \text{gal}}{\text{bbl}} \Big| \frac{\text{hr}}{60\ \text{min}} = 420\ \text{gpm}$

3. $\rho = 62.364 \times 0.875 = 54.57$

$R_e = \frac{50.66 \times 420 \times 54.57}{7.981 \times 470} = 309.5$

$R_e < 2000$

4. $f = \frac{64}{309.5} = 0.207$ pipe

$K_1 = 340 \times 0.014 = 4.76$ valve

$K = \frac{0.207 \times 200 \times 12}{7.981} = 62.18$ pipe

$K = 4.76 + 62.18 = 66.94$ total system

5. $\Delta P = \frac{1.801 \times 10^{-5} \times 66.94 \times 54.57 \times 420^2}{7.981^4}$

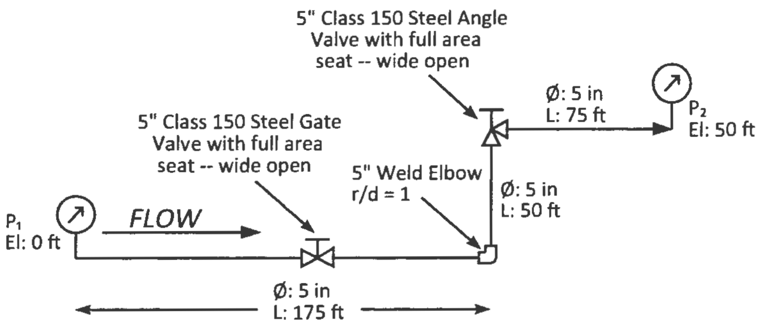
$\Delta P = 2.86$

Laminar Flow in Valves, Fittings and Pipe

In flow problems where viscosity is high, calculate the Reynolds Number to determine whether the flow is laminar or turbulent.

Example 7-9

Given: S.A.E. 50 Oil at 100°F is flowing through 5" Schedule 40 pipe at a rate of 600 gallons per minute, as shown in the following sketch.



Find: The velocity in feet per second and pressure difference between gauges P_1 and P_2.

Solution:

1. $v = \dfrac{0.4085Q}{d^2}$ — page 6-2

   $R_e = \dfrac{50.66Q\rho}{d\,\mu}$ — page 6-2

   $\Delta P = \dfrac{1.801 \times 10^{-5} K\rho Q^2}{d^4}$ — loss due to flow; page 6-4

   $\Delta P = \dfrac{h_L \rho}{144}$ — loss due to elevation change; 6-2

2. $K_1 = 8\, f_T$ — gate valve; page A-28

   $K_1 = 150\, f_T$ — angle valve; page A-28

   $K = 20\, f_T$ — elbow; page A-30

   $K = f\, \dfrac{L}{D}$ — pipe; page 6-4

   $f = \dfrac{64}{R_e}$ — pipe; page 6-2

3. $d = 5.047$ — 5" Sched. 40 pipe; page B-14

   $S = 0.893$ at 60°F — page A-8

   $S = 0.875$ at 100°F — page A-8

   $\mu = 470$ — page A-3

   $\rho = 62.364 \times 0.875 = 54.57$

   $f_T = 0.015$ — page A-27

4. $R_e = \dfrac{50.66 \times 600 \times 54.57}{5.047 \times 470} = 699.24$

   $R_e < 2000$ therefore flow is laminar.

5. $f = \dfrac{64}{699.24} = 0.0915$

6. Summarizing K for the entire system (gate valve, angle valve, elbow, and pipe)

   $K = (8 \times 0.015) + (150 \times 0.015) + (20 \times 0.015) +$

   $\dfrac{(0.0915 \times 300 \times 12)}{5.047} = 67.96$

7. $v = \dfrac{0.4085 \times 600}{5.047^2} = 9.6$

8. $\Delta P = \dfrac{1.801 \times 10^{-5} \times 67.96 \times 54.57 \times 600^2}{5.047^4} + \dfrac{50 \times 54.57}{144}$

   $\Delta P = 56.0$ total

Pressure Drop and Velocity in Piping Systems

Example 7-10 Piping Systems - Steam

Given: 600 psig steam at 850°F flows through 400 feet of horizontal 6" Schedule 80 pipe at a rate of 90,000 pounds per hour. The system contains three 90 degree weld elbows having a relative radius of 1.5, one fully-open 6 x 4" Class 600 venturi gate valve as described in Example 7-4, and one 6" Class 600 y-pattern globe valve. Latter has a seat diameter equal to 0.9 of the inside diameter of Schedule 80 pipe, disc fully lifted.

Find: The pressure drop through the system.

Solution:

1. $\Delta P = \dfrac{2.799 \times 10^{-7}\, K W^2 \bar{V}}{d^4}$ — page 6-4

2. For globe valve (see page A-28), and formula 7 page A-27

$$K_2 = \frac{K_1 + \beta\left[0.5\left(1-\beta^2\right) + \left(1-\beta^2\right)^2\right]}{\beta^4}$$

$K_1 = 55\, f_T$

$\beta = 0.9$

3. $K = 14\, f_T$ — 90° weld elbows; page A-30

$K = f\,\dfrac{L}{D}$ — pipe; page 6-4

$R_e = 6.315\,\dfrac{W}{d\mu}$ — page 6-2

4. $d = 5.761$ — 6" Sched. 80 pipe; page B-14

$\bar{V} = 1.430$ — 600 psi steam, 850°F; page A-18 by linear interpolation

$\mu = 0.027$ — page A-2

$f_T = 0.015$ — page A-26

5. For globe valve,

$$K_2 = \frac{55 \times 0.015 + 0.9\left[0.5\left(1-0.9^2\right) + \left(1-0.9^2\right)^2\right]}{0.9^4}$$

$K_2 = 1.44$

6. $R_e = \dfrac{6.315 \times 90000}{5.761 \times 0.027} = 3.65 \times 10^6$

$f = 0.015$ — pipe; page A-26

$K = \dfrac{0.015 \times 400 \times 12}{5.761} = 12.5$ — pipe

$K = 3 \times 14 \times 0.015 = 0.63$ — 3 elbows; page A-30

$K_2 = 1.44$ — 6 x 4" gate valve; Example 7-4

7. Summarizing K for the entire system (globe valve, pipe, venturi gate valve and elbows)

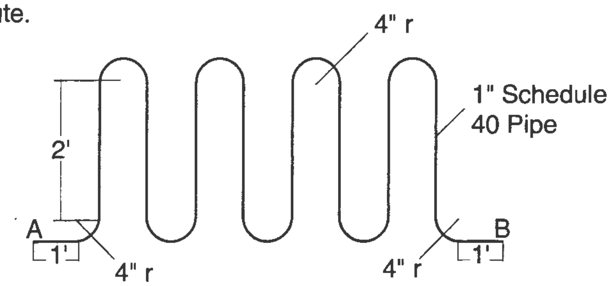
$K = 1.44 + 12.5 + 0.63 + 1.44 = 16$

8. $\Delta P = \dfrac{2.799 \times 10^{-7} \times 16 \times 9^2 \times 10^8 \times 1.430}{5.761^4}$

$\Delta P = 47.1$

Example 7-11 Flat Heating Coils - Water

Given: Water at 180°F is flowing through a flat heating coil, shown in the sketch below, at a rate of 15 gallons per minute.



Find: The pressure drop from Point A to B.

Solution:

1. $\Delta P = \dfrac{1.801 \times 10^{-5} K \rho Q^2}{d^4}$ — page 6-4

$R_e = \dfrac{50.66\, Q\rho}{d\,\mu}$ — page 6-2

$K = f\,\dfrac{L}{D}$ — straight pipe; page 6-4

$\dfrac{r}{d} = 4$ — pipe blends

$K_{90} = 14\, f_T$ — 90° bends; page A-30

$K_B = (\eta - 1)\left(0.25\,\pi\, f_T\,\dfrac{r}{d} + 0.5\,K_{90}\right) + K_{90}$ — 180° bends; page A-30

2. $\rho = 60.58$ — water, 180°F; page A-7

$\mu = 0.34$ — water, 180°F; page A-3

$d = 1.049$ — 1" Sched. 40 pipe; page B-13

$f_T = 0.022$ — 1" Sched. 40 pipe; page A-27

3. $R_e = \dfrac{50.66 \times 15 \times 60.58}{1.049 \times 0.34} = 1.29 \times 10^5$

$f = 0.024$ — pipe

$K = \dfrac{0.024 \times 18 \times 12}{1.049} = 4.94$ — 18" straight pipe

$K = 2 \times 14 \times 0.022 = 0.616$ — two 90° bends

4. For seven 180° bends,

$K_B = 7[(2-1)(0.25\pi \times 0.022 \times 4) + (0.5 \times 0.308) + 0.308] = 3.72$

5. $K_{TOTAL} = 4.94 + 0.616 + 3.72 = 9.28$

6. $\Delta P = \dfrac{1.801 \times 10^{-5} \times 9.28 \times 60.58 \times 15^2}{1.049^4} = 1.88$

Pressure Drop and Velocity in Piping Systems

Example 7-12
Orifice Size for Given Pressure Drop and Velocity

Given: A 12" Schedule 40 steel pipe 60' long, containing a standard gate valve 10' from the entrance, discharges 60°F water to atmosphere from a reservoir. The entrance projects inward into the reservoir and its center line is 12' below the water level in the reservoir.

Find: The diameter of thin-plate orifice that must be centrally installed in the pipe to restrict the velocity of flow to 10' per second when the gate valve is wide open.

Solution:

1. $h_L = K \frac{v^2}{2g}$ or system $K = \frac{2gh_L}{v^2}$ page 6-4

   $R_e = \frac{124.0\, dv\rho}{\mu}$ page 6-2

2. $K = 0.78$ entrance; page A-30

   $K = 1.0$ exit; page A-30

   $K_1 = 8 f_T$ gate valve; page A-28

   $K = f \frac{L}{D}$ pipe; page 6-4

3. $d = 11.938$ pipe; page B-14

   $f_T = 0.013$ page A-27

   $\rho = 62.364$ page A-7

   $\mu = 1.1$ page A-3

4. $R_e = \frac{124.0 \times 11.938 \times 10 \times 62.364}{1.1} = 8.39 \times 10^5$

   $f = 0.014$ page A-26

5. total K required $= 64.4 \times \frac{12}{10^2} = 7.73$

   $K_1 = 8 \times 0.013 = 0.10$ gate valve

   $K = 60 \times 0.014 = 0.84$ pipe

   Then, exclusive of orifice,
   $K_{total} = 0.78 + 1.0 + 0.1 + 0.84 = 2.72$

6. $K_{orifice} = 7.73 - 2.72 = 5.01$

7. $K_{orifice} = \left[\frac{\sqrt{1 - \beta^4(1 - C_d^2)}}{C_d\beta^2} - 1\right]^2$ and $C_d = C\sqrt{1 - \beta^4}$

8. Assume $\beta = 0.7$ $\therefore C = 0.7$ page A-21
   then Cd = 0.6102 and $K_{orifice} \cong 4.34$ $\therefore \beta$ is too large

9. Assume $\beta = 0.65$ $\therefore C = 0.67$ page A-21
   then Cd = 0.6073 and $K_{orifice} \cong 7.14$ $\therefore \beta$ is too small

10. Assume $\beta = 0.67$ $\therefore C = 0.682$ page A-21
    then Cd = 0.6094 and $K_{orifice} \cong 5.84$ $\therefore$ use $\beta = 0.665$

11. Orifice size $\cong 11.938 \times 0.665 = 7.94$"

Example 7-13
Flow Given in International Metric System (SI) Units-Oil

Given: Fuel oil with a density of 0.815 grams per cubic centimeter and a kinematic viscosity of 2.7 centistokes is flowing through 50 millimeter I.D. steel pipe (30 meters long) at a rate of 7.0 liters per second.

Find: Head loss in meters of fluid and pressure drop in kg/cm², bar, and megapascal (MPa).

Solution:

1. Define symbols in SI units as follows:

   A ...cross-sectional area of pipe, in meters²
   D ...internal diameter of pipe, in meters
   g ...acceleration of gravity = 9.8 meters/sec/sec
   h_L...head loss, in meters of fluid
   L ...length of pipe, in meters
   q ...rate of flow, in meters³/second
   v ...mean velocity of flow, in meters/second
   ρ'...fluid density, in grams/centimeter³
   ΔP (kg/cm²) ...pressure drop, in kilograms/centimeter²
   ΔP (bar) ...pressure drop, in bars
   ΔP (MPa) ...pressure drop, in megapascals

2. Use metric-imperial equivalents as indicated below and on pages B-8 and B-10.

   meter (1) = 3.28 feet = 39.37 inches
   bar = 0.98067 x kg/cm²
   megapascal = 0.098067 x kg/cm²

   A column of fluid one square centimeter in cross sectional area and one meter high is equal to a pressure of 0.1 ρ kg/cm²; therefore:

   ΔP (kg/cm²) = $0.1\rho h_L$
   ΔP (bar) = 0.98067 ΔP (kg/cm²)
   ΔP (MPa) = 0.098067 ΔP (kg/cm²)

3. $v = \frac{q}{A} = \frac{7 \times 10^{-3}}{\left(\frac{\pi}{4}\right) \times 50^2 \times 10^{-6}} = 3.566$ page 6-2

   $R_e = \frac{7742 \times 39.37D \times 3.28v}{\nu} = \frac{Dv \times 10^6}{\nu}$ page 6-2

   $R_e = \frac{0.050 \times 3.566 \times 10^6}{2.7} = 6.6 \times 10^4$

   $f = 0.023$ page A-26

4. $h_L = f \frac{L}{D} \frac{v^2}{2g} = \frac{0.023 \times 30 \times 3.566^2}{0.050 \times 2 \times 9.8} = 8.95$ page 6-4

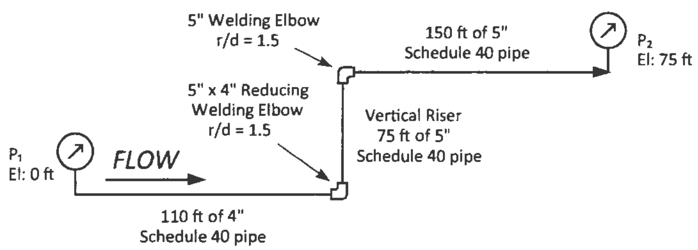
   $\Delta P \left(\frac{kg}{cm^2}\right) = 0.1 \times 0.815 \times 8.95 = 0.729$

   $\Delta P\ (bar) = 0.98067 \times 0.729 = 0.715$

   $\Delta P\ (MPa) = 0.098067 \times 0.729 = 0.0715$

Example 7-14 Bernoulli's Theorem-Water

Given: Water at 60°F is flowing through the piping system, shown in the sketch below, at a rate of 400 gallons per minute.



Find: The velocity in both the 4 and 5" pipe sizes and the pressure differential between gauges P_1 and P_2.

Solution:

1. Use Bernoulli's theorem (see page 6-2):

$$Z_1 + 144\frac{P_1}{\rho_1} + \frac{v_1^2}{2g} = Z_2 + 144\frac{P_2}{\rho_2} + \frac{v_2^2}{2g} + h_L$$

Since, $\rho_1 = \rho_2$

$$P_1 - P_2 = \frac{\rho}{144}\left[(Z_2 - Z_1) + \frac{v_2^2 - v_1^2}{2g} + h_L\right]$$

2. $h_L = \dfrac{0.002593\,KQ^2}{d^4}$ — page 6-4

$R_e = \dfrac{50.66\,Q\rho}{d\,\mu}$ — page 6-2

$K = f\dfrac{L}{D}$ — page 6-4

$K = \dfrac{f\,L}{D\,\beta^4}$ — small pipe, in terms of larger pipe; page 2-9

$K = 14\,f_T$ — 90° elbow; page A-30

$K = 14\,f_T + \dfrac{\left(1-\beta^2\right)^2}{\beta^4}$ — reducing 90° elbow; page A-27

Note: In the absence of test data for increasing elbows, the resistance is conservatively estimated to be equal to the summation of the resistance due to a straight size elbow and a sudden enlargement using formula 4.

$\beta = \dfrac{d_1}{d_2}$ — page A-27

3. $\rho = 62.364$ — page A-7

$\mu = 1.1$ — page A-3

$d_1 = 4.026$ — 4" Sched. 40 pipe; page B-14

$d_2 = 5.047$ — 5" Sched. 40 pipe; page B-14

$f_T = 0.015$ — 5" size; page A-27

4. $\beta = \dfrac{4.026}{5.047} = 0.80$

$Z_2 - Z_1 = 75 - 0 = 75$ feet

$v_1 = 10.08$ — 4" pipe, page B-11

$v_2 = 6.41$ — 5" pipe, page B-11

$$\frac{v_2^2 - v_1^2}{2g} = \frac{6.41^2 - 10.08^2}{2 \times 32.2} = -0.94 \text{ feet}$$

5. For Schedule 40 pipe,

$R_e = \dfrac{50.66 \times 400 \times 62.364}{4.026 \times 1.1} = 2.85 \times 10^5$ — 4" pipe

$R_e = \dfrac{50.66 \times 400 \times 62.364}{5.047 \times 1.1} = 2.28 \times 10^5$ — 5" pipe

$f = 0.018$ — 4" or 5" pipe; page A-26

6. $K = \dfrac{0.018 \times 225 \times 12}{5.047}$ or

$K = 9.6$ — for 225' of 5" Sched. 40 pipe

$K = \dfrac{0.018 \times 110 \times 12}{4.026}$ or

$K = 5.9$ — for 110' of 4" Sched. 40 pipe

With reference to velocity in 5" pipe,

$K_2 = \dfrac{5.9}{0.8^4} = 14.4$

$K = 14 \times 0.015 = 0.21$ — 5" 90° elbow

$K = 0.21 + \dfrac{0.36^2}{0.8^4} = 0.53$ — 5 x 4" 90° elbow

7. Then, in terms of 5" pipe,

$K_{TOTAL} = 9.6 + 14.4 + 0.21 + 0.53 = 24.7$

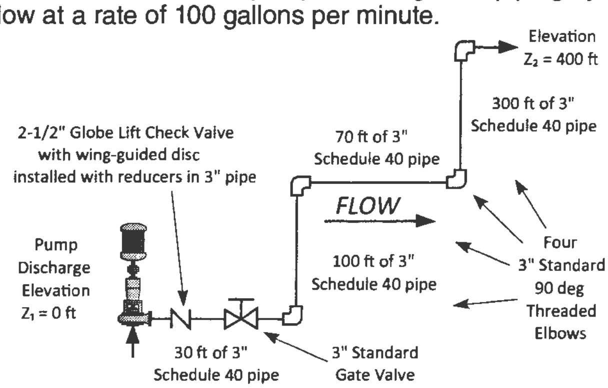
8. $h_L = \dfrac{0.002593 \times 24.7 \times 400^2}{5.047^4} = 15.8$

9. $P_1 - P_2 = \dfrac{62.364}{144}(75 - 0.94 + 15.8) = 38.9$

Pressure Drop and Velocity in Piping Systems

Example 7-15 Power Required for Pumping

Given: Water at 70°F is pumped through the piping system below at a rate of 100 gallons per minute.



Find: The total discharge head (H) at flowing conditions and the brake horsepower (bhp) required for a pump having an efficiency (η_p) of 70 percent.

Solution: 1. Use Bernoulli's theorem (see page 6-2):

$$Z_1 + 144\frac{P_1}{\rho_1} + \frac{v_1^2}{2g} = Z_2 + 144\frac{P_2}{\rho_2} + \frac{v_2^2}{2g} + h_L$$

2. Since $\rho_1 = \rho_2$ and $v_1 = v_2$, the equation can be rewritten to establish the pump head, H:

$$\frac{144}{\rho}\left(P_1 - P_2\right) = \left(Z_2 - Z_1\right) + h_L$$

3. $h_L = \dfrac{0.002593\,K\,Q^2}{d^4}$ — page 6-4

$R_e = 124.0\dfrac{d\,v\,\rho}{\mu}$ — page 6-2

$v = \dfrac{0.4085Q}{d^2}$ — page 6-2

$bhp = \dfrac{Q\,H\,\rho}{247000\,\eta_p}$ — page B-7

4. $K = 30\,f_T$ — 90° elbow; page A-30

$K_1 = 8\,f_T$ — gate valve; page A-28

$K = f\dfrac{L}{D}$ — straight pipe; page 6-4

$K = 1.0$ — exit; page A-30

5. $d = 3.068$ — 3" Sched. 40 pipe; page B-13

$\rho = 62.298$ — page A7

$\mu = 0.95$ — page A-3

$f_T = 0.017$ — page A-26

6. $v = \dfrac{0.4085 \times 100}{3.068^2} = 4.34$

$R_e = \dfrac{124.0 \times 3.068 \times 4.34 \times 62.298}{0.95} = 1.08 \times 10^5$

$f = 0.021$ — page A-25

7. $K = 4 \times 30 \times 0.017 = 2.04$ — four 90° elbows

$K_1 = 8 \times 0.017 = 0.14$ — gate valve

$K = 27.0$ — lift check valve with reducers; Example 7-5

For 500 feet of 3" Schedule 40 pipe,

$$K = \frac{0.021 \times 500 \times 12}{3.068} = 41.07$$

$$K_{TOTAL} = 2.04 + 0.14 + 27.0 + 41.07 + 1 = 71.3$$

8. $h_L = \dfrac{0.002593 \times 71.3 \times 100^2}{3.068^4} = 21$

9. $H = 400 + 21 = 421$

$$bhp = \frac{100 \times 421 \times 62.298}{247000 \times 0.70} = 15.2$$

Example 7-16 Air Lines

Given: Air at 65 psig and 110°F is flowing through 75' of 1" Schedule 40 pipe at a rate of 100 standard cubic feet per minute (scfm).

Find: The pressure drop in pounds per square inch and the velocity in feet per minute at both up stream and downstream gauges.

Solution:

1. Referring to the table on page B-12, read pressure drop of 2.21 psi for 100 psi, 60°F air at a flow rate of 100 scfm through 100 feet of 1" Schedule 40 pipe.

2. Correction for length, pressure, and temperature (page B-12):

$$\Delta P = 2.21\left(\frac{75}{100}\right)\left(\frac{100 + 14.7}{65 + 14.7}\right)\left(\frac{460 + 110}{520}\right)$$

$$\Delta P = 2.61$$

3. To find the velocity, the rate of flow in cubic feet per minute at flowing conditions must be determined from page B-12.

$$q_m = q'_m\left(\frac{14.7}{14.7 + P}\right)\left(\frac{460 + t}{520}\right)$$

At upstream gauge:

$$q_m = 100\left(\frac{14.7}{14.7 + 65}\right)\left(\frac{460 + 110}{520}\right) = 20.2$$

At downstream gauge:

$$q_m = 100\left[\frac{14.7}{14.7 + (65 - 2.61)}\right]\left(\frac{460 + 110}{520}\right) = 20.9$$

4. $V = \dfrac{q_m}{A}$ since $V = \dfrac{q}{A}$ — page 6-2

5. $A = 0.006$ — page B-13

6. $V = \dfrac{20.2}{0.006} = 3367$ — at upstream gauge

$V = \dfrac{20.9}{0.006} = 3483$ — at downstream gauge

Note: Example 7-16 may also be solved by use of the pressure drop formula shown on page 6-2 or the velocity formula shown on page 6-2.

Pipeline Flow Problems

Example 7-17 Sizing of Pump for Oil Pipelines

Given: Crude oil 30 degree API at 15.6°C with a viscosity of 75 Universal Saybolt seconds is flowing through a 12" Schedule 30 steel pipe at a rate of 1900 barrels per hour. The pipeline is 50 miles long with discharge at an elevation of 2,000 feet above the pump inlet. Assume the pump has an efficiency of 67 percent.

Find: The brake horsepower of the pump.

Solution:

1. $\Delta P = 2.161 \times 10^{-4} \left(\frac{f L \rho Q^2}{d^5} \right)$ page 6-3

   1 Barrel = 42 us gal page B-8

   $t = 1.8\, t_e + 32$ page B-10

   $R_e = 50.66 \frac{Q \rho}{d \mu}$ page 6-2

   $h_L = \frac{144 \Delta P}{\rho}$ page 6-2

   brake horsepower = QHp $bhp = \frac{Q H \rho}{247000\, \eta_p}$ page B-7

2. $t = (1.8 \times 15.6) + 32 = 60\,^{\circ}F$

3. $Q = \left(\frac{1900 \text{ bbl}}{\text{hr}} \right) \left(\frac{42 \text{ gal}}{\text{bbl}} \right) \left(\frac{\text{hr}}{60 \text{ min}} \right) = 1330$

4. $\rho = 54.64$ page B-6

   $S = 0.8762$ page B-6

5. $d = 12.09$ page B-14

   $d^5 = 258304$

6. 75 USS = 12.5 centipoise page B-5

7. $R_e = \frac{50.66 \times 1330 \times 54.64}{12.09 \times 12.5} = 24360$

8. $f = 0.025$ page A-26

9. $\Delta P = 2.161 \times 10^{-4} \left(\frac{0.025 \times 50 \times 5280 \times 54.64 \times 1330^2}{258304} \right)$

   $\Delta P = 533.7$

10. $h_L = \frac{144 \times 533.7}{54.64} = 1406.5$

11. The total discharge head at the pump is:

    $H = 1406.5 + 2000 = 3406.5$

12. Then, the brake horsepower is:

    $bhp = \frac{1330 \times 3406.5 \times 54.64}{247000 \times 0.67} = 1496$

Pipeline Flow Problems

Example 7-18 Gas

Given: A natural gas pipeline, made of 14" Schedule 20 pipe, is 100 miles long. The inlet pressure is 1300 psia, the outlet pressure is 300 psia, and the average temperature is 40°F. The gas consists of 75% methane (CH_4), 21% ethane (C_2H_6), and 4% propane (C_3H_8).

Find: The flow rate in millions of standard cubic feet per day (MMscfd).

Solutions: Three solutions to this example are presented for the purpose of illustrating the variations in results obtained by use of the Simplified Isothermal Flow, Weymouth, and the Panhandle A equations.

Simplified Isothermal Equation (page 6-3)

1. $$q'_h = 3.2308\left(\frac{T_b}{P'_b}\right)\left[\frac{(P'_1)^2 - (P'_2)^2}{f\,L_m\,T_{avg}\,S_g}\right]^{0.5} d^{2.5}$$

2. $d = 13.376$ — page B-15

   $d^{2.5} = 654.36$

3. $f = 0.0128$ — turbulent flow assumed; page A-26

4. $T = 460 + t = 460 + 40 = 500$

5. Approximate atomic masses:

   Carbon $C = 12.0$
   Hydrogen $H = 1.0$

6. Approximate molecular masses:

   Methane (CH_4)
   $M_r = (1 \times 12.0) + (4 \times 1.0) = 16$
   Ethane (C_2H_6)
   $M_r = (2 \times 12.0) + (6 \times 1.0) = 30$
   Propane (C_3H_8)
   $M_r = (3 \times 12.0) + (8 \times 1.0) = 44$
   Natural Gas
   $M_r = (16 \times 0.75) + (30 \times 0.21) + (44 \times 0.04)$
   $M_r = 20.06$

7. $$S_g = \frac{M_r\,(\text{gas})}{M_r\,(\text{air})} = \frac{20.06}{28.966} = 0.693$$ page 6-7

8. $$q'_h = 3.2308\left(\frac{520}{14.7}\right)\left[\frac{(1300^2 - 300^2)}{0.0128 \times 100 \times 500 \times 0.693}\right]^{0.5}(654.36)$$

   $q'_h = 4490000$

9. $$q'_d = \left(\frac{4490000\text{ft}^3}{1000000\text{hr}}\right)\left(\frac{24\text{hr}}{\text{day}}\right) = 107.8$$

10. $$R_e = \frac{0.4821\,q'_h\,S_g}{d\,\mu}$$ page 6-2

11. $\mu = 0.011$ — estimated; page A-6

12. $$R_e = \frac{0.4821 \times 4490000 \times 0.693}{13.376 \times 0.011}$$

    $R_e = 10200000$ or 1.02×10^7

13. $f = 0.0128$ — page A-26

14. Since the assumed friction factor ($f = 0.0128$) is correct, the flow rate is 107.8 MMscfd. If the assumed friction factor were incorrect, it would have to be adjusted and Steps 8, 9, 12, and 13 repeated until the assumed friction factor was in reasonable agreement with that based upon the calculated Reynolds number.

Weymouth Equation (page 6-3)

15. $$q'_h = 18.062\left(\frac{T_b}{P'_b}\right)\left[\frac{(P'_1)^2 - (P'_2)^2}{L_m\,T_{avg}\,S_g}\right]^{0.5} d^{2.667}$$

16. $d^{2.667} = 1009$

17. $$q'_h = 18.062\left(\frac{520}{14.7}\right)\left[\frac{1300^2 - 300^2}{100 \times 500 \times 0.693}\right]^{0.5}(1009)$$

    $q'_h = 4380000$

18. $$q'_d = \left(\frac{4380000\text{ft}^3}{1000000\text{hr}}\right)\left(\frac{24\text{hr}}{\text{day}}\right) = 105.1$$

Panhandle A Equation (page 6-3)

19. $$q'_h = 18.161\,E\left(\frac{T_b}{P'_b}\right)^{1.0788}\left[\frac{(P'_1)^2 - (P'_2)^2}{L_m\,T_{avg}\,S_g^{0.8539}}\right]^{0.5394} d^{2.6182}$$

20. Assume average operation conditions; then efficiency is 92 percent:

    $E = 0.92$

21. $d^{2.6182} = 889$

22. $$q'_h = 18.161 \times 0.92\left(\frac{520}{14.7}\right)^{1.0788}\left[\frac{(1300^2 - 300^2)}{100 \times 500 \times 0.693^{0.8539}}\right]^{0.5394}(889)$$

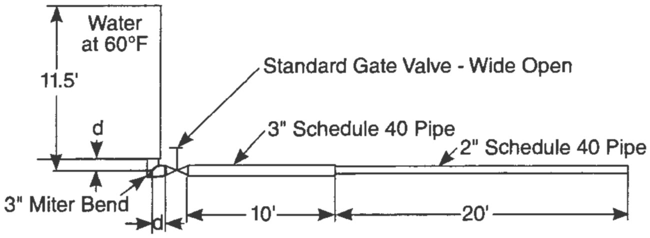
    $q'_h = 5340000$

23. $$q'_d = \left(\frac{5340000\text{ft}^3}{1000000\text{hr}}\right)\left(\frac{24\text{hr}}{\text{day}}\right) = 128.2$$

Discharge of Fluids from Piping Systems

Example 7-19 Water

Given: Water at 60°F is flowing from a reservoir through the piping system below. The reservoir has a constant head of 11.5'.



Find: The flow rate in gallons per minute.

Solution:

1. $Q = 19.64\, d^2 \sqrt{\frac{h_L}{K}}$ page 6-5

$R_e = \frac{50.66\, Q \rho}{d \mu}$ page 6-2

$\beta = \frac{d_1}{d_2}$ page A-27

2. $K = 0.5$ entrance; page A-30

$K = 60\, f_T$ mitre bend; page A-30

$K_1 = 8\, f_T$ gate valve; page A-28

$K = f\, \frac{L}{D}$ straight pipe; page 6-4

$K_2 = \frac{0.5\left(1 - \beta^2\right)\sqrt{\sin\frac{\theta}{2}}}{\beta^4}$ sudden contraction; page A-27

$K = \frac{f\,L}{D\,\beta^4}$ small pipe in terms of larger pipe, page 2-9

$K = \frac{1}{\beta^4}$ exit from small pipe in terms of larger pipe

3. $d = 2.067$ 2" Sched. 40 pipe; page B-13

$d = 3.068$ 3" Sched. 40 pipe; page B-13

$\mu = 1.1$ page A-3

$\rho = 62.364$ page A-7

$f_T = 0.019$ 2" pipe; page A-27

$f_T = 0.017$ 3" pipe; page A-27

4. $\beta = \frac{2.067}{3.068} = 0.67$

$K = 0.5$ 3" entrance

$K = 60 \times 0.017 = 1.02$ 3" mitre bend

$K_1 = 8 \times 0.017 = 0.14$ 3" gate valve

$K = \frac{0.017 \times 10 \times 12}{3.068} = 0.67$ 10 feet, 3" pipe

For 20 feet of 2" pipe, in terms of 3" pipe,

$K = \frac{0.019 \times 20 \times 12}{2.067 \times 0.67^4} = 10.9$

For 2" exit, in terms of 3" pipe,

$K_1 = \frac{1}{0.67^4} = 5.0$

For sudden contraction,

$K_2 = \frac{0.5\left(1 - 0.67^2\right)\cdot 1}{0.67^4} = 1.37$ and,

$K_{TOTAL} = 0.5 + 1.02 + 0.14 + 0.67 + 10.9 + 5.0 + 1.37 = 19.6$

5. $Q = 19.64 \times 3.068^2 \sqrt{\frac{11.5}{19.6}} = 142$

(this solution assumes flow in fully turbulent zone)

6. Calculate Reynolds numbers and check friction factors for flow in straight pipe of the 2" size:

$R_e = \frac{50.66 \times 142 \times 62.364}{2.067 \times 1.1} = 1.97 \times 10^5$

$f = 0.021$ page A-26

and for flow in straight pipe of the 3" size:

$R_e = \frac{50.66 \times 142 \times 62.364}{3.068 \times 1.1} = 1.33 \times 10^5$

$f = 0.020$ page A-26

7. Since assumed friction factors used for straight pipe in Step 4 are not in agreement with those based on the approximate flow rate, the K factors for these items and the total system should be corrected accordingly.

$K = \frac{0.020 \times 10 \times 12}{3.068} = 0.78$ 10 feet, 3" pipe

For 20 feet of 2" pipe, in terms of 3" pipe,

$K = \frac{0.021 \times 20 \times 12}{2.067 \times 0.67^4} = 12.1$ and,

$K_{TOTAL} = 0.5 + 1.02 + 0.14 + 0.78 + 12.1 + 5.0 + 1.37 = 20.9$

8. $Q = 19.64 \times 3.068^2 \sqrt{\frac{11.5}{20.9}} = 137$

Pipeline Flow Problems

Example 7-18 Gas

Given: A natural gas pipeline, made of 14" Schedule 20 pipe, is 100 miles long. The inlet pressure is 1300 psia, the outlet pressure is 300 psia, and the average temperature is 40°F. The gas consists of 75% methane (CH_4), 21% ethane (C_2H_6), and 4% propane (C_3H_8).

Find: The flow rate in millions of standard cubic feet per day (MMscfd).

Solutions: Three solutions to this example are presented for the purpose of illustrating the variations in results obtained by use of the Simplified Isothermal Flow, Weymouth, and the Panhandle A equations.

Simplified Isothermal Equation (page 6-3)

1. $$q'_h = 3.2308\left(\frac{T_b}{P'_b}\right)\left[\frac{(P'_1)^2 - (P'_2)^2}{f\, L_m\, T_{avg}\, S_g}\right]^{0.5} d^{2.5}$$

2. $d = 13.376$ page B-15

   $d^{2.5} = 654.36$

3. $f = 0.0128$ turbulent flow assumed; page A-26

4. $T = 460 + t = 460 + 40 = 500$

5. Approximate atomic masses:

   Carbon $C = 12.0$
   Hydrogen $H = 1.0$

6. Approximate molecular masses:
   Methane (CH_4)
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   Propane (C_3H_8)
   $M_r = (3 \times 12.0) + (8 \times 1.0) = 44$
   Natural Gas
   $M_r = (16 \times 0.75) + (30 \times 0.21) + (44 \times 0.04)$
   $M_r = 20.06$

7. $S_g = \dfrac{M_r\,(\text{gas})}{M_r\,(\text{air})} = \dfrac{20.06}{28.966} = 0.693$ page 6-7

8. $$q'_h = 3.2308\left(\frac{520}{14.7}\right)\left[\frac{\left(1300^2 - 300^2\right)}{0.0128 \times 100 \times 500 \times 0.693}\right]^{0.5} (654.36)$$

   $q'_h = 4490000$

9. $$q'_d = \left(\frac{4490000\,\text{ft}^3}{1000000\,\text{hr}}\right)\left(\frac{24\,\text{hr}}{\text{day}}\right) = 107.8$$

10. $R_e = \dfrac{0.4821\, q'_h\, S_g}{d\, \mu}$ page 6-2

11. $\mu = 0.011$ estimated; page A-6

12. $R_e = \dfrac{0.4821 \times 4490000 \times 0.693}{13.376 \times 0.011}$

    $R_e = 10200000$ or 1.02×10^7

13. $f = 0.0128$ page A-26

14. Since the assumed friction factor (f = 0.0128) is correct, the flow rate is 107.8 MMscfd. If the assumed friction factor were incorrect, it would have to be adjusted and Steps 8, 9, 12, and 13 repeated until the assumed friction factor was in reasonable agreement with that based upon the calculated Reynolds number.

Weymouth Equation (page 6-3)

15. $$q'_h = 18.062\left(\frac{T_b}{P'_b}\right)\left[\frac{(P'_1)^2 - (P'_2)^2}{L_m\, T_{avg}\, S_g}\right]^{0.5} d^{2.667}$$

16. $d^{2.667} = 1009$

17. $$q'_h = 18.062\left(\frac{520}{14.7}\right)\left[\frac{1300^2 - 300^2}{100 \times 500 \times 0.693}\right]^{0.5} (1009)$$

    $q'_h = 4380000$

18. $$q'_d = \left(\frac{4380000\,\text{ft}^3}{1000000\,\text{hr}}\right)\left(\frac{24\,\text{hr}}{\text{day}}\right) = 105.1$$

Panhandle A Equation (page 6-3)

19. $$q'_h = 18.161\, E\left(\frac{T_b}{P'_b}\right)^{1.0788}\left[\frac{(P'_1)^2 - (P'_2)^2}{L_m\, T_{avg}\, S_g^{0.8539}}\right]^{0.5394} d^{2.6182}$$

20. Assume average operation conditions; then efficiency is 92 percent:

    $E = 0.92$

21. $d^{2.6182} = 889$

22. $$q'_h = 18.161 \times 0.92\left(\frac{520}{14.7}\right)^{1.0788}\left[\frac{\left(1300^2 - 300^2\right)}{100 \times 500 \times 0.693^{0.8539}}\right]^{0.5394} (889)$$

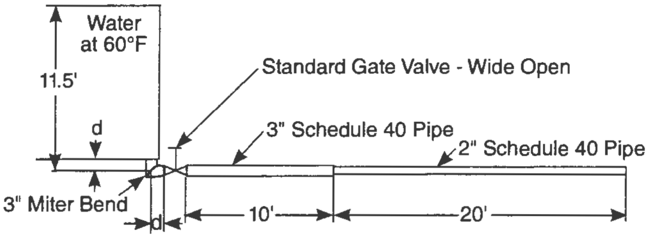
    $q'_h = 5340000$

23. $$q'_d = \left(\frac{5340000\,\text{ft}^3}{1000000\,\text{hr}}\right)\left(\frac{24\,\text{hr}}{\text{day}}\right) = 128.2$$

Discharge of Fluids from Piping Systems

Example 7-19 Water

Given: Water at 60°F is flowing from a reservoir through the piping system below. The reservoir has a constant head of 11.5'.



Find: The flow rate in gallons per minute.

Solution:

1. $Q = 19.64\, d^2 \sqrt{\frac{h_L}{K}}$ — page 6-5

   $R_e = \frac{50.66\, Q \rho}{d \mu}$ — page 6-2

   $\beta = \frac{d_1}{d_2}$ — page A-27

2. $K = 0.5$ — entrance; page A-30

   $K = 60\, f_T$ — mitre bend; page A-30

   $K_1 = 8\, f_T$ — gate valve; page A-28

   $K = f\, \frac{L}{D}$ — straight pipe; page 6-4

   $K_2 = \frac{0.5\left(1 - \beta^2\right)\sqrt{\sin\frac{\theta}{2}}}{\beta^4}$ — sudden contraction; page A-27

   $K = \frac{f\, L}{D\, \beta^4}$ — small pipe in terms of larger pipe, page 2-9

   $K = \frac{1}{\beta^4}$ — exit from small pipe in terms of larger pipe

3. $d = 2.067$ — 2" Sched. 40 pipe; page B-13

   $d = 3.068$ — 3" Sched. 40 pipe; page B-13

   $\mu = 1.1$ — page A-3

   $\rho = 62.364$ — page A-7

   $f_T = 0.019$ — 2" pipe; page A-27

   $f_T = 0.017$ — 3" pipe; page A-27

4. $\beta = \frac{2.067}{3.068} = 0.67$

   $K = 0.5$ — 3" entrance

   $K = 60 \times 0.017 = 1.02$ — 3" mitre bend

   $K_1 = 8 \times 0.017 = 0.14$ — 3" gate valve

   $K = \frac{0.017 \times 10 \times 12}{3.068} = 0.67$ — 10 feet, 3" pipe

   For 20 feet of 2" pipe, in terms of 3" pipe,

   $K = \frac{0.019 \times 20 \times 12}{2.067 \times 0.67^4} = 10.9$

   For 2" exit, in terms of 3" pipe,

   $K_1 = \frac{1}{0.67^4} = 5.0$

   For sudden contraction,

   $K_2 = \frac{0.5\left(1 - 0.67^2\right) \cdot 1}{0.67^4} = 1.37$ and,

   $K_{TOTAL} = 0.5 + 1.02 + 0.14 + 0.67 + 10.9 + 5.0 + 1.37 = 19.6$

5. $Q = 19.64 \times 3.068^2 \sqrt{\frac{11.5}{19.6}} = 142$

   (this solution assumes flow in fully turbulent zone)

6. Calculate Reynolds numbers and check friction factors for flow in straight pipe of the 2" size:

   $R_e = \frac{50.66 \times 142 \times 62.364}{2.067 \times 1.1} = 1.97 \times 10^5$

   $f = 0.021$ — page A-26

   and for flow in straight pipe of the 3" size:

   $R_e = \frac{50.66 \times 142 \times 62.364}{3.068 \times 1.1} = 1.33 \times 10^5$

   $f = 0.020$ — page A-26

7. Since assumed friction factors used for straight pipe in Step 4 are not in agreement with those based on the approximate flow rate, the K factors for these items and the total system should be corrected accordingly.

   $K = \frac{0.020 \times 10 \times 12}{3.068} = 0.78$ — 10 feet, 3" pipe

   For 20 feet of 2" pipe, in terms of 3" pipe,

   $K = \frac{0.021 \times 20 \times 12}{2.067 \times 0.67^4} = 12.1$ and,

   $K_{TOTAL} = 0.5 + 1.02 + 0.14 + 0.78 + 12.1 + 5.0 + 1.37 = 20.9$

8. $Q = 19.64 \times 3.068^2 \sqrt{\frac{11.5}{20.9}} = 137$

Discharge of Fluids from Piping Systems

Example 7-20 Steam at Sonic Velocity

Given: A header with 170 psia saturated steam is feeding a pulp stock digester through 30 feet of 2" Schedule 40 pipe which includes one standard 90 degree elbow and a fully-open conventional plug type disc globe valve. The initial pressure in the digester is atmospheric.

Find: The initial flow rate in pounds per hour, using both the modified Darcy equation and the sonic velocity and continuity equations.

Solutions - for theory, see page 1-11:

Modified Darcy Formula

1. $W = 1890\, Y\, d^2 \sqrt{\dfrac{\Delta P}{K \bar{V}_1}}$ page 6-5

   $K = f \dfrac{L}{D}$ pipe; page 6-4

2. $K_1 = 340\, f_T$ globe valve; page A-28

   $K = 30\, f_T$ 90° elbow; page A-30

   $K = 0.5$ entrance from header; page A-30

   $K = 1.0$ exit to digester; page A-30

3. $k = 1.297$ page A-10

   $d = 2.067$ $d^2 = 4.272$ 2" pipe; page B-13

   $f_T = 0.019$ page A-27

   $\bar{V} = 2.6746$ page A-14

4. $K = \dfrac{0.019 \times 30 \times 12}{2.067} = 3.31$ 30 feet, 2" pipe

   $K_1 = 340 \times 0.019 = 6.46$ 2" globe valve

   $K = 30 \times 0.019 = 0.57$ 2" 90° elbow

   and, for the entire system,

   $K = 3.31 + 6.46 + 0.57 + 0.5 + 1.0 = 11.84$

5. $\dfrac{\Delta P}{P'_1} = \dfrac{170 - 14.7}{170} = \dfrac{155.3}{170} = 0.914$

6. Using the chart on page A-23 for k = 1.3, It is found that for K = 11.84, the maximum $\Delta P/P'_1$ is 0.785 (interpolated from table on page A-23). Since $\Delta P/P'_1$ is less than indicated in Step 5, sonic velocity occurs at the end of the pipe, and ΔP in the equation of Step 1 is:

   $\Delta P = 0.785 \times 170 = 133.5$

7. $Y = 0.710$ interpolated from table; page A-23

8. $W = 1890 \times 0.71 \times 4.272 \sqrt{\dfrac{133.5}{11.84 \times 2.6746}}$

   $W = 11770$

Sonic Velocity and Continuity Equations

9. $v_s = \sqrt{k\, g\, 144 P' \bar{V}}$ page 6-4

   $W = \dfrac{v\, d^2}{0.05093\, \bar{V}}$ Equation 6-2; page 6-2

10. $P' = P'_1 - \Delta P$

    $P' = 170 - 133.5 = 36.5$

    ΔP determined in Step 6.

11. $h_g = 1196.5$ 170 psia saturated steam; page A-14

12. At 36.5 psia, the temperature of steam with total heat of 1196.5 Btu/lb equals 317.5°F, and

    $\bar{V} = 12.43$ page A-13

13. $v_s = \sqrt{1.3 \times 32.2 \times 144 \times 36.5 \times 12.43}$

    $v_s = 1654$

    $W = \dfrac{1654 \times 4.272}{0.05093 \times 12.4} = 11160$

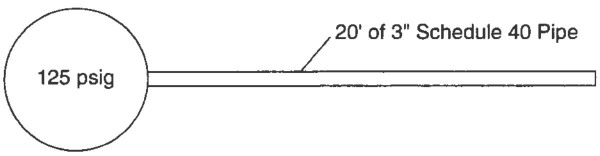
NOTE:

In Steps 11 and 12 constant total heat h_g is assumed. But the increase in specific volume from inlet to outlet requires that the velocity must increase. Source of the kinetic energy increase is the internal heat energy of the fluid. Consequently, the heat energy actually decreases toward the outlet. Calculation of the correct h_g at the outlet yields a flow rate commensurate with the answer in Step 8.

Discharge of Fluids from Piping Systems

Example 7-21 Gases at Sonic Velocity

Given: Coke oven gas having a specific gravity of 0.42, a header pressure of 125 psig, and a temperature of 140°F is flowing through 20 feet of 3" Schedule 40 pipe before discharging to atmosphere. Assume ratio of specific heats, k = 1.4.



Find: The flow rate in standard cubic feet per hour (scfh).

Solution - for theory, see page 1-9:

1. $q'_h = 40700\ Y d^2 \sqrt{\dfrac{\Delta P\ P'_1}{K T_1 S_g}}$ page 6-5

   $K = f \dfrac{L}{D}$ page 6-4

2. $P'_1 = 125 + 14.7 = 139.7$

3. $f = 0.0175$ page A-26

Note: The Reynolds number need not be calculated since gas discharged to atmosphere through a short pipe will have a high R_e, and flow will always be in a fully turbulent range, in which the friction factor is constant.

4. $d = 3.068$ $d^2 = 9.413$ page B-13

   $D = 0.2557$

5. $K = f \dfrac{L}{D} = \dfrac{0.0175 \times 20}{0.2557} = 1.369$ for pipe

   $K = 0.5$ for entrance; page A-30

   $K = 1.0$ for exit; page A-30

   $K = 1.369 + 0.5 + 1.0 = 2.87$ total

6. $\dfrac{\Delta P}{P'_1} = \dfrac{139.7 - 14.7}{139.7} = \dfrac{125.0}{139.7} = 0.895$

7. Using the chart on page A-23 for k = 1.4, it is found that for K = 2.87, the maximum $\Delta P/P'_1$ is 0.657 (interpolated from table on page A-23). Since $\Delta P/P'_1$ is less than indicated in Step 6, sonic velocity occurs at the end of the pipe and ΔP in Step 1 is:

   $\Delta P = 0.657\ \ P'_1 = 0.657 \times 139.7 = 91.8$

8. $T_1 = 140 + 460 = 600$

9. $Y = 0.637$ interpolated from table; page A-23

10. $q'_h = 40700 \times 0.637 \times 9.413 \sqrt{\dfrac{91.8 \times 139.7}{2.87 \times 600 \times 0.42}}$

    $q'_h = 1028000$

Example 7-22 Compressible Fluids at Subsonic Velocity

Given: Air at a pressure of 19.3 psig and a temperature of 100°F is measured at a point 10' from the outlet of a ½" Schedule 80 pipe discharging to atmosphere.

Find: The flow rate in standard cubic feet per hour (scfh).

Solution:

1. $q'_h = 40700\ Y d^2 \sqrt{\dfrac{\Delta P\ P'_1}{K T_1 S_g}}$ page 6-5

   $K = f \dfrac{L}{D}$ page 6-4

2. $P'_1 = 19.3 + 14.7 = 34.0$

3. $\Delta P = 19.3$

4. $d = 0.546$ $d^2 = 0.2981$ page B-13

   $D = 0.0455$

5. $f = 0.0275$ fully turbulent flow; page A-26

6. $K = f \dfrac{L}{D} = \dfrac{0.0275 \times 10}{0.0455} = 6.04$ for pipe

   $K = 1.0$ for exit; page A-30

   $K = 6.04 + 1.0 = 7.04$ total

7. $\dfrac{\Delta P}{P'_1} = \dfrac{19.3}{34.0} = 0.568$

8. $Y = 0.76$ page A-23

9. $T_1 = 460 + t_1 = 460 + 100 = 560$

10. $q'_h = 40700 \times 0.76 \times 0.2981 \sqrt{\dfrac{19.3 \times 34.0}{7.04 \times 560 \times 1.0}}$

    $q'_h = 3762$

Flow Through Orifice Meters

Example 7-23 Liquid Service

Given: A square edged orifice of 2.0" diameter is installed in a 4" Schedule 40 pipe having a mercury manometer connected between taps located 1 diameter upstream and 0.5 diameter downstream.

Find: (a) The theoretical calibration constant for the meter when used on 60°F water and for the flow range where the orifice flow coefficient C is constant and (b), the flow rate of 60°F water when the mercury deflection is 4.4".

Solution - **(a)**

1. $Q = 235.6\, d_1^2 C \sqrt{\frac{\Delta P}{\rho}}$ page 6-5

$R_e = \frac{50.66\, Q\, \rho}{d\, \mu}$ page 6-2

2. To determine differential pressure across the taps,

$\Delta P = \frac{\Delta h_m \rho}{12 \times 144}$ page 6-1

where: Δh_m = differential head in inches of mercury

3. The weight density of mercury under water equals $\rho_w(S_{Hg} - S_w)$, where (at 60°F):

ρ_w = density of water = 62.364 page A-7
S_{Hg} = specific gravity of mercury = 13.57 page A-8
S_w = specific gravity of water = 1.00 page A-7

4. And ρ of H_g under H_2O = 62.364(13.57 - 1.00) = 784 lb/ft³

5. $\Delta P = \frac{\Delta h_m (784)}{12 \times 144} = 0.454\, \Delta h_m$

6. $d_2 = 4.026$ page B-14

7. $\frac{d_1}{d_2} = \frac{2.00}{4.026} = 0.497$

8. $C = 0.625$ page A-21

9. $Q = 235.6 \times (2.0)^2 \times 0.625 \sqrt{\frac{0.454 \times \Delta h_m}{62.364}}$

$Q = 50.3 \sqrt{\Delta h_m}$ calibration constant

Solution - **(b):**

10. $Q = 50.3 \sqrt{\Delta h_m} = 50.3 \sqrt{4.4} = 106$

11. $\mu = 1.1$ page A-3

12. $R_e = \frac{50.66 \times 106 \times 62.364}{4.026 \times 1.1}$ $R_e = 75600$ or 7.56×10^4

13. C = 0.625 is correct for $R_e = 7.55 \times 10^4$, per page A-21; therefore, the flow rate through the pipe is 106 gallons per minute.

14. When the C factor on page A-21 is incorrect, for the Reynolds number based on calculated flow, it must be adjusted until reasonable agreement is reached by repeating Steps 9, 10, and 12.

Example 7-24 Laminar Flow

In flow problems where the viscosity is high, calculate the Reynolds number to determine the type of flow.

Given: S.A.E. 10W Oil is flowing through a 3" Schedule 40 pipe and produces 0.4 psi pressure differential between the pipe taps of a 2.15" I.D. square edged orifice.

Find: The flow rate in gallons per minute.

Solution:

1. $Q = 235.6\, d_1^2 C \sqrt{\frac{\Delta P}{\rho}}$ page 6-5

$R_e = \frac{50.66\, Q\, \rho}{d\, \mu}$ page 6-2

2. $\mu = 40$ suspect laminar flow; page A-3

3. $d_2 = 3.068$ page B-13

4. $\frac{d_1}{d_2} = \frac{2.15}{3.068} = 0.70$

5. $C = 0.75$ page A-21; assumed value based on laminar flow

6. $S = 0.874$ at 60°F page A-8

$S = 0.86$ at 90°F page A-8

7. $\rho = 62.36 \times 0.86 = 53.6$ page A-8

8. $Q = 235.6 \times (2.15)^2 \times 0.75 \sqrt{\frac{0.4}{53.6}} = 70.6$

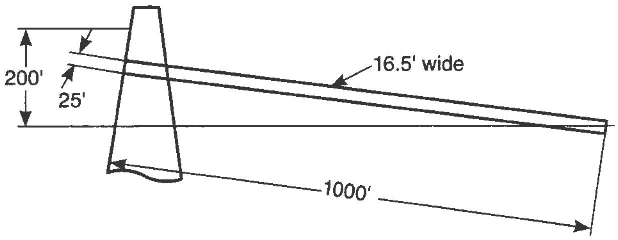
9. $R_e = \frac{50.66 \times 70.6 \times 53.6}{3.068 \times 40} = 1562$

Since the Reynolds number in the pipe with the assumed flow rate is less than 5,000 (the lower Reynolds number limit of the ASME formula) the calculated value of the flow rate through the restriction orifice can not be determined and a meter calibration must be performed.

Application of Hydraulic Radius to Flow Problems

Example 7-25 Rectangular Duct

Given: A rectangular concrete overflow aqueduct, 25 feet high and 16.5 feet wide, has an absolute roughness (ε) of 0.01 foot.



Find: The discharge rate in cubic feet per second when the liquid in the reservoir has reached the maximum height indicated in the above sketch. Assume the average temperature of the water is 60°F.

Solution:

1. $h_L = \frac{v^2}{2g}\left(K_e + K_a\right) = \frac{v^2}{2g}\left(K_e + \frac{fL}{4R_H}\right)$

2. $v = \frac{q}{A}$

3. $q = 0.04375\, d^2 \sqrt{\frac{h_L}{K_e + K_a}}$ page 6-5

   $q = 8.02\, A \sqrt{\frac{h_L}{K_e + K_a}}$

   $q = 8.02\, A \sqrt{\frac{h_L}{K_e + \frac{fL}{4R_H}}}$

   where; K_e = resistance of entrance and exit
   K_a = resistance of aqueduct

To determine the friction factor from the Moody diagram, an equivalent diameter four times the hydraulic radius is used; refer to page 6-7.

$R_H = \frac{\text{cross sectional flow area}}{\text{wetted perimeter}}$

$d = 48\, R_H$ page 6-7

$R_e = \frac{22740\, q\, \rho}{d\, \mu} = \frac{473\, q\, \rho}{R_H\, \mu}$ page 6-2

4. Assuming a sharp edged entrance,
   $K = 0.5$ page A-30

   Assuming a sharp edged exit to atmosphere,
   $K = 1.0$ page A-30

   Then, resistance of entrance and exit,
   $K_e = 0.5 + 1.0 = 1.5$

5. $R_H = \frac{16.5 \times 25}{2\,(16.5 + 25)} = 4.97\text{ft}$

6. Equivalent diameter relationship:

   $D = 4R_H = 4 \times 4.97 = 19.88$ page 6-7

   $d = 48R_H = 48 \times 4.97 = 239$ page 6-7

7. Relative roughness, $\frac{\varepsilon}{D} = 0.0005$ page A-24

8. $f = 0.017$ fully turbulent flow assumed; page A-24

9. $q = 8.05 \times 25 \times 16.5 \sqrt{\frac{200}{1.5 + \frac{0.017 \times 1000}{19.88}}}$

   $q = 30600$

10. Calculate R_e and check, f = 0.017 for q = 30,600 cfs flow.

11. $\rho = 62.364$ page A-7

12. $\mu = 1.1$ page A-3

13. $R_e = \frac{473 \times 30600 \times 62.364}{4.97 \times 1.1}$

    $R_e = 165000000 \text{ or } 1.65 \times 10^8$

14. $f = 0.017$ for calculated R_e; page A-25

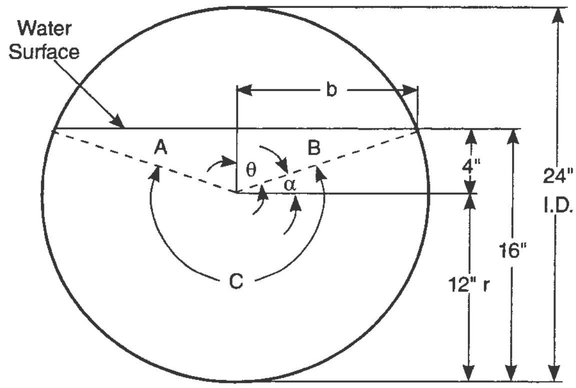
15. Since the friction factor assumed in Step 8 and that determined in Step 14 are in agreement, the discharge flow will be 30,600 cfs.

16. If the assumed friction factor and the friction factor based on the calculated Reynolds number were not in reasonable agreement, the former should be adjusted and calculations repeated until reasonable agreement is reached.

Application of Hydraulic Radius to Flow Problems

Example 7-26 Pipe Partially Filled with Flowing Water

Given: A cast iron pipe is two-thirds full of steady, uniform flowing water (60°F). The pipe has an inside diameter of 24" and a slope of ¾" per foot. Note the sketch that follows.



Find: The flow rate in gallons per minute.

Solution:

1. $Q = 19.64\, d^2 \sqrt{\dfrac{h_L D}{f L}}$ page 6-5

Since pipe is flowing partially full an equivalent hydraulic diameter based upon hydraulic radius is substituted for D in Equation 1 (see page 1-4).

$D_H = 4R_H$ or $d_H = 48R_H$ page 6-7

2. $Q = 19.64\, d^2 \sqrt{\dfrac{h_L 4R_H}{f L}} = 39.28\, d^2 \sqrt{\dfrac{h_L R_H}{f L}}$ page 6-5

3. $R_H = \dfrac{\text{cross sectional flow area}}{\text{wetted perimeter}}$ page 6-7

4. $R_e = \dfrac{50.66\, Q \rho}{d \mu} = 1.055 \dfrac{Q \rho}{R_H \mu}$ page 6-2

5. Depth of flowing water equals:

$\dfrac{2}{3}(24) = 16\text{in}$

6. $\text{Cos}\theta = \dfrac{4}{r} = \dfrac{4}{12} = 0.333$

$\theta = 70^\circ\ 32'$

$\alpha = 90^\circ - 70^\circ\ 32' = 19^\circ\ 28' = 19.47^\circ$

7. $\text{Area C} = \dfrac{\pi d^2}{4}\left[\dfrac{180 + (2 \times 19.47)}{360}\right]$

$\text{Area C} = \dfrac{\pi 24^2}{4}\left(\dfrac{218.94}{360}\right) = 275\text{in}^2$

8. $b = \sqrt{r^2 - 4^2} = \sqrt{12^2 - 16} = 11.31\text{in}$

9. $\text{Area A} = \text{Area B} = \dfrac{1}{2}(4\,b) = \dfrac{1}{2}(4 \times 11.31)$

$\text{Area A or B} = 22.6\text{in}^2$

10. The cross sectional flow area equals:

$A + B + C = 22.6 + 22.6 + 275 = 320.2\text{in}^2$

$A + B + C = \dfrac{320.2}{144} = 2.22\text{ft}^2$

11. $d^2 = \dfrac{4\,a}{\pi} = \dfrac{4 \times 320.2}{\pi} = 408$

12. $\dfrac{h_L}{L} = \dfrac{0.75}{12} = 0.0625\text{ft per ft}$

13. The wetted perimeter equals:

$\pi\, d\left(\dfrac{218.94}{360}\right)$

$\pi\, 24\left(\dfrac{218.94}{360}\right) = 45.9\text{in}$

$\dfrac{45.9}{12} = 3.83\text{ft}$

14. $R_H = \dfrac{2.22}{3.83} = 0.580$

15. Equivalent hydraulic diameter $d_H = 48\, R_H$ page 6-7

$d_H = 48\,(0.580) = 27.8$

16. Relative roughness $\dfrac{\varepsilon}{D} = 0.00036$ page A-24

17. $f = 0.0155$ assuming fully turbulent flow; page A-24

18. $Q = 39.28 \times 408 \sqrt{\dfrac{0.0625 \times 0.580}{0.0155}}$

$Q = 24500\text{gpm}$

19. Calculate the Reynolds number to check the friction factor assumed in Step 17.

20. $\rho = 62.364$ page A-7

21. $\mu = 1.1$ page A-3

22. $R_e = \dfrac{1.055 \times 24500 \times 62.364}{0.580 \times 1.1}$

$R_e = 2530000 \text{ or } 2.53 \times 10^6$

23. $f = 0.0155$ page A-25

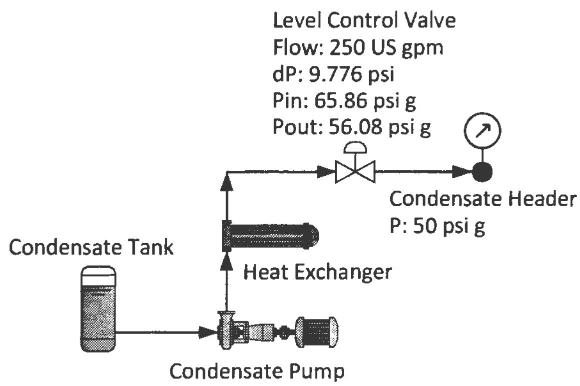
24. Since the friction factor assumed in Step 17 and that determined in Step 23 are in agreement, the flow rate will be 24,500 gpm.

25. If the assumed friction factor and the friction factor based on the calculated Reynolds number were not in reasonable agreement, the former should be adjusted and calculations repeated until reasonable agreement is reached.

Control Valve Calculations

Example 7-27 Sizing Control Valves for Liquid Service
Given: 250 gpm of condensate from a pressurized condensate tank is cooled from 225°F to 160°F in a heat exchanger then pumped to a 50 psig pressurized header. Inlet and outlet pipe of the control valve is 4" schedule 40 with no valves or fittings within 5 feet of the valve. The system is located in a facility at sea level. A single port globe style valve is desired and the following table lists the valve sizes available and their corresponding C_V at 100% open.

Valve Size	2"	2 ½"	3"	4"
100 % C_V	41	73	114	175



Find: An appropriate size valve for the level control valve.
Solution:

1. Fluid properties for water at 160°F
   Density: 60.998 lb/ft³ (S=0.978)
   Viscosity: 0.39 cP
   Vapor pressure: 4.75 psia
   Critical pressure: 3,198 psia

2. System properties
   Flow rate: 250 gpm
   Valve inlet pressure = 65.9 psig = 80.6 psia
   Valve outlet pressure = 56.1 psig = 70.8 psia

3. Initial C_V calculation assuming F_p=1.0: page 6-6

$$C_V = \frac{Q}{F_p \sqrt{\frac{P'_1 - P'_2}{S}}} = \frac{250\text{gpm}}{1.0\sqrt{\frac{80.6 - 70.8}{0.978}}} = 78.98$$

4. Initial valve selection: Based on the above table, a 2 ½" valve would be too small, a 3" valve should have the available capacity, and the 4" valve may be over-sized for the application.

5. Calculate piping geometry factor for a fully open 3" valve with 4" x 3" inlet and outlet reducers: page 6-6

$$\Sigma K = 1.5\left[1 - \left(\frac{d_{nom}}{d_1}\right)^2\right]^2 = 1.5\left[1 - \left(\frac{3}{4.026}\right)^2\right]^2 = 0.297$$

$$F_p = \frac{1}{\sqrt{1 + \frac{\Sigma K}{890}\left(\frac{C_V}{d_{nom}^2}\right)^2}} = \frac{1}{\sqrt{1 + \frac{0.297}{890}\left(\frac{114}{3^2}\right)^2}} = 0.974$$

When fully open, the valve with attached fittings will have an effective C_V of (114)(0.974)=111.

6. Recalculate F_p at the valve position close to the required C_V.

$$F_p = \frac{1}{\sqrt{1 + \frac{0.297}{890}\left(\frac{78.98}{3^2}\right)^2}} = 0.987$$

7. Recalculate the required C_V to confirm that the selected valve is adequately sized:

$$C_V = \frac{Q}{F_p \sqrt{\frac{P'_1 - P'_2}{S}}} = \frac{250\text{gpm}}{0.987\sqrt{\frac{80.6 - 70.8}{0.978}}} = 80.02$$

8. Since the required C_V is less than the valve's C_V at 100% open with attached fittings, the 3" control valve will have adequate capacity for the application and will be throttled to a C_V = 80.02 to control the flow at the desired rate.

Control Valve Calculations

Example 7-28 Checking for Choked Flow Conditions

Given: The manufacturer of the control valve in the previous example lists F_L=0.9.

Find: Confirm that choked flow conditions will not occur at the designed flow rate and position of the control valve.

Solution:

1. Calculate the liquid critical pressure ratio factor F_F:

page 6-6

$$F_F = 0.96 - 0.28\sqrt{\frac{P'_v}{P'_c}}$$

$$F_F = 0.96 - 0.28\sqrt{\frac{4.75}{3198}} = 0.9492$$

2. Calculate inlet reducer resistance coefficient and inlet Bernoulli coefficient:

page 6-6

$$\Sigma K_i = K^{inlet}_{reducer} + K_{B1}$$

$$\Sigma K_i = 0.5\left[1 - \left(\frac{d_{nom}}{d_1}\right)^2\right]^2 + \left[1 - \left(\frac{d_{nom}}{d_1}\right)^4\right]$$

$$\Sigma K_i = 0.5\left[1 - \left(\frac{3}{4.026}\right)^2\right]^2 + \left[1 - \left(\frac{3}{4.026}\right)^4\right]$$

$$\Sigma K_i = 0.0989 + 0.6917 = 0.7906$$

3. Calculate F_{LP}:

page 6-6

$$F_{LP} = \frac{F_L}{\sqrt{1 + F_L^2\left(\frac{\Sigma K_i}{890}\right)\left(\frac{C_v}{d_{nom}^2}\right)^2}}$$

$$F_{LP} = \frac{0.9}{\sqrt{1 + 0.9^2\left(\frac{0.7906}{890}\right)\left(\frac{89.6}{3^2}\right)^2}} = 0.8695$$

4. Calculate Q_{max}:

page 6-6

$$Q_{max} = \frac{F_{LP}}{F_p} C_v \sqrt{\frac{P'_1 - F_F P'_v}{S}}$$

$$Q_{max} = \frac{(0.8695)}{(0.987)}(89.6)\sqrt{\frac{80.6 - (0.9492)\cdot(4.75)}{0.978}}$$

$$Q_{max} = 696.2\text{gpm}$$

Since the desired flow rate of 250 gpm is less than the maximum flow rate at the valve position, choked flow will not occur.

5. Alternatively, check for choked flow using ΔP_{max}:

page 6-6

$$\Delta P_{max} = \left(\frac{F_{LP}}{F_p}\right)^2 (P'_1 - F_F P'_v)$$

$$\Delta P_{max} = \left(\frac{0.8965}{0.987}\right)^2 [80.6 - (0.9492)(4.75)]$$

$$\Delta P_{max} = 59.1\text{psi}$$

6. Since the actual differential pressure across the valve (9.8 psi) is less than the maximum differential pressure for choked flow conditions, the control valve will not be choked.

Flow Meter Calculations

Example 7-29 Orifice Flow Rate Calculation

Given: A differential pressure of 2.5 psi is measured across taps located 1 diameter upstream and 0.5 diameter downstream from the inlet face of a 2.000" ID orifice plate assembled in a 3" Schedule 80 steel pipe carrying water at 60°F.

Find: The flow rate in gallons per minute.

Solution:

1. $Q = 235.6 d_1^2 C \sqrt{\frac{\Delta P}{\rho}}$ Equation 6-31

2. $d_2 = 2.900$ from B-13

3. $\beta = \frac{d_1}{d_2} = \frac{2.000}{2.900} = 0.690$ Equation 6-29

4. $\rho = 62.364$ from A-7

5. $\mu = 1.1$ from A-3

6. A value of C can be calculated using Equation 6-24 and the velocity of approach or using the graph on page A-21. For this example, A-21 will be used. If we assume a R_e of 100,000 for the first iteration, the graph yields C = 0.70.

7. $Q = 235.6 \times 2.00^2 \times 0.70 \sqrt{\frac{2.5}{62.364}} = 132\text{gpm}$

8. Calculate Reynolds number with 132 gpm:

$$R_e = 50.66 \frac{132 \times 62.364}{2.900 \times 1.1} = 131000$$

9. Based on A-21, C = 0.695 for the second iteration.

$$Q = 235.6 \times 2.00^2 \times 0.695 \sqrt{\frac{2.5}{62.371}} = 131\text{gpm}$$

Example 7-30 Nozzle Sizing Calculations

Given: A design flow rate of 225 gpm of 60°F water through 6" Schedule 40 pipe is to be measured using a long radius nozzle. A head loss of 4' is desired when measured across taps 1D upstream and ½ D downstream at the design flow rate.

Find: The diameter of the nozzle.

Solution:

1. $Q = 19.64 d_1^2 C \sqrt{h_L}$ Equation 6-31

$C = \frac{C_d}{\sqrt{1 - \beta^4}}$ Equation 6-30

$C_d = 0.9965 - 0.00653\beta^{0.5}\left(\frac{10^6}{R_e}\right)^{0.5}$ Equation 4-9

2. $R_e = 50.66 \frac{Q\rho}{d\mu}$

3. $d_2 = 6.065$ from B-14

$\rho = 62.364$ from A-7

$\mu = 1.1$ from A-3

4. For the first iteration, assume $\beta = 0.5 \rightarrow 0.5 \times 6.065 = d_1 = 3.033$

5. $R_e = 50.66 \frac{225 \times 62.364}{6.065 \times 1.1} = 107000$

6. $C_d = 0.9965 - 0.00653\,(0.5)^{0.5}\left(\frac{10^6}{107000}\right)^{0.5} = 0.982$

7. $C = \frac{0.982}{\sqrt{1 - 0.5^4}} = 1.01$

8. $Q = 19.64\,(3.033)^2\,(1.01)\sqrt{4} = 366\text{gpm}$

9. This is higher than the desired flow of 225 gpm. Another iteration of steps 4-8 should be performed with a lower value for β.

10. Assume $\beta = 0.4 \rightarrow 0.4 \times 6.065 = d_1 = 2.426$

11. $R_e = 50.66 \frac{225 \times 62.364}{6.065 \times 1.1} = 107000$

12. $C_d = 0.9965 - 0.00653\,(0.4)^{0.5}\left(\frac{10^6}{107000}\right)^{0.5} = 0.9839$

13. $C = \frac{0.9839}{\sqrt{1 - 0.4^4}} = 0.9967$

14. $Q = 19.64\,(2.426)^2\,(0.9967)\sqrt{4} = 230.4\text{gpm}$

Additional iterations can be done to obtain an orifice size of 2.40" with a of 0.395 and Cd = 0.982, which yields a flow of 224.6 gpm.

Flow Meter Calculations

Example 7-31 NPRD Calculations

Given: The non recoverable pressure drop (NRPD) is the permanent pressure drop associated with the energy lost through a device.

Find: The NRPD of the long radius nozzle in Example 2.

Solution:

1. $$NRPD = \Delta P\left[\frac{\sqrt{1-\beta^4\left(1-C_d^2\right)}-c_d\,\beta^2}{\sqrt{1-\beta^4\left(1-C_d^2\right)}+c_d\,\beta^2}\right]$$ Equation 4-1

2. $$\Delta P = \frac{\rho h_L}{144}$$

3. $h_L = 4\text{ft}$

   $\rho = 62.364$

   $\beta = 0.395$

   $C_d = 0.982$

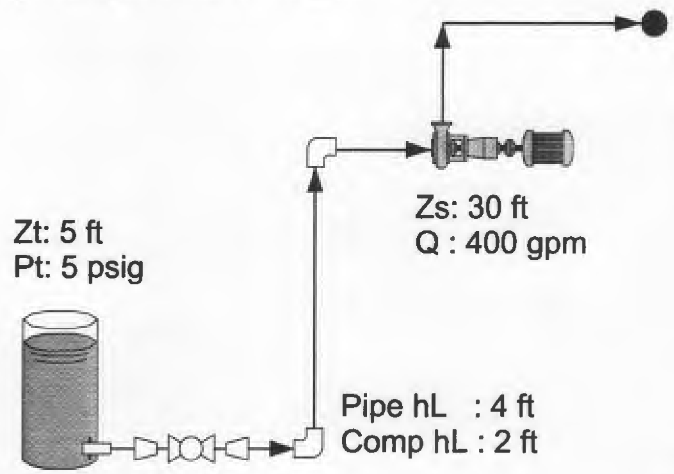
4. $$\Delta P = \frac{62.364 \times 4}{144} = 1.732\text{psi}$$

5. $$NRPD = 1.732\left[\frac{\sqrt{1-0.395^4\left(1-0.982^2\right)}-0.982 \times 0.395^2}{\sqrt{1-0.395^4\left(1-0.982^2\right)}+0.982 \times 0.395^2}\right] = 1.272\text{psi}$$

Pump Examples

Example 7-32 NPSH Available Calculation

Given: A pump, located 30' above sea level, has a NPSHr of 20' at 400 gpm and is fed by a tank of 60°F water with a surface pressure of 5 psig and a liquid level 25' below the suction of the pump. The head losses in the suction pipeline and installed components were found to be 2' and 4' respectively at a flow of 400 gpm.



Find: The NPSH available and compare it to the NPSHr to ensure that sufficient head is provided to the suction side of the pump. An NPSHr margin ratio of 1.3 is desired.

Solution:

1. Use the NPSHa equation (See page 5-2).

$$\text{NPSHa} = \frac{144}{\rho}\left(P'_t - P_{vp}\right) + \left(Z_t - Z_s\right) - h_L \qquad \text{Equation 5-1}$$

2. $\rho = 62.364 \frac{lb}{ft^3}$ from A-7

$$P'_t = P_t + P_{atm} = 5psi + 14.7psi = 19.7psi$$

$$P'_v = 0.25611\,psi$$

$$\left(Z_t - Z_s\right) = -25ft$$

$$h_L = 4ft + 2ft = 6ft$$

3. $\text{NPSHa} = \frac{144}{62.364}(19.7 - 0.25611) + (-25) - 6$

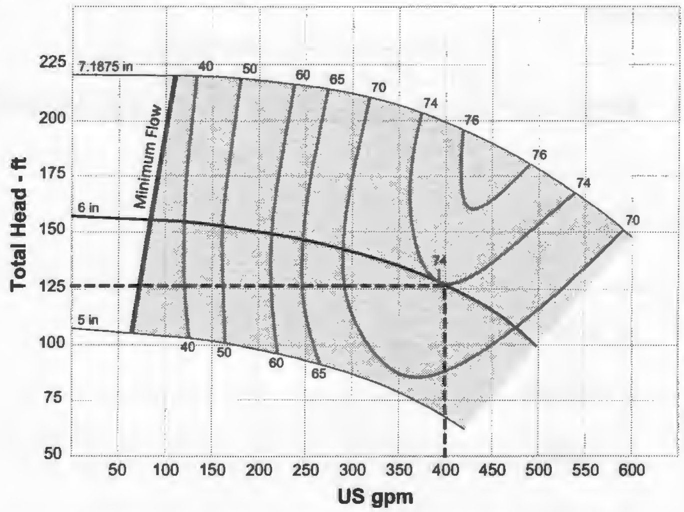
$$\text{NPSHa} = 13.9ft$$

4. NPSHr (adjusted for margin) $= 1.3 \times 20 = 26ft$

The NPSH available is below the NPSH required for the pump and cavitation is the likely result. With the safety margin, 12.1 feet of head is necessary to match the NPSHr. Raising the tank or the liquid level in the tank by 12.1 ft or increasing the pressure by 6 psi would be sufficient to meet the NPSH requirement for the pump.

Example 7-33 Pump Affinity rules

Given: For the 6" impeller trim on the curve below the pump produces 126' of head and 400 gpm while running at 3500 rpm. At this speed the brake horsepower is 17.5 hp.



Find: The flow rate, head and power of this operating point on the pump curve if the speed is changed to 1700 rpm.

Solution:

1. The pump affinity rules for speed (see page 5-):

Flow rate: $$\frac{Q_1}{Q_2} = \frac{N_1}{N_2} \qquad \text{Equation 5-2}$$

Head: $$\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2 \qquad \text{Equation 5-3}$$

Power: $$\frac{P_1}{P_2} = \left(\frac{N_1}{N_2}\right)^3 \qquad \text{Equation 5-4}$$

2. $\frac{400gpm}{Q_2} = \frac{3500rpm}{1700rpm}$

$$Q_2 = \frac{400rpm \times 1700rpm}{3500rpm} = 194.3gpm$$

3. $\frac{126ft}{H_2} = \left(\frac{3500rpm}{1700rpm}\right)^2$

$$H_2 = 126\,ft\left(\frac{1700rpm}{3500rpm}\right)^2 = 29.7ft$$

4. $\frac{17.5hp}{P_2} = \left(\frac{3500rpm}{1700rpm}\right)^3$

$$P_2 = 17.5hp\left(\frac{1700rpm}{3500rpm}\right)^3 = 2.01hp$$

Pump Examples

Example 7-34 Pump Power and Operating Cost

Given: A pump provides 428 ft of head pumping 700 gpm of 60°F water. The efficiencies of the pump, motor and VFD are 70.7%, 95% and 96% respectively.

Find: The brake horsepower, electrical horsepower and operating cost for 8000 hours of operation with an average power cost of $0.12/kWh.

Solution:

1. Pump power calculations (see page 5-4):

$$bhp = \frac{QH\rho}{247000\eta_p} \qquad \text{Equation 5-8}$$

$$ehp = \frac{bhp}{\eta_m \eta_{VFD}} \qquad \text{Equation 5-9}$$

$$OC = \frac{0.7457QH\rho}{247000\eta_p \eta_m \eta_{VFD}}(\text{hours})\left(\frac{\$}{kWh}\right) \qquad \text{Equation 5-10}$$

2. $\rho = 62.364\frac{lb}{ft^3}$ from A-7

3. $$bhp = \frac{700gpm \times 428ft \times 62.364\frac{lb}{ft^3}}{247000 \times 0.707} = 107hp$$

4. $$ehp = \frac{107hp}{0.95 \times 0.96} = 117.3hp$$

5. $$OC = 0.7457 \times 117.3\ hp \times (8000hrs) \times \frac{\$0.12}{kWh} = \$83{,}970$$

Tees and Wyes Calculations

Example 7-35 Hydraulic Resistance of a Converging Tee

Given: A 4" schedule 40 tee with equal leg diameters has 300 gpm of 60°F water flowing into the straight leg and 100 gpm converging in from the 90° branch leg.

Find: The resistance coefficients for the straight leg and the branch leg along with the head loss across each flow path.

Solution:

1. Find branch diameter ratio, branch flow ratio, and fluid velocity in the combined leg:

$$\beta_{branch}^{\ 2} = \left(\frac{d_{branch}}{d_{comb}}\right)^2 = \left(\frac{4.026}{4.026}\right)^2 = 1.0$$ page 2-14 and B-14

$$\frac{Q_{branch}}{Q_{comb}} = \frac{100\text{gpm}}{400\text{gpm}} = 0.25$$

fluid velocity in the combined leg =

$$v = 0.4085\frac{Q}{d^2} = 0.4085\left(\frac{400\text{gpm}}{4.026^2}\right) = 10.08\frac{\text{ft}}{\text{sec}}$$ page 6-2

2. Find K_{branch} using Equation 2-35: page 2-15

$$K_{branch} = C\left[1 + D\left(\frac{Q_{branch}}{Q_{comb}}\frac{1}{\beta_{branch}^{\ 2}}\right)^2 - E\left(1 - \frac{Q_{branch}}{Q_{comb}}\right)^2 - F\frac{1}{\beta_{branch}^{\ 2}}\left(\frac{Q_{branch}}{Q_{comb}}\right)^2\right]$$

$C = 0.9(1 - 0.25) = 0.675$ Table 2-2 on page 2-15

$D = 1$ Table 2-1 on page 2-15

$E = 2$ Table 2-1 on page 2-15

$F = 0$ Table 2-1 on page 2-15

$$K_{branch} = 0.675\left[1 + 1\left(0.25 \times \frac{1}{1}\right)^2 - 2(1 - 0.25)^2 - 0 \times \frac{1}{1} \times (0.25)^2\right] = -0.0422$$

3. Find head loss across the branch to the combined leg:

$$h_L = K\frac{v^2}{2g} = (-0.0442)\frac{10.08^2}{2\left(32.2\frac{\text{ft}}{\text{sec}^2}\right)} = -0.0666\text{ ft}$$ page 6-4

Note: there is a fluid energy gain.

4. From Table 2-1, find K_{run} using Equation 2-36:

$$K_{run} \cong 1.55\left(\frac{Q_{branch}}{Q_{comb}}\right) - \left(\frac{Q_{branch}}{Q_{comb}}\right)^2 \cong 1.55(0.25) - (0.25)^2 \cong 0.325$$ page 2-15

5. Find head loss across the run to the combined leg:

$$h_L = K\frac{v^2}{2g} = (0.325)\frac{10.08^2}{2\left(32.2\frac{\text{ft}}{\text{sec}^2}\right)} = 0.513\text{ft}$$ page 6-4

Tees and Wyes Calculations

Example 7-36 Hydraulic Resistance of a Diverging Wye

Given: 60°F water diverges in a 6"schedule 80 45° wye with equal leg diameters, 250 gpm flows through the branch leg and 400 gpm flows through the straight leg.

Find: The resistance coefficients for the straight leg and the branch leg along with the head loss across each flow path.

Solution:

1. Find branch diameter ratio, branch flow ratio, and fluid velocity in the combined leg:

$$\beta_{branch}^{2} = \left(\frac{d_{branch}}{d_{comb}}\right)^2 = \left(\frac{5.761}{5.761}\right)^2 = 1.0$$

page 2-14 and B-14

$$\frac{Q_{branch}}{Q_{comb}} = \frac{250\text{gpm}}{650\text{gpm}} = 0.385$$

$$v = 0.4085\frac{Q}{d^2} = 0.4085\left(\frac{650\text{gpm}}{5.761^2}\right) = 8.0\frac{\text{ft}}{\text{sec}}$$

page 6-2

2. Find K_{branch} using Equation 2-37: page 2-15

$$K_{branch} = G\left[1 + H\left(\frac{Q_{branch}}{Q_{comb}}\frac{1}{\beta_{branch}^{2}}\right)^2 - J\left(\frac{Q_{branch}}{Q_{comb}}\frac{1}{\beta_{branch}^{2}}\right)\cos\alpha\right]$$

$G = 1.0 - 0.6(0.385) = 0.769$ Table 2-4 on Page 2-15

$H = 1$ Table 2-3 on Page 2-15

$J = 2$ Table 2-3 on Page 2-15

$$K_{branch} = 0.769\left[1 + 1\left(0.385\frac{1}{1}\right)^2 - 2\left(0.385\frac{1}{1}\right)\cos(45)\right] = 0.464$$

3. Find head loss across the branch to the combined leg:

$$h_L = K\frac{v^2}{2g} = (0.464)\frac{8.0^2}{2\left(32.2\frac{\text{ft}}{\text{sec}^2}\right)} = 0.461\text{ft}$$

page 6-4

4. Find K_{run} using Equation 2-38: page 2-15

$$K_{run} = M\left(\frac{Q_{branch}}{Q_{comb}}\right)^2$$

$M = 2[2(0.385) - 1] = -0.460$ Table 2-5 on page 2-15

$$K_{run} = -0.46(0.385)^2 = -0.0682$$

5. Find head loss across the run to the combined leg:

$$h_L = K\frac{v^2}{2g} = (-0.0682)\frac{8.0^2}{2\left(32.2\frac{\text{ft}}{\text{sec}^2}\right)} = -0.0678\text{ft}$$

page 6-4

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Appendix A

Physical Properties of Fluids and Flow Characteristics of Valves, Fittings & Pipe

The physical properties of many commonly used fluids are required for the solution of flow problems. These properties, compiled from many varied reference sources, are presented in this appendix. The convenience of a condensed presentation of these data will be readily apparent.

Most texts on the subject of fluid mechanics cover in detail the flow through pipe, but the flow characteristics of valves and fittings are given little, if any, attention, probably because the information has not been available. A means of estimating the resistance coefficients for valves, deviating in minor detail from the standard forms for which the coefficients are known, is presented in Chapter 2.

The Y net expansion factors for discharge of compressible fluids from piping systems, which are presented here for the first time, provide means for a greatly simplified solution of a heretofore complex problem.

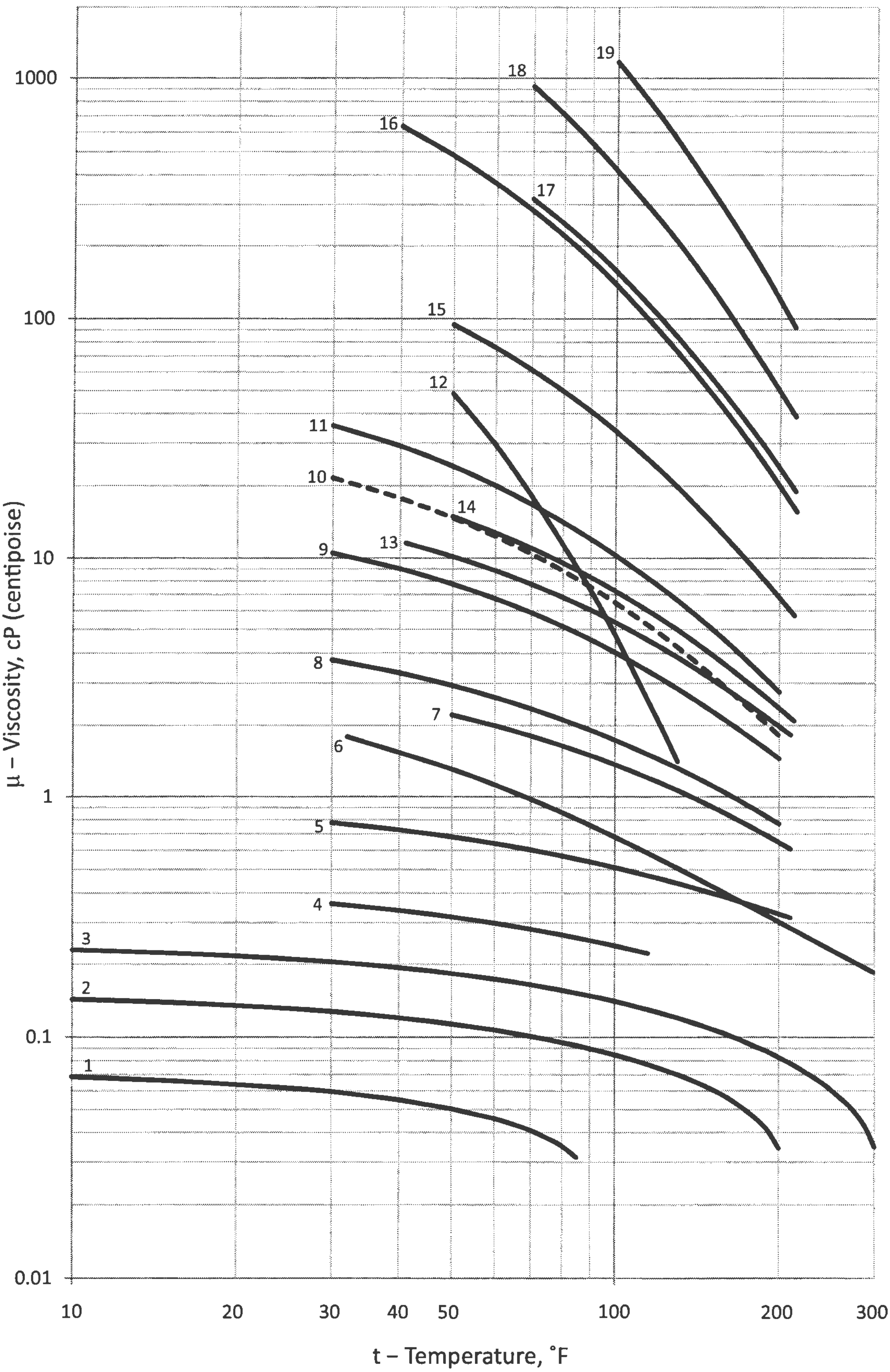
Viscosity of Steam and Water[33]

Temp. °F	1 psia	2 psia	5 psia	10 psia	20 psia	50 psia	100 psia	200 psia	500 psia	1000 psia	2000 psia	5000 psia	7500 psia	10000 psia	12000 psia
	Dynamic Viscosity of Steam and Water - in centipoise (cP), μ														
Sat. water	0.668	0.527	0.391	0.316	0.257	0.199	0.165	0.138	0.110	0.092	0.072	...	...	...	...
Sat. steam	0.010	0.011	0.011	0.012	0.013	0.014	0.014	0.016	0.017	0.019	0.022	...	...	...	...
1500	0.041	0.041	0.041	0.041	0.041	0.041	0.041	0.041	0.041	0.041	0.042	0.043	0.045	0.048	0.050
1450	0.040	0.040	0.040	0.040	0.040	0.040	0.040	0.040	0.040	0.040	0.041	0.042	0.045	0.047	0.049
1400	0.039	0.039	0.039	0.039	0.039	0.039	0.039	0.039	0.039	0.039	0.040	0.041	0.044	0.046	0.049
1350	0.038	0.038	0.038	0.038	0.038	0.038	0.038	0.038	0.038	0.038	0.039	0.041	0.043	0.046	0.048
1300	0.037	0.037	0.037	0.037	0.037	0.037	0.037	0.037	0.037	0.037	0.037	0.040	0.042	0.045	0.048
1250	0.035	0.036	0.036	0.036	0.036	0.036	0.036	0.036	0.036	0.036	0.036	0.039	0.041	0.045	0.048
1200	0.035	0.035	0.035	0.035	0.035	0.035	0.035	0.035	0.035	0.035	0.035	0.038	0.041	0.044	0.048
1150	0.033	0.033	0.033	0.033	0.033	0.033	0.033	0.033	0.034	0.034	0.034	0.037	0.040	0.044	0.048
1100	0.032	0.032	0.032	0.032	0.032	0.032	0.032	0.032	0.032	0.033	0.033	0.036	0.039	0.045	0.049
1050	0.031	0.031	0.031	0.031	0.031	0.031	0.031	0.031	0.031	0.031	0.032	0.035	0.039	0.046	0.051
1000	0.030	0.030	0.030	0.030	0.030	0.030	0.030	0.030	0.030	0.030	0.031	0.034	0.039	0.048	0.054
950	0.029	0.029	0.029	0.029	0.029	0.029	0.029	0.029	0.029	0.029	0.030	0.033	0.041	0.051	0.058
900	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.033	0.044	0.056	0.063
850	0.027	0.027	0.027	0.027	0.027	0.027	0.027	0.027	0.027	0.027	0.027	0.033	0.051	0.062	0.068
800	0.026	0.026	0.026	0.026	0.026	0.026	0.026	0.026	0.026	0.026	0.026	0.038	0.060	0.069	0.074
750	0.024	0.024	0.024	0.024	0.024	0.024	0.024	0.024	0.024	0.024	0.025	0.056	0.069	0.076	0.080
700	0.023	0.023	0.023	0.023	0.023	0.023	0.023	0.023	0.023	0.023	0.023•	0.070	0.078	0.083	0.087
650	0.022	0.022	0.022	0.022	0.022	0.022	0.022	0.022	0.022	0.022	0.022	0.080	0.086	0.090	0.094
600	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.020	0.082	0.089	0.094	0.098	0.101
550	0.020	0.020	0.020	0.020	0.020	0.020	0.020	0.020	0.019	0.019	0.093	0.099	0.103	0.107	0.110
500	0.019	0.019	0.019	0.019	0.019	0.019	0.019	0.018	0.018	0.102	0.104	0.109	0.113	0.117	0.120
450	0.017	0.017	0.017	0.017	0.017	0.017	0.017	0.017	0.115	0.116	0.118	0.122	0.126	0.130	0.133
400	0.016	0.016	0.016	0.016	0.016	0.016	0.016	0.016	0.132	0.132	0.134	0.139	0.143	0.146	0.149
350	0.015	0.015	0.015	0.015	0.015	0.015	0.015	0.153	0.154	0.154	0.156	0.161	0.165	0.169	0.172
300	0.014	0.014	0.014	0.014	0.014	0.014	0.184	0.184	0.185	0.185	0.187	0.192	0.196	0.200	0.203
250	0.013	0.013	0.013	0.013	0.013	0.230	0.230	0.230	0.230	0.231	0.233	0.238	0.243	0.247	0.251
200	0.012	0.012	0.012	0.012	0.303	0.303	0.303	0.303	0.303	0.304	0.306	0.312	0.316	0.321	0.325
150	0.011	0.011	0.429	0.429	0.430	0.430	0.430	0.430	0.430	0.431	0.433	0.438	0.443	0.447	0.451
100	0.681	0.681	0.681	0.681	0.681	0.681	0.681	0.681	0.681	0.681	0.682	0.684	0.687	0.690	0.693
50	1.305	1.305	1.305	1.305	1.305	1.305	1.304	1.303	1.301	1.298	1.291	1.275	1.265	1.258	1.254
32	1.790	1.790	1.790	1.790	1.790	1.789	1.789	1.787	1.782	1.774	1.758	1.719	1.693	1.673	1.661

• Critical point (704.93°F, 3200.11 psia). Values directly below underscored viscosities are for water.

Viscosity of Water and Liquid Petroleum Products[34,37,44]

(crude oils, fuel oils, and lubricants are at a constant pressure of 1 atm, all other fluids are at their saturation pressure)



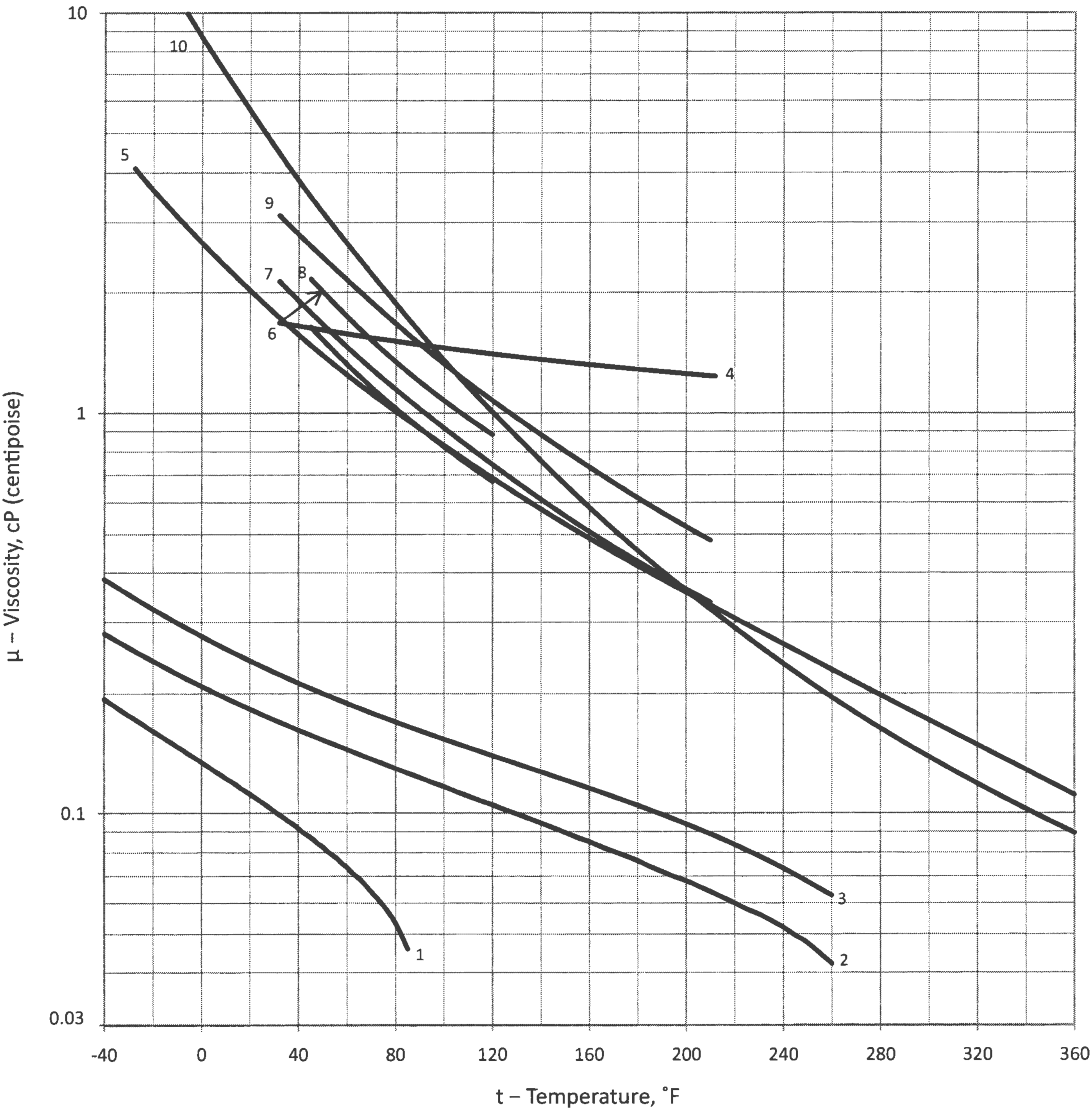
1. Ethane
2. Propane
3. Butane
4. Natural Gasoline
5. Gasoline
6. Water
7. Kerosene
8. 48 Deg API Crude
9. 40 Deg API Crude
10. 35.6 Deg API Crude
11. 32.6 Deg API Crude
12. Salt Creek Crude
13. Fuel 3 Max
14. Fuel 5 Min
15. SAE 10 Lube
16. SAE 30 Lube
17. Fuel 5 Max, Fuel 6 Min
18. SAE 70 Lube
19. Bunker C Fuel

Data taken from ASHRAE Handbook: Fundamentals, CRC Handbook of Chemistry and Physics (90 edition) and Petroleum Refinery Engineering.

CRANE

Viscosity of Various Liquids[34,35,10,38,42]

(percent solutions and brines are at a constant pressure of 1 atm, all other fluids are at their saturation pressure)

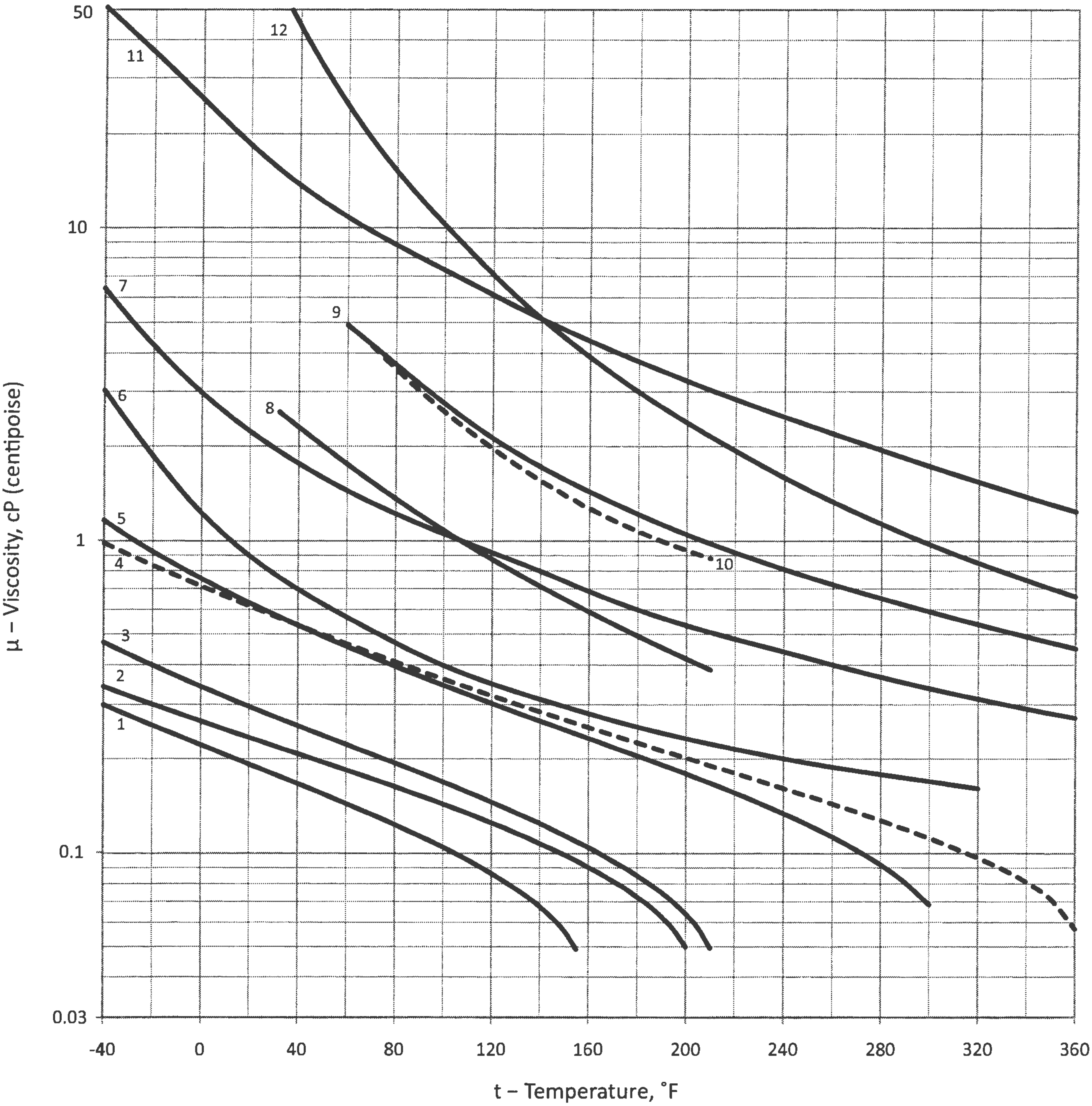


1. Carbon Dioxide
2. Ammonia
3. Methyl Chloride
4. Mercury
5. Ethanol
6. 10% Sodium Chloride Brine
7. 10% Calcium Chloride Brine
8. 20% Sodium Chloride Brine
9. 20% Calcium Chloride Brine
10. Isopropanol

Data taken from ASHRAE Handbook: Fundamentals, Marks' Handbook for Mechanical Engineers (11th edition), Perry's Chemical Engineers Handbook (8th edition), Yaws' Handbook of Thermodynamic and Physical Properties of Chemical Compounds, and Molecular Knowledge System's Fluid Database.

Viscosity of Various Liquids[34,35,42]

(percent solutions and brines are at a constant pressure of 1 atm, all other fluids are at their saturation pressure)



1. R-507, R-404a, and R410A
2. R-22
3. R-134a
4. R-123
5. R-245fa
6. Dowtherm J
7. Slytherm XLT
8. 20% Sulphuric Acid
9. Dowtherm A
10. 20% Sodium Hydroxide
11. Slytherm 800
12. Dowtherm MX

Data taken from ASHRAE Handbook: Fundamentals, Marks' Handbook for Mechanical Engineers (11th edition), Yaws' Handbook of Thermodynamic and Physical Properties of Chemical Compounds, and Molecular Knowledge System's Fluid Database.

Viscosity of Gases and Vapors[10,41]

(gases are at a constant pressure of 14.7 psia)

Sutherland's Formula:

$$\mu = \mu_0 \left(\frac{0.555T_0 + C}{0.555T + C} \right) \left(\frac{T}{T_0} \right)^{3/2}$$

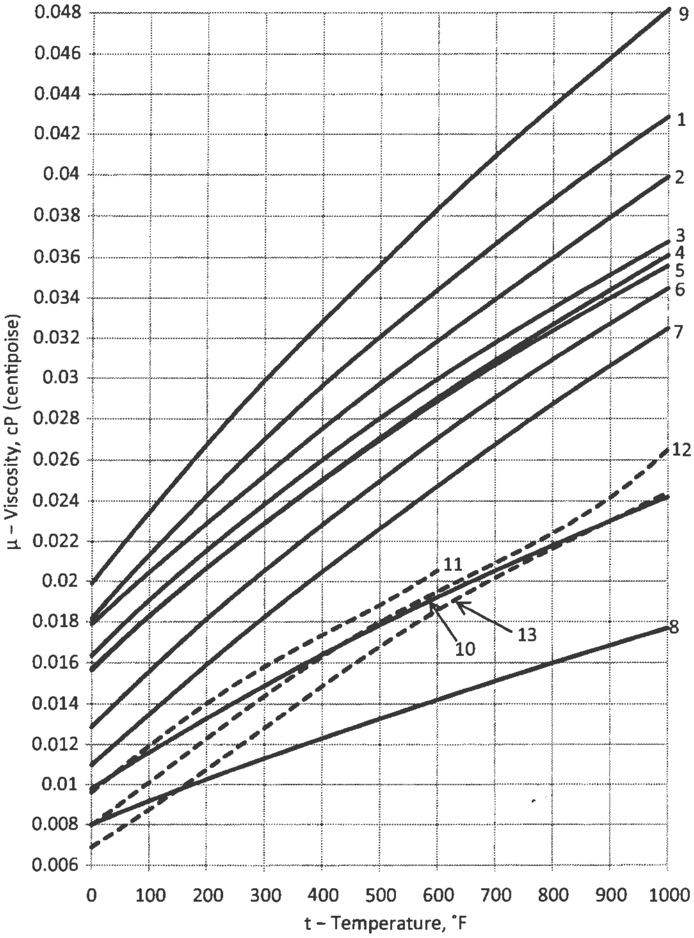
where:

μ = viscosity (cP) at temperature T (°R)
μ_0 = viscosity (cP) at temperature T_0 (°R)
T = absolute temperature in degrees Rankine (459.67 + °F)
T_0 = absolute temperature (°R), for which viscosity is known.
C = Sutherland's constant

***Note:** The variation of viscosity with pressure is small for most gases. For gases given on this page, the correction of viscosity for pressure is less than 10% for pressures up to 500 psia*

Fluid	Approximate Values of C	μ_0 (for T_0 = 527.67°R)
O_2	140	0.02304
Air	110	0.01822
N_2	120	0.01749
CO_2	244	0.01473
CO	116	0.01745
SO_2	349	0.01270
NH_3	528	0.009945
H_2	96	0.008804
Ar	157	0.02234
CH_4	173	0.01103

***Note:** Reference viscosity taken from Perry's Chemical Engineer's Handbook (8th edition). Values of C were calculated to match viscosity curves from Perry's Handbook.*

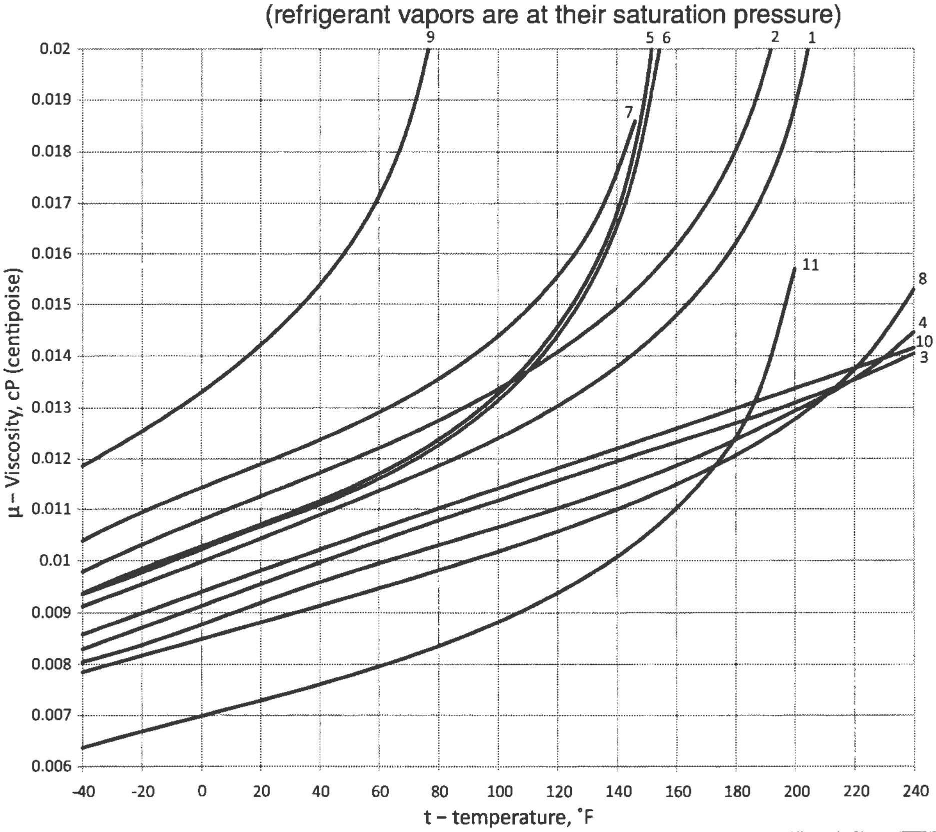


1. Oxygen
2. Helium
3. Air
4. Nitrogen
5. Carbon Monoxide
6. Carbon Dioxide
7. Sulfur Dioxide
8. Hydrogen
9. Argon
10. Methane
11. SG = .5 Hydrocarbon Vapor
12. SG = .75 Hydrocarbon Vapor
13. SG = 1.00 Hydrocarbon Vapor

Data taken from Perry's Chemical Engineer's Handbook (8th edition). An estimate of these curves can be created using Sutherland's formula, Sutherland's constant, o reference viscosity and reference temperature. The hydrocarbon vapors are calculated using the Natural Gas Engineering Handbook using mole fractions for N2 of 0.1, CO2 of 0.08, H2S of 0.02. These are representative only. Exact characteristics of a hydrocarbon vapor require knowledge of its specific constituents and properties.

Viscosity of Refrigerant Vapors[34,10]

(refrigerant vapors are at their saturation pressure)



1. R-134a
2. R-22
3. R-123
4. R-245fa
5. R-507
6. R-404A
7. R-410A
8. R-717 Ammonia
9. R-744 Carbon Dioxide
10. Methyl Chloride
11. R-290 Propane

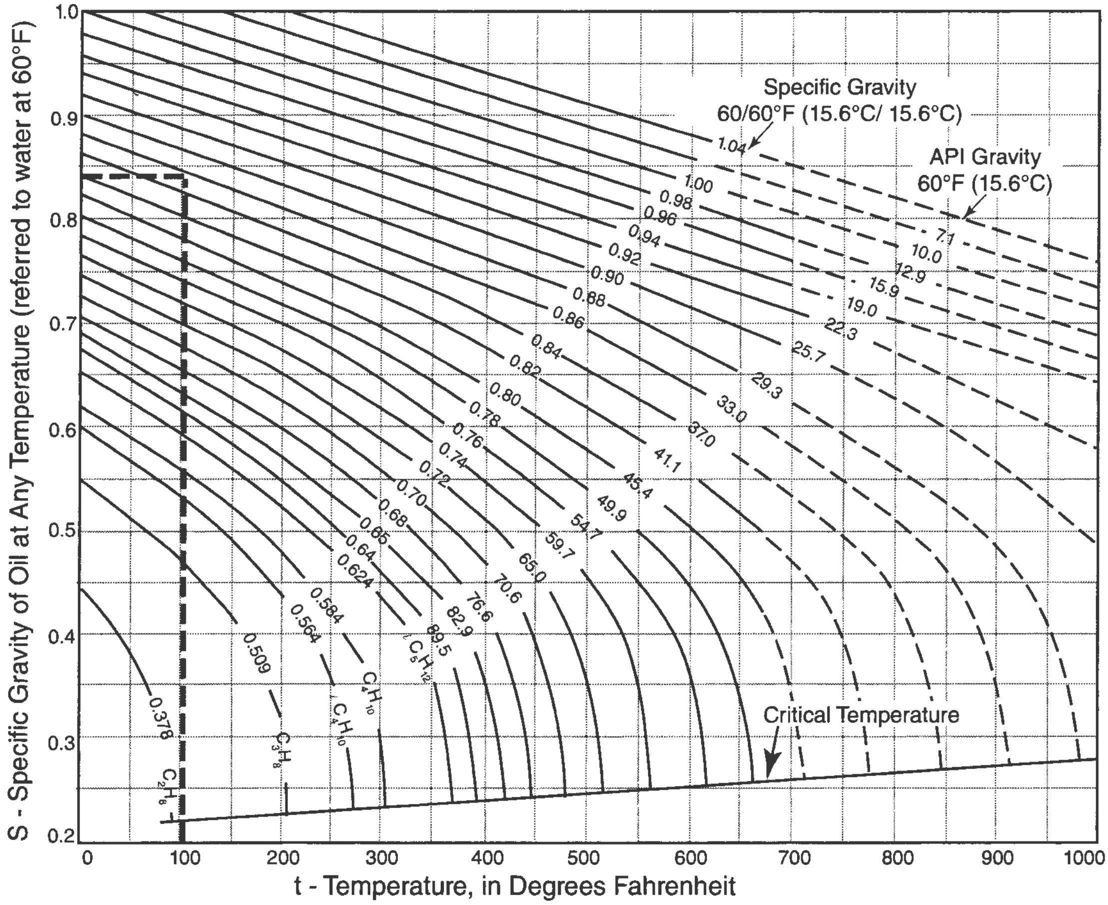
Data taken from the ASHRAE Handbook: Fundamentals.
Data for methyl chloride taken from Perry's Chemical Engineers' Handbook (8th edition).

Physical Properties of Water[33]

Specific Gravity of Water at 60°F = 1.00				
Temperature of Water t °F	Saturation (vapor) Pressure P_v' psia	Specific Volume $\overline{V}$ ft³/lb	Weight Density ρ lb/ft³	Weight per gallon lb/gal
32	0.08865	0.016022	62.414	8.3436
40	0.12173	0.016020	62.422	8.3446
50	0.17813	0.016024	62.406	8.3425
60	0.25639	0.016035	62.364	8.3368
70	0.36334	0.016052	62.298	8.3280
80	0.50744	0.016074	62.212	8.3166
90	0.69899	0.016100	62.112	8.3031
100	0.95044	0.016131	61.992	8.2872
110	1.2766	0.016166	61.858	8.2692
120	1.6949	0.016205	61.709	8.2493
130	2.2258	0.016247	61.550	8.2280
140	2.8929	0.016293	61.376	8.2048
150	3.7231	0.016342	61.192	8.1802
160	4.7472	0.016394	60.998	8.1542
170	5.9998	0.016449	60.794	8.1270
180	7.5196	0.016507	60.580	8.0984
190	9.3497	0.016569	60.354	8.0681
200	11.538	0.016633	60.121	8.0371
210	14.136	0.016701	59.877	8.0043
212	14.709	0.016715	59.827	7.9976
220	17.201	0.016771	59.627	7.9709
230	20.795	0.016845	59.365	7.9359
240	24.985	0.016921	59.098	7.9003
250	29.843	0.017001	58.820	7.8631
260	35.445	0.017084	58.534	7.8249
270	41.874	0.017170	58.241	7.7857
280	49.218	0.017259	57.941	7.7456
290	57.567	0.017352	57.630	7.7040
300	67.021	0.017449	57.310	7.6612
350	134.60	0.017987	55.596	7.4321
400	247.22	0.018639	53.651	7.1721
450	422.42	0.019437	51.448	6.8776
500	680.53	0.020442	48.919	6.5395
550	1044.8	0.021761	45.954	6.1431
600	1542.5	0.023631	42.317	5.6570
650	2207.7	0.026720	37.425	5.0030
700	3092.9	0.036825	27.155	3.6302

Weight per gallon is based on 7.48052 gallons per cubic foot.

Specific Gravity-Temperature Relationship for Petroleum Oils[44]



Weight Density and Specific Gravity of Various Liquids[34,35,37,38]

Liquid	Temperature t °F	Weight Density ρ lb/ft³	Specific Gravity S
Acetone	60	49.7	0.797
Benzene	41	55.7	0.893
Benzene	60	55.1	0.883
CaCl Brine, 10%	32	67.9	1.089
CaCl Brine, 20%	32	74.0	1.186
NaCl Brine, 10%	32	67.2	1.077
NaCl Brine, 20%	32	72.2	1.157
Carbon disulphide	32	80.8	1.295
Carbon disulphide	60	79.3	1.271
*Gasoline	60	49.9	0.801
*Diesel fuel	60	60.6	0.971
*Kerosene	60	51.2	0.821
Mercury	19.4	850	13.63
Mercury	39.2	848	13.60
Mercury	60.8	846	13.57
Mercury	80.6	845	13.55
Mercury	100.4	843	13.52
*Milk min	...	64.2	1.029
*Milk max	...	64.6	1.036
*Olive Oil	60	57.3	0.919
Pentane-n	32	40.3	0.646
Pentane-n	60	39.4	0.631
*SAE 10W	60	54.6	0.876
*SAE 30	60	55.3	0.887
*SAE 50	60	55.6	0.892
32.6° API Crude	60	53.8	0.862
32.6° API Crude	130	52.4	0.840
35.6° API Crude	60	52.8	0.847
35.6° API Crude	130	51.7	0.829
40° API Crude	60	51.5	0.825
40° API Crude	130	50.2	0.805
48° API Crude	60	49.2	0.788
48° API Crude	130	47.4	0.760

*Approximate values, properties for these fluids can vary.

Values in the table at the left were taken from references 34, 35, 37, 38 and from Smithsonian Physical Tables.

Physical Properties of Gases[10,37]

(Approximate values at 68°F and 14.7psia)

Name of Gas	Chemical Formula	Molecular Mass M_r	Weight Density ρ lb/ft^3	Specific Gravity relative to air S_g	Individual Gas Constant R $ft \cdot lb / lb_m \cdot °R$	Specific Heat c_p Btu/(lb °R)	Specific Heat c_v Btu/(lb °R)	Specific Heat c_p Btu/(ft^3 °R)	Specific Heat c_v Btu/(ft^3 °R)	k c_p/c_v
Acetylene (ethyne)	C_2H_2	26.080	0.0677	0.900	59.254	0.3987	0.3226	0.0270	0.0218	1.24
Air	---	28.966	0.0752	1.000	53.350	0.2390	0.1705	0.0180	0.0128	1.40
Ammonia	NH_3	17.031	0.0442	0.588	90.737	0.4967	0.3801	0.0220	0.0168	1.31
Argon	Ar	39.948	0.1037	1.379	38.684					
Butane	C_4H_{10}	58.124	0.1509	2.007	26.587	0.3987	0.3645	0.0602	0.0550	1.09
Carbon dioxide	CO_2	44.010	0.1142	1.519	35.114	0.2007	0.1556	0.0229	0.0178	1.29
Carbon monoxide	CO	28.010	0.0727	0.967	55.171	0.2484	0.1775	0.0181	0.0129	1.40
Chlorine	Cl_2	70.906	0.1841	2.448	21.794	0.1142	0.0862	0.0210	0.0159	1.32
Ethane	C_2H_6	30.069	0.0781	1.038	51.393	0.4112	0.3451	0.0321	0.0269	1.19
Ethylene	C_2H_4	28.053	0.0728	0.968	55.087	0.3615	0.2907	0.0263	0.0212	1.24
Helium	He	4.003	0.0104	0.138	386.048					
Hydrogen chloride	HCl	36.461	0.0946	1.259	42.384	0.1909	0.1364	0.0181	0.0129	1.40
Hydrogen	H_2	2.016	0.0052	0.070	766.542	3.4053	2.4203	0.0178	0.0127	1.41
Hydrogen sulphide	H_2S	34.076	0.0885	1.176	45.350	0.2392	0.1810	0.0212	0.0160	1.32
Methane	CH_4	16.043	0.0416	0.554	96.325	0.5285	0.4047	0.0220	0.0169	1.31
Methyl chloride	CH_3Cl	50.490	0.1311	1.743	30.607	0.1906	0.1513	0.0250	0.0198	1.26
Natural gas	---	19.500	0.0506	0.673	79.249	0.5600	0.4410	0.0281	0.0221	1.27
Nitric oxide	NO	30.006	0.0779	1.036	51.501	0.2382	0.1721	0.0186	0.0134	1.38
Nitrogen	N_2	28.013	0.0727	0.967	55.165	0.2483	0.1774	0.0181	0.0129	1.40
Nitrous oxide	N_2O	44.013	0.1143	1.519	35.111	0.2086	0.1635	0.0238	0.0187	1.28
Oxygen	O_2	31.999	0.0831	1.105	48.294	0.2189	0.1568	0.0182	0.0130	1.40
Propane	C_3H_8	44.097	0.1145	1.522	35.044	0.3923	0.3472	0.0449	0.0397	1.13
Propene (propylene)	C_3H_6	42.081	0.1092	1.453	36.723	0.3631	0.3159	0.0397	0.0345	1.15
Sulphur dioxide	SO_2	64.059	0.1663	2.212	24.124	0.1481	0.1171	0.0246	0.0195	1.26

Molecular mass taken from the CRC Handbook of Chemistry and Physics (90th edition). Weight density, specific gravity relative to air and individual gas constant are based off of the ideal gas law and the universal gas constant given in the CRC Handbook.

Ideal gas law calculations were based on conditions at 68°F and 14.70 psia.

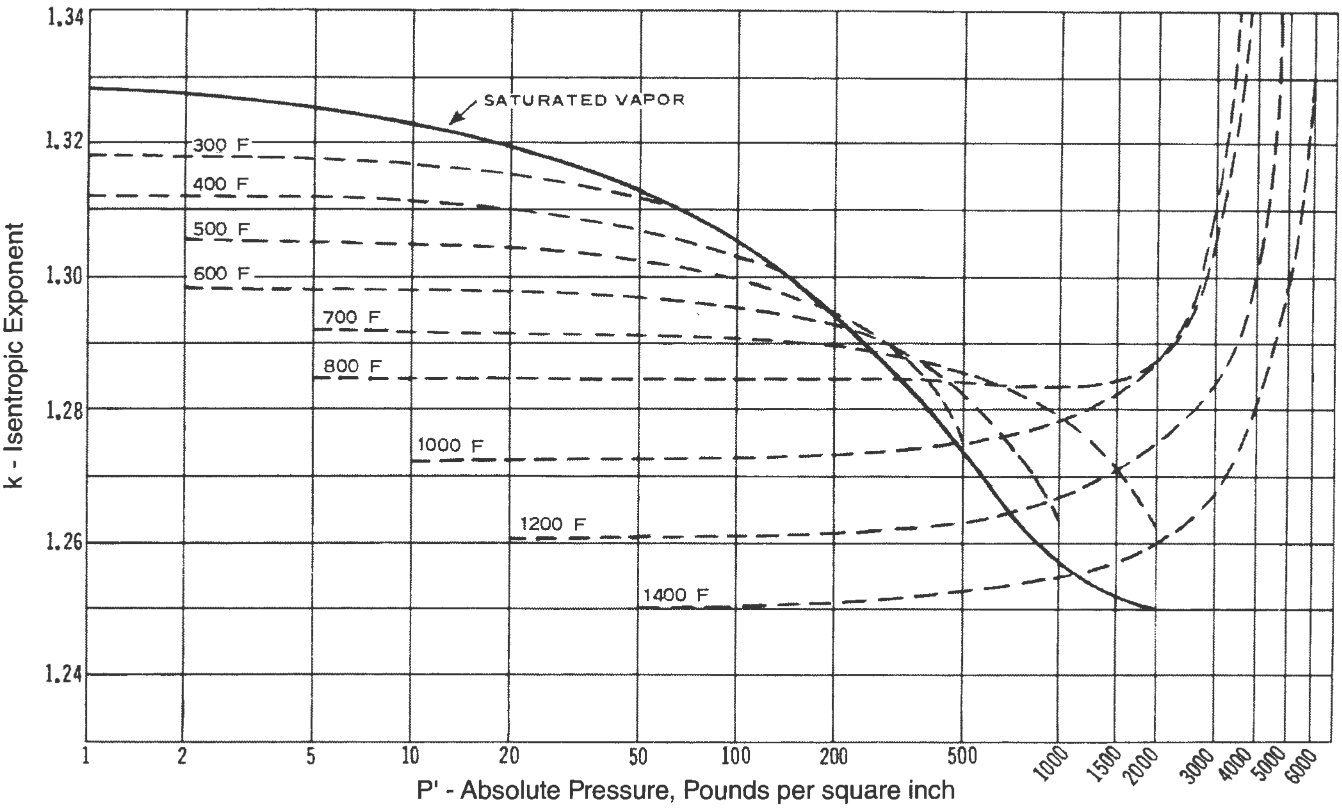
Values for isobaric heat capacity were taken from Perry's Chemical Engineer Handbook (8th edition). Values for isochoric heat capacity were calculated from the approximate relationship of $c_p - R = c_v$.

Volumetric Composition and Specific Gravity of Gaseous Fuels[38]

Type of Gas	Chemical Compositon - Percent by Volume								Specific Gravity Relative to Air S_g
	CO	H_2	CH_4	C_2H_6	Illuminates (assumed C_2H_4)	CO_2	O_2	N_2	
Carbureted water gas	24.1	32.5	9.0	2.2	10.3	4.6	0.6	16.7	0.666
Coal gas	5.9	53.2	29.6	0.0	2.7	1.4	0.7	6.5	0.376
Natural gas	0.0	0.0	78.8	14.0	0.0	0.4	0.0	6.8	0.654
Producer gas	26.0	3.0	0.5	0.0	0.0	2.5	0.0	56.0	0.950
Blast furnace gas	26.5	3.5	0.2	0.0	0.0	12.8	0.1	56.9	1.006
Coke oven gases:									
Pittsburgh bed	6.8	56.4	31.1	0.0	3.2	1.3	0.0	1.1	0.339
Elkhorn bed	7.7	55.0	31.0	0.2	4.0	1.1	0.0	1.0	0.352
Sewell bed	5.5	64.8	26.5	0.0	2.5	0.7	0.0	1.0	0.287
Pocahontas no. 4	5.0	75.0	18.0	0.0	1.1	0.4	0.0	0.5	0.222
Illinois, Franklin Co.	14.5	56.9	21.0	0.0	2.8	3.8	0.0	1.0	0.391
Utah, Sunnyside	14.5	51.3	26.0	0.5	3.7	3.0	0.0	1.0	0.416

Specific gravity calculated from percent composition and molecular mass given in "Physical Properties of Gases" table. Compositions taken from Mark's Standard Handbook for Mechanical Engineers (11th edition).

Steam Values of Isentropic Exponent, k[45]



For small changes in pressure (or volume) along an isentropic, pv^k = constant.

Reasonable Velocities For the Flow of Water Through Pipe	
Service Condition	Reasonable Velocity (ft/sec)
Boiler Feed	8 to 15
Pump Suction and Drain Lines	4 to 7
General Service	4 to 10
City	up to 7

Reasonable Velocities for Flow of Steam Through Pipe			
Condition of Steam	Pressure (psig)	Service	Reasonable Velocity (ft/min)
Saturated	0 to 25	Heating (short lines)	4,000 to 6,000
	25 and up	Power house equipment, process piping, etc.	6,000 to 10,000
Superheated	200 and up	Boiler and turbine leads, etc.	7,000 to 20,000

Weight Density and Specific Volume of Gases and Vapors

Weight Density of Air (lb/ft^3) for Absolute Pressures (psia) and Gauge Pressures (psig) Indicated
(Based on an atmospheric pressure of 14.696 and a molecular mass of 28.966 for air)

Air Temp °F	°R	0 14.696	5 19.696	10 24.696	20 34.696	30 44.696	40 54.696	50 64.696	60 74.696	70 84.696	80 94.696	90 104.696	100 114.696	110 124.696	120 134.696	130 144.696	140 154.696	150 164.696
30	489.67	0.0810	0.1086	0.1361	0.1912	0.2464	0.3015	0.3566	0.4117	0.4669	0.5220	0.5771	0.6322	0.6873	0.7425	0.7976	0.8527	0.9078
40	499.67	0.0794	0.1064	0.1334	0.1874	0.2414	0.2955	0.3495	0.4035	0.4575	0.5115	0.5656	0.6196	0.6736	0.7276	0.7816	0.8356	0.8897
50	509.67	0.0778	0.1043	0.1308	0.1837	0.2367	0.2897	0.3426	0.3956	0.4485	0.5015	0.5545	0.6074	0.6604	0.7133	0.7663	0.8192	0.8722
60	519.67	0.0763	0.1023	0.1283	0.1802	0.2321	0.2841	0.3360	0.3880	0.4399	0.4918	0.5438	0.5957	0.6477	0.6996	0.7515	0.8035	0.8554
70	529.67	0.0749	0.1004	0.1258	0.1768	0.2278	0.2787	0.3297	0.3806	0.4316	0.4826	0.5335	0.5845	0.6354	0.6864	0.7374	0.7883	0.8393
80	539.67	0.0735	0.0985	0.1235	0.1735	0.2235	0.2736	0.3236	0.3736	0.4236	0.4736	0.5236	0.5736	0.6237	0.6737	0.7237	0.7737	0.8237
90	549.67	0.0722	0.0967	0.1213	0.1704	0.2195	0.2686	0.3177	0.3668	0.4159	0.4650	0.5141	0.5632	0.6123	0.6614	0.7105	0.7596	0.8087
100	559.67	0.0709	0.0950	0.1191	0.1673	0.2156	0.2638	0.3120	0.3602	0.4085	0.4567	0.5049	0.5531	0.6014	0.6496	0.6978	0.7461	0.7943
110	569.67	0.0696	0.0933	0.1170	0.1644	0.2118	0.2592	0.3065	0.3539	0.4013	0.4487	0.4961	0.5434	0.5908	0.6382	0.6856	0.7330	0.7803
120	579.67	0.0684	0.0917	0.1150	0.1616	0.2081	0.2547	0.3012	0.3478	0.3944	0.4409	0.4875	0.5341	0.5806	0.6272	0.6738	0.7203	0.7669
130	589.67	0.0673	0.0902	0.1130	0.1588	0.2046	0.2504	0.2961	0.3419	0.3877	0.4335	0.4792	0.5250	0.5708	0.6166	0.6623	0.7081	0.7539
140	599.67	0.0661	0.0887	0.1112	0.1562	0.2012	0.2462	0.2912	0.3362	0.3812	0.4262	0.4712	0.5163	0.5613	0.6063	0.6513	0.6963	0.7413
150	609.67	0.0651	0.0872	0.1093	0.1536	0.1979	0.2422	0.2864	0.3307	0.3750	0.4192	0.4635	0.5078	0.5521	0.5963	0.6406	0.6849	0.7291
175	634.67	0.0625	0.0838	0.1050	0.1476	0.1901	0.2326	0.2751	0.3177	0.3602	0.4027	0.4453	0.4878	0.5303	0.5728	0.6154	0.6579	0.7004
200	659.67	0.0601	0.0806	0.1010	0.1420	0.1829	0.2238	0.2647	0.3056	0.3465	0.3875	0.4284	0.4693	0.5102	0.5511	0.5920	0.6330	0.6739
225	684.67	0.0579	0.0776	0.0974	0.1368	0.1762	0.2156	0.2550	0.2945	0.3339	0.3733	0.4127	0.4522	0.4916	0.5310	0.5704	0.6098	0.6493
250	709.67	0.0559	0.0749	0.0939	0.1320	0.1700	0.2080	0.2461	0.2841	0.3221	0.3602	0.3982	0.4362	0.4743	0.5123	0.5503	0.5884	0.6264
275	734.67	0.0540	0.0724	0.0907	0.1275	0.1642	0.2009	0.2377	0.2744	0.3112	0.3479	0.3846	0.4214	0.4581	0.4949	0.5316	0.5683	0.6051
300	759.67	0.0522	0.0700	0.0877	0.1233	0.1588	0.1943	0.2299	0.2654	0.3009	0.3365	0.3720	0.4075	0.4430	0.4786	0.5141	0.5496	0.5852
350	809.67	0.0490	0.0657	0.0823	0.1157	0.1490	0.1823	0.2157	0.2490	0.2823	0.3157	0.3490	0.3824	0.4157	0.4490	0.4824	0.5157	0.5490
400	859.67	0.0461	0.0618	0.0775	0.1089	0.1403	0.1717	0.2031	0.2345	0.2659	0.2973	0.3287	0.3601	0.3915	0.4229	0.4543	0.4857	0.5171
450	909.67	0.0436	0.0584	0.0733	0.1029	0.1326	0.1623	0.1920	0.2216	0.2513	0.2810	0.3106	0.3403	0.3700	0.3997	0.4293	0.4590	0.4887
500	959.67	0.0413	0.0554	0.0695	0.0976	0.1257	0.1538	0.1820	0.2101	0.2382	0.2663	0.2945	0.3226	0.3507	0.3788	0.4070	0.4351	0.4632
550	1009.67	0.0393	0.0527	0.0660	0.0928	0.1195	0.1462	0.1730	0.1997	0.2264	0.2531	0.2799	0.3066	0.3333	0.3601	0.3868	0.4135	0.4403
600	1059.67	0.0374	0.0502	0.0629	0.0884	0.1138	0.1393	0.1648	0.1903	0.2157	0.2412	0.2667	0.2921	0.3176	0.3431	0.3686	0.3940	0.4195

°F	°R	175 189.696	200 214.696	225 239.696	250 264.696	300 314.696	400 414.696	500 514.696	600 614.696	700 714.696	800 814.696	900 914.696	1000 1014.696
30	489.67	1.0456	1.1834	1.3212	1.4590	1.7347	2.2859	2.8371	3.3883	3.9395	4.4907	5.0419	5.5932
40	499.67	1.0247	1.1598	1.2948	1.4298	1.6999	2.2401	2.7803	3.3205	3.8607	4.4009	4.9410	5.4812
50	509.67	1.0046	1.1370	1.2694	1.4018	1.6666	2.1962	2.7258	3.2553	3.7849	4.3145	4.8441	5.3737
60	519.67	0.9853	1.1151	1.2450	1.3748	1.6345	2.1539	2.6733	3.1927	3.7121	4.2315	4.7509	5.2703
70	529.67	0.9667	1.0941	1.2215	1.3489	1.6037	2.1132	2.6228	3.1324	3.6420	4.1516	4.6612	5.1708
80	539.67	0.9488	1.0738	1.1988	1.3239	1.5739	2.0741	2.5742	3.0744	3.5745	4.0747	4.5748	5.0750
90	549.67	0.9315	1.0543	1.1770	1.2998	1.5453	2.0363	2.5274	3.0184	3.5095	4.0005	4.4916	4.9826
100	559.67	0.9149	1.0354	1.1560	1.2766	1.5177	2.0000	2.4822	2.9645	3.4468	3.9291	4.4113	4.8936
110	569.67	0.8988	1.0172	1.1357	1.2541	1.4911	1.9649	2.4387	2.9125	3.3863	3.8601	4.3339	4.8077
120	579.67	0.8833	0.9997	1.1161	1.2325	1.4653	1.9310	2.3966	2.8622	3.3279	3.7935	4.2591	4.7248
130	589.67	0.8683	0.9827	1.0972	1.2116	1.4405	1.8982	2.3560	2.8137	3.2714	3.7292	4.1869	4.6446
140	599.67	0.8538	0.9664	1.0789	1.1914	1.4165	1.8666	2.3167	2.7668	3.2169	3.6670	4.1171	4.5672
150	609.67	0.8398	0.9505	1.0612	1.1719	1.3932	1.8359	2.2787	2.7214	3.1641	3.6068	4.0495	4.4923
175	634.67	0.8067	0.9131	1.0194	1.1257	1.3383	1.7636	2.1889	2.6142	3.0395	3.4648	3.8900	4.3153
200	659.67	0.7762	0.8785	0.9808	1.0830	1.2876	1.6968	2.1060	2.5151	2.9243	3.3334	3.7426	4.1518
225	684.67	0.7478	0.8464	0.9449	1.0435	1.2406	1.6348	2.0291	2.4233	2.8175	3.2117	3.6060	4.0002
250	709.67	0.7215	0.8166	0.9117	1.0067	1.1969	1.5772	1.9576	2.3379	2.7183	3.0986	3.4789	3.8593
275	734.67	0.6969	0.7888	0.8806	0.9725	1.1562	1.5236	1.8910	2.2584	2.6258	2.9931	3.3605	3.7279
300	759.67	0.6740	0.7628	0.8516	0.9405	1.1181	1.4734	1.8287	2.1840	2.5393	2.8946	3.2499	3.6053
350	809.67	0.6324	0.7157	0.7991	0.8824	1.0491	1.3824	1.7158	2.0492	2.3825	2.7159	3.0493	3.3826
400	859.67	0.5956	0.6741	0.7526	0.8311	0.9881	1.3020	1.6160	1.9300	2.2440	2.5579	2.8719	3.1859
450	909.67	0.5629	0.6370	0.7112	0.7854	0.9338	1.2305	1.5272	1.8239	2.1206	2.4173	2.7140	3.0108
500	959.67	0.5335	0.6038	0.6742	0.7445	0.8851	1.1664	1.4476	1.7289	2.0101	2.2914	2.5726	2.8539
550	1009.67	0.5071	0.5739	0.6408	0.7076	0.8413	1.1086	1.3759	1.6433	1.9106	2.1779	2.4452	2.7126
600	1059.67	0.4832	0.5469	0.6105	0.6742	0.8016	1.0563	1.3110	1.5657	1.8204	2.0752	2.3299	2.5846

Air Density Table
The table at the left is calculated using the ideal gas law. Correction for compressibility, the deviation from the ideal gas law, would be less than three percent and has not been applied.

For Gases Other than Air
The weight density of gases other than air can be determined from this table by multiplying the density listed for air by the specific gravity of the gas relative to air, as listed in the tables on page A-9.

Properties of Saturated Steam and Saturated Water[33]

Uses conversion of 1 psi = 2.03602 in of Hg. Gauge pressure based on a 14.696 psia reference (EL 0 ft).

Absolute Pressure		Vacuum Inches of Hg	Temperature °F t_s	Specific Enthalpy Btu/lb		Specific Latent Heat of Evaporation	Specific Volume ft³/lb	
psia, P'	inches of Hg			h_f	h_g	Btu/lb h_{fg}	$\bar{V}_f$	$\bar{V}_g$
0.08859	0.18	29.74	31.984	-0.034466	1075.2	1075.2344	0.016022	3304.1
0.10	0.20	29.72	35.005	3.0088	1076.5	1073.4912	0.01602	2945
0.15	0.31	29.62	45.429	13.483	1081.1	1067.617	0.016021	2004.3
0.20	0.41	29.51	53.132	21.204	1084.4	1063.196	0.016027	1525.9
0.25	0.51	29.41	59.293	27.371	1087.1	1059.729	0.016034	1235.2
0.30	0.61	29.31	64.452	32.532	1089.4	1056.868	0.016042	1039.4
0.35	0.71	29.21	68.906	36.985	1091.3	1054.315	0.01605	898.41
0.40	0.81	29.11	72.834	40.911	1093	1052.089	0.016057	791.86
0.45	0.92	29.00	76.355	44.428	1094.5	1050.072	0.016065	708.44
0.50	1.02	28.90	79.549	47.618	1095.9	1048.282	0.016073	641.32
0.60	1.22	28.70	85.18	53.242	1098.3	1045.058	0.016087	539.9
0.70	1.43	28.50	90.046	58.1	1100.4	1042.3	0.0161	466.81
0.80	1.63	28.29	94.342	62.389	1102.3	1039.911	0.016113	411.57
0.90	1.83	28.09	98.195	66.236	1103.9	1037.664	0.016125	368.32
1.0	2.04	27.88	101.69	69.728	1105.4	1035.672	0.016137	333.51
1.2	2.44	27.48	107.87	75.892	1108.1	1032.208	0.016158	280.89
1.4	2.85	27.07	113.21	81.225	1110.3	1029.075	0.016178	242.95
1.6	3.26	26.66	117.93	85.939	1112.3	1026.361	0.016196	214.27
1.8	3.66	26.26	122.17	90.172	1114.1	1023.928	0.016214	191.8
2.0	4.07	25.85	126.03	94.019	1115.8	1021.781	0.01623	173.72
2.2	4.48	25.44	129.56	97.551	1117.2	1019.649	0.016245	158.84
2.4	4.89	25.03	132.83	100.82	1118.6	1017.78	0.01626	146.37
2.6	5.29	24.63	135.88	103.86	1119.9	1016.04	0.016273	135.77
2.8	5.70	24.22	138.73	106.71	1121.1	1014.39	0.016287	126.65
3.0	6.11	23.81	141.42	109.39	1122.2	1012.81	0.016299	118.7
3.5	7.13	22.79	147.52	115.49	1124.7	1009.21	0.016329	102.72
4.0	8.14	21.78	152.91	120.89	1126.9	1006.01	0.016356	90.628
4.5	9.16	20.76	157.77	125.74	1128.9	1003.16	0.016382	81.154
5.0	10.18	19.74	162.18	130.16	1130.7	1000.54	0.016406	73.523
5.5	11.20	18.72	166.24	134.23	1132.4	998.17	0.016428	67.242
6.0	12.22	17.70	170	137.99	1133.9	995.91	0.016449	61.979
6.5	13.23	16.69	173.51	141.5	1135.3	993.8	0.016469	57.503
7.0	14.25	15.67	176.79	144.79	1136.6	991.81	0.016488	53.649
7.5	15.27	14.65	179.88	147.9	1137.8	989.9	0.016507	50.293
8.0	16.29	13.63	182.81	150.83	1139	988.17	0.016524	47.345
8.5	17.31	12.61	185.58	153.61	1140.1	986.49	0.016541	44.733
9.0	18.32	11.60	188.22	156.27	1141.1	984.83	0.016558	42.404
9.5	19.34	10.58	190.75	158.8	1142.1	983.3	0.016573	40.312
10.0	20.36	9.56	193.16	161.22	1143.1	981.88	0.016589	38.423
11.0	22.40	7.52	197.7	165.79	1144.8	979.01	0.016618	35.145
12.0	24.43	5.49	201.91	170.02	1146.4	976.38	0.016646	32.398
13.0	26.47	3.45	205.83	173.97	1148	974.03	0.016672	30.061
14.0	28.50	1.42	209.52	177.68	1149.4	971.72	0.016697	28.048

Pressure psi		Temperature °F t_s	Specific Enthalpy Btu/lb		Specific Latent Heat of Evaporation	Specific Volume ft³/lb	
P'	Gauge P		h_f	h_g	Btu/lb h_{fg}	$\bar{V}_f$	$\bar{V}_g$
14.696	0.0	211.95	180.13	1150.3	970.17	0.016714	26.804
15.0	0.3	212.99	181.18	1150.7	969.52	0.016721	26.295
16.0	1.3	216.27	184.49	1151.9	967.41	0.016745	24.755
17.0	2.3	219.39	187.63	1153.1	965.47	0.016767	23.39
18.0	3.3	222.36	190.63	1154.2	963.57	0.016788	22.173
19.0	4.3	225.2	193.5	1155.2	961.7	0.016809	21.079
20.0	5.3	227.92	196.25	1156.2	959.95	0.016829	20.092
21.0	6.3	230.53	198.88	1157.1	958.22	0.016849	19.196
22.0	7.3	233.03	201.42	1158	956.58	0.016868	18.378
23.0	8.3	235.45	203.86	1158.9	955.04	0.016886	17.629
24.0	9.3	237.78	206.23	1159.7	953.47	0.016904	16.941
25.0	10.3	240.03	208.51	1160.5	951.99	0.016922	16.306
26.0	11.3	242.21	210.72	1161.3	950.58	0.016939	15.719
27.0	12.3	244.33	212.86	1162.1	949.24	0.016955	15.173
28.0	13.3	246.38	214.94	1162.8	947.86	0.016972	14.665
29.0	14.3	248.37	216.96	1163.5	946.54	0.016988	14.192
30.0	15.3	250.3	218.93	1164.1	945.17	0.017003	13.748
31.0	16.3	252.18	220.84	1164.8	943.96	0.017019	13.333
32.0	17.3	254.02	222.7	1165.4	942.7	0.017034	12.942

CRANE

Properties of Saturated Steam and Saturated Water[33]

Pressure psi		Temperature °F	Specific Enthalpy Btu/lb		Specific Latent Heat of Evaporation	Specific Volume ft³/lb	
P'	Gauge P	t_s	h_f	h_g	Btu/lb h_{fg}	$\bar{V}_f$	$\bar{V}_g$
33.0	18.3	255.81	224.52	1166	941.48	0.017049	12.574
34.0	19.3	257.55	226.3	1166.6	940.3	0.017063	12.228
35.0	20.3	259.25	228.03	1167.2	939.17	0.017078	11.9
36.0	21.3	260.92	229.73	1167.7	937.97	0.017092	11.59
37.0	22.3	262.55	231.38	1168.3	936.92	0.017105	11.296
38.0	23.3	264.14	233.01	1168.8	935.79	0.017119	11.018
39.0	24.3	265.7	234.59	1169.3	934.71	0.017132	10.753
40.0	25.3	267.22	236.15	1169.8	933.65	0.017146	10.5
41.0	26.3	268.72	237.68	1170.3	932.62	0.017159	10.26
42.0	27.3	270.18	239.17	1170.8	931.63	0.017172	10.031
43.0	28.3	271.62	240.64	1171.3	930.66	0.017184	9.8119
44.0	29.3	273.03	242.08	1171.7	929.62	0.017197	9.6026
45.0	30.3	274.42	243.5	1172.2	928.7	0.017209	9.4023
46.0	31.3	275.78	244.89	1172.6	927.71	0.017221	9.2104
47.0	32.3	277.12	246.26	1173	926.74	0.017233	9.0264
48.0	33.3	278.43	247.6	1173.4	925.8	0.017245	8.8498
49.0	34.3	279.72	248.93	1173.8	924.87	0.017257	8.6802
50.0	35.3	280.99	250.23	1174.2	923.97	0.017268	8.5171
51.0	36.3	282.24	251.51	1174.6	923.09	0.01728	8.3602
52.0	37.3	283.47	252.77	1175	922.23	0.017291	8.2091
53.0	38.3	284.69	254.02	1175.4	921.38	0.017302	8.0636
54.0	39.3	285.88	255.24	1175.7	920.46	0.017314	7.9232
55.0	40.3	287.06	256.45	1176.1	919.65	0.017325	7.7878
56.0	41.3	288.22	257.64	1176.5	918.86	0.017335	7.657
57.0	42.3	289.36	258.81	1176.8	917.99	0.017346	7.5307
58.0	43.3	290.49	259.97	1177.1	917.13	0.017357	7.4086
59.0	44.3	291.6	261.11	1177.5	916.39	0.017367	7.2905
60.0	45.3	292.69	262.24	1177.8	915.56	0.017378	7.1762
61.0	46.3	293.77	263.35	1178.1	914.75	0.017388	7.0655
62.0	47.3	294.84	264.45	1178.4	913.95	0.017398	6.9583
63.0	48.3	295.9	265.53	1178.8	913.27	0.017409	6.8543
64.0	49.3	296.94	266.6	1179.1	912.5	0.017419	6.7535
65.0	50.3	297.96	267.66	1179.4	911.74	0.017429	6.6557
66.0	51.3	298.98	268.7	1179.7	911	0.017438	6.5607
67.0	52.3	299.98	269.74	1180	910.26	0.017448	6.4685
68.0	53.3	300.97	270.76	1180.2	909.44	0.017458	6.3789
69.0	54.3	301.95	271.77	1180.5	908.73	0.017468	6.2918
70.0	55.3	302.92	272.76	1180.8	908.04	0.017477	6.2071
71.0	56.3	303.87	273.75	1181.1	907.35	0.017487	6.1247
72.0	57.3	304.82	274.73	1181.3	906.57	0.017496	6.0445
73.0	58.3	305.75	275.69	1181.6	905.91	0.017506	5.9665
74.0	59.3	306.68	276.65	1181.9	905.25	0.017515	5.8905
75.0	60.3	307.59	277.59	1182.1	904.51	0.017524	5.8164
76.0	61.3	308.5	278.53	1182.4	903.87	0.017533	5.7442
77.0	62.3	309.4	279.46	1182.6	903.14	0.017542	5.6739
78.0	63.3	310.28	280.37	1182.9	902.53	0.017551	5.6052
79.0	64.3	311.16	281.28	1183.1	901.82	0.01756	5.5383
80.0	65.3	312.03	282.18	1183.3	901.12	0.017569	5.4729
81.0	66.3	312.89	283.07	1183.6	900.53	0.017578	5.4092
82.0	67.3	313.74	283.95	1183.8	899.85	0.017587	5.3469
83.0	68.3	314.58	284.83	1184	899.17	0.017596	5.2861
84.0	69.3	315.42	285.69	1184.3	898.61	0.017604	5.2266
85.0	70.3	316.25	286.55	1184.5	897.95	0.017613	5.1686
86.0	71.3	317.07	287.4	1184.7	897.3	0.017621	5.1118
87.0	72.3	317.88	288.24	1184.9	896.66	0.01763	5.0563
88.0	73.3	318.68	289.08	1185.1	896.02	0.017638	5.002
89.0	74.3	319.48	289.91	1185.3	895.39	0.017647	4.9489
90.0	75.3	320.27	290.73	1185.6	894.87	0.017655	4.8969
91.0	76.3	321.06	291.54	1185.8	894.26	0.017663	4.846
92.0	77.3	321.83	292.35	1186	893.65	0.017672	4.7962
93.0	78.3	322.6	293.15	1186.2	893.05	0.01768	4.7474
94.0	79.3	323.37	293.94	1186.4	892.46	0.017688	4.6996
95.0	80.3	324.12	294.73	1186.6	891.87	0.017696	4.6528
96.0	81.3	324.87	295.51	1186.7	891.19	0.017704	4.607
97.0	82.3	325.62	296.29	1186.9	890.61	0.017712	4.562
98.0	83.3	326.36	297.05	1187.1	890.05	0.01772	4.518
99.0	84.3	327.09	297.82	1187.3	889.48	0.017728	4.4748
100.0	85.3	327.82	298.57	1187.5	888.93	0.017736	4.4324
101.0	86.3	328.54	299.32	1187.7	888.38	0.017744	4.3908
102.0	87.3	329.25	300.07	1187.9	887.83	0.017752	4.35

Properties of Saturated Steam and Saturated Water[33]

Pressure psi		Temperature °F	Specific Enthalpy Btu/lb		Specific Latent Heat of Evaporation	Specific Volume ft³/lb	
P'	Gauge P	t_s	h_f	h_g	Btu/lb h_{fg}	$\overline{V}_f$	$\overline{V}_g$
103.0	88.3	329.96	300.81	1188	887.19	0.017759	4.31
104.0	89.3	330.67	301.54	1188.2	886.66	0.017767	4.2708
105.0	90.3	331.37	302.27	1188.4	886.13	0.017775	4.2322
106.0	91.3	332.06	303	1188.5	885.5	0.017782	4.1944
107.0	92.3	332.75	303.72	1188.7	884.98	0.01779	4.1572
108.0	93.3	333.43	304.43	1188.9	884.47	0.017798	4.1207
109.0	94.3	334.11	305.14	1189	883.86	0.017805	4.0849
110.0	95.3	334.78	305.84	1189.2	883.36	0.017813	4.0496
111.0	96.3	335.45	306.54	1189.4	882.86	0.01782	4.015
112.0	97.3	336.12	307.23	1189.5	882.27	0.017828	3.981
113.0	98.3	336.77	307.92	1189.7	881.78	0.017835	3.9476
114.0	99.3	337.43	308.61	1189.8	881.19	0.017842	3.9147
115.0	100.3	338.08	309.29	1190	880.71	0.01785	3.8824
116.0	101.3	338.72	309.96	1190.2	880.24	0.017857	3.8506
117.0	102.3	339.37	310.63	1190.3	879.67	0.017864	3.8194
118.0	103.3	340	311.3	1190.5	879.2	0.017871	3.7886
119.0	104.3	340.64	311.96	1190.6	878.64	0.017879	3.7584
120.0	105.3	341.26	312.62	1190.7	878.08	0.017886	3.7286
121.0	106.3	341.89	313.27	1190.9	877.63	0.017893	3.6993
122.0	107.3	342.51	313.92	1191	877.08	0.0179	3.6705
123.0	108.3	343.12	314.57	1191.2	876.63	0.017907	3.6422
124.0	109.3	343.74	315.21	1191.3	876.09	0.017914	3.6142
125.0	110.3	344.35	315.85	1191.5	875.65	0.017921	3.5867
126.0	111.3	344.95	316.48	1191.6	875.12	0.017928	3.5597
127.0	112.3	345.55	317.12	1191.7	874.58	0.017935	3.533
128.0	113.3	346.15	317.74	1191.9	874.16	0.017942	3.5067
129.0	114.3	346.74	318.37	1192	873.63	0.017949	3.4809
130.0	115.3	347.33	318.98	1192.1	873.12	0.017956	3.4554
131.0	116.3	347.92	319.6	1192.3	872.7	0.017963	3.4303
132.0	117.3	348.5	320.21	1192.4	872.19	0.01797	3.4055
133.0	118.3	349.08	320.82	1192.5	871.68	0.017976	3.3812
134.0	119.3	349.65	321.43	1192.6	871.17	0.017983	3.3571
135.0	120.3	350.23	322.03	1192.8	870.77	0.01799	3.3334
136.0	121.3	350.8	322.63	1192.9	870.27	0.017997	3.3101
137.0	122.3	351.36	323.22	1193	869.78	0.018003	3.2871
138.0	123.3	351.92	323.82	1193.1	869.28	0.01801	3.2644
139.0	124.3	352.48	324.4	1193.2	868.8	0.018017	3.242
140.0	125.3	353.04	324.99	1193.4	868.41	0.018023	3.2199
141.0	126.3	353.59	325.57	1193.5	867.93	0.01803	3.1981
142.0	127.3	354.14	326.15	1193.6	867.45	0.018037	3.1767
143.0	128.3	354.69	326.73	1193.7	866.97	0.018043	3.1555
144.0	129.3	355.23	327.3	1193.8	866.5	0.01805	3.1346
145.0	130.3	355.77	327.87	1193.9	866.03	0.018056	3.1139
146.0	131.3	356.31	328.44	1194.1	865.66	0.018063	3.0936
147.0	132.3	356.85	329.01	1194.2	865.19	0.018069	3.0735
148.0	133.3	357.38	329.57	1194.3	864.73	0.018076	3.0537
149.0	134.3	357.91	330.13	1194.4	864.27	0.018082	3.0341
150.0	135.3	358.44	330.68	1194.5	863.82	0.018089	3.0148
152.0	137.3	359.48	331.79	1194.7	862.91	0.018101	2.9769
154.0	139.3	360.51	332.88	1194.9	862.02	0.018114	2.9399
156.0	141.3	361.54	333.96	1195.1	861.14	0.018127	2.9039
158.0	143.3	362.55	335.03	1195.3	860.27	0.018139	2.8688
160.0	145.3	363.55	336.1	1195.5	859.4	0.018152	2.8345
162.0	147.3	364.55	337.15	1195.7	858.55	0.018164	2.801
164.0	149.3	365.53	338.19	1195.9	857.71	0.018176	2.7683
166.0	151.3	366.51	339.23	1196.1	856.87	0.018189	2.7363
168.0	153.3	367.47	340.25	1196.3	856.05	0.018201	2.7051
170.0	155.3	368.43	341.27	1196.5	855.23	0.018213	2.6746
172.0	157.3	369.38	342.27	1196.6	854.33	0.018225	2.6448
174.0	159.3	370.32	343.27	1196.8	853.53	0.018237	2.6157
176.0	161.3	371.25	344.26	1197	852.74	0.018249	2.5872
178.0	163.3	372.17	345.24	1197.1	851.86	0.018261	2.5593
180.0	165.3	373.08	346.21	1197.3	851.09	0.018272	2.532
182.0	167.3	373.99	347.18	1197.5	850.32	0.018284	2.5052
184.0	169.3	374.89	348.14	1197.6	849.46	0.018296	2.4791
186.0	171.3	375.78	349.09	1197.8	848.71	0.018307	2.4535
188.0	173.3	376.66	350.03	1197.9	847.87	0.018319	2.4284
190.0	175.3	377.54	350.96	1198.1	847.14	0.01833	2.4038
192.0	177.3	378.41	351.89	1198.2	846.31	0.018342	2.3797
194.0	179.3	379.27	352.81	1198.4	845.59	0.018353	2.3561

Properties of Saturated Steam and Saturated Water[33]

Pressure psi		Temperature °F	Specific Enthalpy Btu/lb		Specific Latent Heat of Evaporation	Specific Volume ft³/lb	
P'	Gauge P	t_s	h_f	h_g	Btu/lb h_{fg}	$\overline{V}_f$	$\overline{V}_g$
196.0	181.3	380.12	353.72	1198.5	844.78	0.018364	2.333
198.0	183.3	380.97	354.63	1198.7	844.07	0.018376	2.3103
200.0	185.3	381.81	355.53	1198.8	843.27	0.018387	2.288
205.0	190.3	383.89	357.75	1199.1	841.35	0.018415	2.2342
210.0	195.3	385.92	359.94	1199.5	839.56	0.018442	2.1829
215.0	200.3	387.92	362.08	1199.8	837.72	0.018469	2.1339
220.0	205.3	389.89	364.19	1200.1	835.91	0.018496	2.087
225.0	210.3	391.81	366.26	1200.3	834.04	0.018523	2.0422
230.0	215.3	393.71	368.3	1200.6	832.3	0.018549	1.9992
235.0	220.3	395.57	370.31	1200.9	830.59	0.018576	1.958
240.0	225.3	397.41	372.29	1201.1	828.81	0.018601	1.9184
245.0	230.3	399.21	374.24	1201.4	827.16	0.018627	1.8804
250.0	235.3	400.98	376.16	1201.6	825.44	0.018653	1.8439
255.0	240.3	402.73	378.05	1201.8	823.75	0.018678	1.8087
260.0	245.3	404.45	379.92	1202	822.08	0.018703	1.7749
265.0	250.3	406.15	381.76	1202.2	820.44	0.018728	1.7423
270.0	255.3	407.82	383.58	1202.4	818.82	0.018753	1.7108
275.0	260.3	409.46	385.37	1202.6	817.23	0.018777	1.6805
280.0	265.3	411.09	387.14	1202.8	815.66	0.018801	1.6512
285.0	270.3	412.69	388.89	1202.9	814.01	0.018825	1.6229
290.0	275.3	414.27	390.61	1203.1	812.49	0.018849	1.5955
295.0	280.3	415.83	392.32	1203.2	810.88	0.018873	1.5691
300.0	285.3	417.37	394	1203.4	809.4	0.018897	1.5434
320.0	305.3	423.33	400.54	1203.9	803.36	0.01899	1.4487
340.0	325.3	429.01	406.81	1204.3	797.49	0.019081	1.3647
360.0	345.3	434.43	412.82	1204.6	791.78	0.01917	1.2898
380.0	365.3	439.63	418.6	1204.9	786.3	0.019257	1.2224
400.0	385.3	444.63	424.18	1205	780.82	0.019343	1.1616
420.0	405.3	449.43	429.56	1205.1	775.54	0.019427	1.1064
440.0	425.3	454.06	434.78	1205.2	770.42	0.01951	1.056
460.0	445.3	458.53	439.84	1205.2	765.36	0.019592	1.0098
480.0	465.3	462.86	444.75	1205.1	760.35	0.019672	0.96733
500.0	485.3	467.05	449.53	1205	755.47	0.019752	0.92815
520.0	505.3	471.11	454.19	1204.9	750.71	0.019831	0.89188
540.0	525.3	475.05	458.72	1204.7	745.98	0.019909	0.8582
560.0	545.3	478.89	463.15	1204.4	741.25	0.019987	0.82684
580.0	565.3	482.62	467.48	1204.2	736.72	0.020064	0.79756
600.0	585.3	486.25	471.71	1203.9	732.19	0.02014	0.77016
620.0	605.3	489.79	475.85	1203.5	727.65	0.020216	0.74447
640.0	625.3	493.24	479.91	1203.2	723.29	0.020291	0.72032
660.0	645.3	496.62	483.88	1202.8	718.92	0.020365	0.69758
680.0	665.3	499.91	487.79	1202.4	714.61	0.02044	0.67614
700.0	685.3	503.14	491.62	1201.9	710.28	0.020514	0.65587
720.0	705.3	506.29	495.38	1201.4	706.02	0.020587	0.63668
740.0	725.3	509.38	499.08	1200.9	701.82	0.02066	0.61849
760.0	745.3	512.4	502.72	1200.4	697.68	0.020734	0.60122
780.0	765.3	515.36	506.3	1199.9	693.6	0.020806	0.58481
800.0	785.3	518.27	509.83	1199.3	689.47	0.020879	0.56918
820.0	805.3	521.12	513.3	1198.7	685.4	0.020951	0.55428
840.0	825.3	523.92	516.73	1198.1	681.37	0.021024	0.54007
860.0	845.3	526.67	520.1	1197.5	677.4	0.021096	0.52649
880.0	865.3	529.37	523.43	1196.8	673.37	0.021168	0.5135
900.0	885.3	532.02	526.72	1196.2	669.48	0.02124	0.50107
920.0	905.3	534.63	529.96	1195.5	665.54	0.021312	0.48915
940.0	925.3	537.2	533.17	1194.8	661.63	0.021384	0.47772
960.0	945.3	539.72	536.34	1194.1	657.76	0.021456	0.46675
980.0	965.3	542.21	539.47	1193.3	653.83	0.021528	0.4562
1000.0	985.3	544.65	542.56	1192.6	650.04	0.0216	0.44606
1050.0	1035.3	550.61	550.15	1190.6	640.45	0.02178	0.42232
1100.0	1085.3	556.35	557.55	1188.6	631.05	0.021961	0.40065
1150.0	1135.3	561.9	564.77	1186.4	621.63	0.022143	0.38077
1200.0	1185.3	567.26	571.84	1184.2	612.36	0.022326	0.36247
1250.0	1235.3	572.46	578.76	1181.9	603.14	0.02251	0.34555
1300.0	1285.3	577.5	585.55	1179.5	593.95	0.022696	0.32987
1350.0	1335.3	582.39	592.21	1177	584.79	0.022884	0.31529
1400.0	1385.3	587.14	598.77	1174.4	575.63	0.023075	0.30169
1450.0	1435.3	591.76	605.23	1171.8	566.57	0.023267	0.28896
1500.0	1485.3	596.27	611.59	1169	557.41	0.023463	0.27703
1600.0	1585.3	604.93	624.07	1163.3	539.23	0.023863	0.25525
1700.0	1685.3	613.19	636.28	1157.2	520.92	0.024277	0.23584

Properties of Saturated Steam and Saturated Water[33] - concluded

Pressure psi		Temperature °F	Specific Enthalpy Btu/lb		Specific Latent Heat of Evaporation	Specific Volume ft³/lb	
P'	Gauge P	t_s	h_f	h_g	Btu/lb h_{fg}	$\bar{V}_f$	$\bar{V}_g$
1800.0	1785.3	621.07	648.27	1150.7	502.43	0.024708	0.2184
1900.0	1885.3	628.62	660.09	1143.8	483.71	0.02516	0.2026
2000.0	1985.3	635.85	671.8	1136.5	464.7	0.025635	0.18819
2100.0	2085.3	642.81	683.44	1128.7	445.26	0.026138	0.17496
2200.0	2185.3	649.5	695.09	1120.4	425.31	0.026677	0.16273
2300.0	2285.3	655.94	706.8	1111.5	404.7	0.027258	0.15135
2400.0	2385.3	662.16	718.67	1101.9	383.23	0.027892	0.1407
2500.0	2485.3	668.17	730.78	1091.6	360.82	0.028591	0.13068
2600.0	2585.3	673.98	743.27	1080.2	336.93	0.029376	0.12111
2700.0	2685.3	679.6	756.32	1067.6	311.28	0.030276	0.11191
2800.0	2785.3	685.03	770.2	1053.4	283.2	0.031337	0.10291
2900.0	2885.3	690.3	785.39	1036.8	251.41	0.032645	0.093909
3000.0	2985.3	695.41	802.9	1016.5	213.6	0.03438	0.084525
3100.0	3085.3	700.35	825.57	988.14	162.57	0.037079	0.07381
3200.0	3185.3	705.1	893.86	901.05	7.19	0.048974	0.050519
3200.11	3185.4	705.1	896.96	897.83	0.87	0.049635	0.049821

Uses conversion of 1 psi = 2.03602 in of Hg. Gauge pressure based on a 14.696 psia reference (EL 0 ft).

Properties of Superheated Steam[33]

$\bar{V}$ = specific volume, ft³/lb

h_g = total heat of steam, Btu/lb

Pressure psi		Sat Temp t_s °F		Total Temperature - °F (t)										
Abs. P'	Gauge P			350	400	500	600	700	800	900	1000	1100	1300	1500
15	0.304	212.99	$\bar{V}$	31.943	33.966	37.986	41.988	45.981	49.968	53.95	57.93	61.909	69.861	77.811
			h_g	1216.3	1239.9	1287.3	1335.3	1383.9	1433.3	1483.6	1534.7	1586.8	1693.7	1804.3
20	5.304	227.92	$\bar{V}$	23.903	25.43	28.458	31.467	34.467	37.461	40.451	43.438	46.424	52.391	58.355
			h_g	1215.5	1239.3	1286.9	1334.9	1383.6	1433.1	1483.4	1534.6	1586.7	1693.6	1804.2
30	15.304	250.3	$\bar{V}$	15.863	16.894	18.93	20.947	22.954	24.955	26.952	28.946	30.939	34.92	38.899
			h_g	1213.8	1237.9	1286	1334.3	1383.1	1432.7	1483.1	1534.3	1586.4	1693.4	1804.1
40	25.304	267.22	$\bar{V}$	11.841	12.625	14.165	15.686	17.197	18.702	20.202	21.7	23.197	26.185	29.171
			h_g	1212	1236.5	1285	1333.6	1382.6	1432.3	1482.7	1534	1586.2	1693.2	1803.9
50	35.304	280.99	$\bar{V}$	9.4273	10.063	11.306	12.53	13.743	14.95	16.153	17.353	18.551	20.944	23.334
			h_g	1210.2	1235.1	1284.1	1332.9	1382.1	1431.9	1482.4	1533.8	1586	1693.1	1803.8
60	45.304	292.69	$\bar{V}$	7.8173	8.3549	9.4004	10.425	11.44	12.448	13.453	14.454	15.454	17.45	19.442
			h_g	1208.4	1233.7	1283.2	1332.2	1381.5	1431.4	1482.1	1533.5	1585.7	1692.9	1803.7
70	55.304	302.92	$\bar{V}$	6.6666	7.1344	8.0389	8.9223	9.795	10.662	11.524	12.384	13.242	14.954	16.663
			h_g	1206.5	1232.3	1282.2	1331.5	1381	1431	1481.7	1533.2	1585.5	1692.7	1803.6
80	65.304	312.03	$\bar{V}$	5.803	6.2186	7.0176	7.7949	8.5614	9.3216	10.078	10.831	11.583	13.082	14.578
			h_g	1204.5	1230.8	1281.3	1330.8	1380.5	1430.6	1481.4	1532.9	1585.3	1692.6	1803.4
90	75.304	320.27	$\bar{V}$	5.1307	5.5061	6.2232	6.918	7.6018	8.2794	8.9529	9.6237	10.293	11.626	12.957
			h_g	1202.5	1229.3	1280.3	1330.1	1380	1430.2	1481	1532.6	1585	1692.4	1803.3
100	85.304	327.82	$\bar{V}$	4.5923	4.9358	5.5875	6.2165	6.8342	7.4456	8.0529	8.6576	9.2602	10.462	11.66
			h_g	1200.4	1227.8	1279.3	1329.5	1379.4	1429.8	1480.7	1532.3	1584.8	1692.2	1803.2
120	105.304	341.26	$\bar{V}$	3.7832	4.0796	4.6339	5.164	5.6827	6.1949	6.703	7.2083	7.7117	8.7147	9.7144
			h_g	1196.1	1224.6	1277.3	1328.1	1378.4	1428.9	1480	1531.8	1584.3	1691.9	1802.9
140	125.304	353.04	$\bar{V}$	...	3.4673	3.9524	4.4122	4.8602	5.3015	5.7387	6.1732	6.6057	7.4668	8.3247
			h_g	...	1221.4	1275.3	1326.6	1377.3	1428.1	1479.3	1531.2	1583.8	1691.5	1802.7
160	145.304	363.55	$\bar{V}$	...	3.0073	3.441	3.8483	4.2432	4.6314	5.0155	5.3968	5.7761	6.5309	7.2824
			h_g	...	1218.0	1273.3	1325.2	1376.2	1427.2	1478.7	1530.7	1583.4	1691.2	1802.4
180	165.304	373.08	$\bar{V}$	...	2.6487	3.0431	3.4095	3.7633	4.1103	4.453	4.7929	5.1309	5.8029	6.4717
			h_g	...	1214.5	1271.2	1323.8	1375.1	1426.4	1478	1530.1	1582.9	1690.8	1802.1
200	185.304	381.81	$\bar{V}$	...	2.3612	2.7246	3.0585	3.3794	3.6933	4.003	4.3098	4.6147	5.2206	5.8231
			h_g	...	1210.9	1269.1	1322.3	1374.1	1425.5	1477.3	1529.5	1582.4	1690.5	1801.9
220	205.304	389.89	$\bar{V}$	...	2.1252	2.4638	2.7712	3.0652	3.3521	3.6348	3.9146	4.1924	4.7441	5.2925
			h_g	...	1207.0	1266.9	1320.8	1373	1424.7	1476.6	1529	1581.9	1690.2	1801.6
240	225.304	397.41	$\bar{V}$	...	1.9277	2.2462	2.5317	2.8034	3.0678	3.3279	3.5852	3.8404	4.347	4.8503
			h_g	...	1203.0	1264.7	1319.4	1371.9	1423.8	1475.9	1528.4	1581.5	1689.8	1801.4
260	245.304	404.45	$\bar{V}$	...	...	2.062	2.329	2.5818	2.8272	3.0683	3.3065	3.5426	4.0111	4.4762
			h_g	...	...	1262.5	1317.9	1370.8	1423	1475.2	1527.8	1581	1689.5	1801.1
280	265.304	411.09	$\bar{V}$	...	...	1.9039	2.1552	2.3919	2.621	2.8457	3.0676	3.2874	3.7231	4.1555
			h_g	...	...	1260.2	1316.3	1369.7	1422.1	1474.5	1527.2	1580.5	1689.1	1800.8
300	285.304	417.37	$\bar{V}$	...	...	1.7668	2.0045	2.2272	2.4423	2.6528	2.8605	3.0662	3.4735	3.8775
			h_g	...	...	1257.9	1314.8	1368.5	1421.2	1473.8	1526.7	1580	1688.8	1800.6
320	305.304	423.33	$\bar{V}$	...	...	1.6467	1.8726	2.0831	2.2859	2.4841	2.6793	2.8726	3.2551	3.6343
			h_g	...	...	1255.5	1313.3	1367.4	1420.4	1473.1	1526.1	1579.6	1688.4	1800.3
340	325.304	429.01	$\bar{V}$	...	...	1.5405	1.7562	1.956	2.1478	2.3351	2.5195	2.7018	3.0624	3.4197
			h_g	...	...	1253.1	1311.7	1366.3	1419.5	1472.4	1525.5	1579.1	1688.1	1800.1
360	345.304	434.43	$\bar{V}$	...	...	1.446	1.6526	1.8429	2.0252	2.2028	2.3774	2.55	2.8911	3.229
			h_g	...	...	1250.6	1310.1	1365.2	1418.6	1471.7	1525	1578.6	1687.7	1799.8

Uses conversion of 1 psi = 2.03602 in of Hg. Gauge pressure based on a 14.696 psia reference (EL 0 ft).

CRANE

Properties of Superheated Steam[33]

$\bar{V}$ = specific volume, ft³/lb

h_g = total heat of steam, Btu/lb

Pressure psi		Sat Temp t_s °F		Total Temperature - °F (t)										
Abs. P'	Gauge P			500	600	700	800	900	1000	1100	1200	1300	1400	1500
380	365.304	439.63	$\bar{V}$	1.3613	1.56	1.7418	1.9154	2.0843	2.2502	2.4141	2.5765	2.7379	2.8984	3.0583
			h_g	1248.1	1308.5	1364	1417.8	1471	1524.4	1578.1	1632.4	1687.4	1743.1	1799.5
400	385.304	444.63	$\bar{V}$	1.285	1.4765	1.6507	1.8166	1.9777	2.1358	2.2919	2.4464	2.6	2.7527	2.9047
			h_g	1245.6	1306.9	1362.9	1416.9	1470.3	1523.8	1577.7	1632	1687	1742.8	1799.3
420	405.304	449.43	$\bar{V}$	1.2157	1.4009	1.5683	1.7271	1.8812	2.0323	2.1812	2.3287	2.4752	2.6208	2.7658
			h_g	1242.9	1305.3	1361.7	1416	1469.6	1523.2	1577.2	1631.6	1686.7	1742.5	1799
440	425.304	454.06	$\bar{V}$	1.1527	1.3322	1.4933	1.6458	1.7935	1.9381	2.0807	2.2217	2.3617	2.5009	2.6394
			h_g	1240.3	1303.7	1360.5	1415.1	1468.9	1522.7	1576.7	1631.2	1686.4	1742.2	1798.8
460	445.304	458.53	$\bar{V}$	1.0950	1.2694	1.4249	1.5716	1.7134	1.8522	1.9888	2.124	2.2581	2.3914	2.5241
			h_g	1237.6	1302	1359.4	1414.2	1468.2	1522.1	1576.2	1630.8	1686	1741.9	1798.5
480	465.304	462.86	$\bar{V}$	1.0419	1.2118	1.3621	1.5036	1.64	1.7734	1.9047	2.0345	2.1632	2.2911	2.4183
			h_g	1234.8	1300.3	1358.2	1413.3	1467.5	1521.5	1575.7	1630.4	1685.7	1741.6	1798.2
500	485.304	467.05	$\bar{V}$	0.99299	1.1587	1.3044	1.4409	1.5725	1.7009	1.8272	1.9521	2.0758	2.1987	2.321
			h_g	1231.9	1298.6	1357	1412.4	1466.8	1520.9	1575.3	1630	1685.3	1741.3	1798
520	505.304	471.11	$\bar{V}$	0.94768	1.1097	1.2511	1.3831	1.5101	1.634	1.7558	1.876	1.9952	2.1135	2.2313
			h_g	1229.0	1296.9	1355.8	1411.5	1466.1	1520.3	1574.8	1629.6	1685	1741	1797.7
540	525.304	475.05	$\bar{V}$	0.90558	1.0643	1.2017	1.3296	1.4524	1.572	1.6896	1.8056	1.9205	2.0346	2.1481
			h_g	1226.0	1295.2	1354.6	1410.6	1465.4	1519.8	1574.3	1629.2	1684.6	1740.7	1797.5
560	545.304	478.89	$\bar{V}$	0.86634	1.0221	1.1558	1.2799	1.3988	1.5145	1.6281	1.7402	1.8512	1.9614	2.0709
			h_g	1223.0	1293.4	1353.4	1409.7	1464.7	1519.2	1573.8	1628.8	1684.3	1740.4	1797.2
580	565.304	482.62	$\bar{V}$	0.82966	0.98279	1.1131	1.2336	1.3489	1.4609	1.5709	1.6793	1.7866	1.8931	1.999
			h_g	1219.8	1291.7	1352.2	1408.8	1463.9	1518.6	1573.3	1628.4	1683.9	1740.1	1796.9
600	585.304	486.25	$\bar{V}$	0.79526	0.94604	1.0732	1.1904	1.3023	1.411	1.5175	1.6225	1.7264	1.8295	1.9319
			h_g	1216.5	1289.9	1351	1407.9	1463.2	1518	1572.8	1628	1683.6	1739.8	1796.7
650	635.304	494.94	$\bar{V}$	0.71782	0.86392	0.98413	1.094	1.1983	1.2994	1.3983	1.4957	1.592	1.6874	1.7823
			h_g	1208.0	1285.3	1347.9	1405.6	1461.4	1516.6	1571.6	1626.9	1682.7	1739	1796
700	685.304	503.14	$\bar{V}$	...	0.79333	0.90773	1.0113	1.1092	1.2038	1.2962	1.3871	1.4768	1.5657	1.654
			h_g	...	1280.6	1344.8	1403.3	1459.6	1515.1	1570.4	1625.9	1681.8	1738.3	1795.4
750	735.304	510.9	$\bar{V}$	...	0.73193	0.84143	0.93958	1.032	1.1209	1.2077	1.2929	1.377	1.4602	1.5428
			h_g	...	1275.8	1341.6	1401	1457.8	1513.6	1569.2	1624.9	1681	1737.5	1794.7
800	785.304	518.27	$\bar{V}$	...	0.67801	0.78335	0.87682	0.96435	1.0484	1.1302	1.2105	1.2896	1.3679	1.4455
			h_g	...	1270.8	1338.4	1398.6	1456	1512.1	1568	1623.9	1680.1	1736.8	1794
850	835.304	525.3	$\bar{V}$	...	0.63022	0.73204	0.82141	0.90468	0.98441	1.0619	1.1378	1.2125	1.2864	1.3597
			h_g	...	1265.7	1335.1	1396.2	1454.1	1510.7	1566.7	1622.8	1679.2	1736	1793.4
900	885.304	532.02	$\bar{V}$	...	0.58754	0.68635	0.77212	0.85162	0.92752	1.0011	1.0731	1.144	1.214	1.2834
			h_g	...	1260.4	1331.8	1393.8	1452.3	1509.2	1565.5	1621.8	1678.3	1735.3	1792.7
950	935.304	538.46	$\bar{V}$	...	0.54913	0.64541	0.728	0.80414	0.87661	0.94675	1.0153	1.0827	1.1492	1.2151
			h_g	...	1254.9	1328.4	1391.4	1450.4	1507.7	1564.3	1620.8	1677.4	1734.5	1792.1
1000	985.304	544.65	$\bar{V}$	...	0.51435	0.6085	0.68826	0.76139	0.83078	0.89781	0.96323	1.0275	1.0909	1.1537
			h_g	...	1249.3	1324.9	1389	1448.5	1506.2	1563	1619.7	1676.6	1733.7	1791.4
1050	1035.304	550.61	$\bar{V}$	...	0.48265	0.57504	0.65228	0.7227	0.78931	0.85354	0.91614	0.97758	1.0382	1.0981
			h_g	...	1243.4	1321.4	1386.5	1446.6	1504.7	1561.8	1618.7	1675.7	1733	1790.8
1100	1085.304	556.35	$\bar{V}$	...	0.45361	0.54455	0.61955	0.68752	0.75161	0.81329	0.87332	0.9322	0.99021	1.0476
			h_g	...	1237.2	1317.9	1384	1444.7	1503.2	1560.6	1617.7	1674.8	1732.2	1790.1
1150	1135.304	561.9	$\bar{V}$	...	0.42684	0.51666	0.58964	0.65538	0.71718	0.77653	0.83423	0.89076	0.94643	1.0014
			h_g	...	1230.8	1314.2	1381.5	1442.8	1501.6	1559.3	1616.6	1673.9	1731.5	1789.4

Uses conversion of 1 psi = 2.03602 in of Hg. Gauge pressure based on a 14.696 psia reference (EL 0 ft).

Properties of Superheated Steam[33] - concluded

$\bar{V}$ = specific volume, ft³/lb
h_g = total heat of steam, Btu/lb

Pressure psi		Sat Temp t_s °F		Total Temperature - °F (t)										
Abs. P'	Gauge P			650	700	750	800	900	1000	1100	1200	1300	1400	1500
1200	1185.304	567.26	$\bar{V}$	0.45014	0.49103	0.5279	0.5622	0.62592	0.68562	0.74284	0.79839	0.85277	0.90629	0.95915
			h_g	1271.1	1310.5	1345.9	1378.9	1440.9	1500.1	1558.1	1615.6	1673	1730.7	1788.8
1300	1285.304	577.5	$\bar{V}$	0.40566	0.44551	0.48093	0.51358	0.57375	0.62975	0.68321	0.73499	0.78557	0.83527	0.88433
			h_g	1261.3	1302.9	1339.7	1373.7	1437	1497	1555.6	1613.5	1671.3	1729.2	1787.4
1400	1385.304	587.14	$\bar{V}$	0.36707	0.40627	0.44053	0.47182	0.529	0.58185	0.6321	0.68063	0.72796	0.77441	0.82019
			h_g	1250.8	1295	1333.4	1368.5	1433.1	1493.9	1553.1	1611.4	1669.5	1727.7	1786.1
1500	1485.304	596.27	$\bar{V}$	0.33314	0.37202	0.40539	0.43555	0.49019	0.54033	0.5878	0.63352	0.67803	0.72165	0.76461
			h_g	1239.6	1286.8	1326.9	1363.1	1429.1	1490.8	1550.5	1609.3	1667.7	1726.1	1784.8
1600	1585.304	604.93	$\bar{V}$	0.30291	0.34182	0.37451	0.40374	0.45619	0.50397	0.54903	0.5923	0.63435	0.6755	0.71598
			h_g	1227.7	1278.3	1320.2	1357.6	1425.1	1487.7	1548	1607.2	1665.9	1724.6	1783.5
1700	1685.304	613.19	$\bar{V}$	0.27565	0.31493	0.34714	0.37559	0.42616	0.47188	0.51481	0.55593	0.5958	0.63477	0.67307
			h_g	1214.7	1269.3	1313.3	1351.9	1421	1484.5	1545.4	1605	1664.1	1723	1782.1
1800	1785.304	621.07	$\bar{V}$	0.25075	0.29078	0.32268	0.3505	0.39944	0.44335	0.48439	0.52359	0.56154	0.59857	0.63493
			h_g	1200.6	1259.9	1306.2	1346.2	1416.9	1481.3	1542.8	1602.9	1662.3	1721.5	1780.8
1900	1885.304	628.62	$\bar{V}$	0.22766	0.26891	0.30068	0.32798	0.3755	0.4178	0.45716	0.49466	0.53088	0.56618	0.60081
			h_g	1185.1	1250.1	1298.8	1340.3	1412.7	1478.1	1540.2	1600.8	1660.5	1720	1779.5
2000	1985.304	635.85	$\bar{V}$	0.20587	0.24895	0.28075	0.30764	0.35393	0.3948	0.43266	0.46862	0.50329	0.53703	0.5701
			h_g	1167.5	1239.7	1291.2	1334.3	1408.5	1474.9	1537.6	1598.6	1658.7	1718.4	1778.1
2100	2085.304	642.81	$\bar{V}$	0.18481	0.2306	0.26259	0.28918	0.33439	0.37398	0.41048	0.44505	0.47832	0.51066	0.54231
			h_g	1147.2	1228.7	1283.4	1328.1	1404.2	1471.6	1535	1596.5	1656.9	1716.9	1776.8
2200	2185.304	649.5	$\bar{V}$	0.16352	0.21359	0.24595	0.27232	0.3166	0.35504	0.39031	0.42363	0.45563	0.48669	0.51705
			h_g	1122.0	1217.1	1275.2	1321.8	1399.8	1468.3	1532.4	1594.3	1655.1	1715.3	1775.5
2300	2285.304	655.94	$\bar{V}$	...	0.19770	0.23062	0.25687	0.30034	0.33774	0.3719	0.40407	0.43491	0.4648	0.494
			h_g	...	1204.5	1266.8	1315.4	1395.4	1465	1529.8	1592.1	1653.2	1713.8	1774.1
2400	2385.304	662.16	$\bar{V}$	...	0.18272	0.21644	0.24264	0.28541	0.32187	0.35502	0.38614	0.41592	0.44474	0.47286
			h_g	...	1191.0	1258	1308.8	1390.9	1461.6	1527.1	1589.9	1651.4	1712.2	1772.8
2500	2485.304	668.17	$\bar{V}$	...	0.16847	0.20326	0.22948	0.27165	0.30726	0.33948	0.36965	0.39844	0.42628	0.45341
			h_g	...	1176.3	1248.9	1302	1386.4	1458.2	1524.4	1587.8	1649.6	1710.7	1771.4
2600	2585.304	673.98	$\bar{V}$	...	0.15475	0.19094	0.21727	0.25894	0.29377	0.32514	0.35442	0.38232	0.40925	0.43547
			h_g	...	1160.0	1239.4	1295	1381.8	1454.8	1521.7	1585.6	1647.8	1709.1	1770.1
2700	2685.304	679.6	$\bar{V}$	...	0.14135	0.17939	0.2059	0.24714	0.28128	0.31186	0.34032	0.36738	0.39347	0.41885
			h_g	...	1141.7	1229.4	1287.8	1377.2	1451.4	1519.1	1583.4	1645.9	1707.5	1768.7
2800	2785.304	685.03	$\bar{V}$	...	0.12796	0.1685	0.19528	0.23618	0.26966	0.29952	0.32723	0.35352	0.37883	0.40342
			h_g	...	1120.6	1219	1280.5	1372.5	1447.9	1516.3	1581.2	1644.1	1706	1767.4
2900	2885.304	690.3	$\bar{V}$	...	0.11407	0.15819	0.18533	0.22595	0.25885	0.28803	0.31504	0.34061	0.3652	0.38906
			h_g	...	1094.8	1208	1272.9	1367.7	1444.5	1513.6	1579	1642.2	1704.4	1766
3000	2985.304	695.41	$\bar{V}$	...	0.098359	0.14839	0.17598	0.21638	0.24875	0.27731	0.30366	0.32857	0.35247	0.37566
			h_g	...	1059.8	1196.4	1265.2	1362.9	1441	1510.9	1576.7	1640.4	1702.8	1764.7
3100	3085.304	700.35	$\bar{V}$	...	...	0.13902	0.16718	0.20742	0.2393	0.26728	0.29302	0.3173	0.34057	0.36312
			h_g	...	...	1184.2	1257.2	1358	1437.4	1508.1	1574.5	1638.5	1701.3	1763.4
3200	3185.304	705.1	$\bar{V}$	...	...	0.13002	0.15885	0.19901	0.23043	0.25788	0.28304	0.30674	0.32942	0.35137
			h_g	...	...	1171.0	1248.9	1353	1433.9	1505.4	1572.3	1636.7	1699.7	1762
3300	3285.304		$\bar{V}$	...	...	0.12132	0.15097	0.19109	0.2221	0.24904	0.27367	0.29681	0.31894	0.34033
			h_g	...	...	1157	1240.4	1348	1430.3	1502.6	1570	1634.8	1698.1	1760.7
3400	3385.304		$\bar{V}$	...	...	0.11287	0.14349	0.18362	0.21425	0.24072	0.26485	0.28748	0.30908	0.32994
			h_g	...	...	1141.8	1231.6	1342.9	1426.6	1499.8	1567.8	1633	1696.6	1759.3

Uses conversion of 1 psi = 2.03602 in of Hg. Gauge pressure based on a 14.696 psia reference (EL 0 ft).

Properties of Superheated Steam and Compressed Water[33]

$\bar{V}$ = specific volume, ft³/lb

h_g = total heat of steam, Btu/lb

Absolute Pressure P' psi		Total Temperature - °F (t)												
		200	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500
3500	$\bar{V}$	0.01645	0.0183	0.0199	0.0225	0.0306	0.1364	0.1766	0.2069	0.2329	0.2565	0.2787	0.2998	0.3202
	h_g	176.05	379.12	487.54	608.3	779.86	1222.6	1337.7	1423	1497	1565.5	1631.1	1695	1758
3600	$\bar{V}$	0.016445	0.0183	0.0199	0.0225	0.0301	0.1296	0.1699	0.1999	0.2255	0.2487	0.2704	0.2910	0.3109
	h_g	176.27	379.25	487.56	608	775.67	1213.2	1332.5	1419.3	1494.2	1563.3	1629.2	1693.4	1756.6
3800	$\bar{V}$	0.016435	0.0183	0.0198	0.0224	0.0293	0.1169	0.1576	0.1870	0.2118	0.2342	0.2551	0.2748	0.2939
	h_g	176.73	379.52	487.61	607.43	768.95	1193.4	1321.9	1411.9	1488.5	1558.8	1625.5	1690.3	1753.9
4000	$\bar{V}$	0.016425	0.0182	0.0198	0.0223	0.0287	0.1052	0.1465	0.1754	0.1996	0.2212	0.2413	0.2603	0.2785
	h_g	177.19	379.79	487.66	606.89	763.62	1172.1	1310.9	1404.4	1482.8	1554.2	1621.7	1687.1	1751.2
4200	$\bar{V}$	0.016415	0.0182	0.0198	0.0222	0.0282	0.0944	0.1364	0.1649	0.1885	0.2095	0.2288	0.2471	0.2647
	h_g	177.65	380.07	487.73	606.38	759.21	1149.1	1299.7	1396.8	1477.1	1549.6	1618	1683.9	1748.5
4400	$\bar{V}$	0.016405	0.0182	0.0197	0.0222	0.0278	0.0844	0.1272	0.1553	0.1784	0.1988	0.2175	0.2352	0.2521
	h_g	178.1	380.35	487.79	605.9	755.43	1124.3	1288.2	1389.1	1471.3	1545	1614.2	1680.8	1745.8
4600	$\bar{V}$	0.016395	0.0182	0.0197	0.0221	0.0274	0.0751	0.1188	0.1466	0.1692	0.1890	0.2072	0.2243	0.2406
	h_g	178.56	380.63	487.87	605.46	752.13	1097.6	1276.4	1381.4	1465.5	1540.4	1610.4	1677.6	1743.1
4800	$\bar{V}$	0.016385	0.0182	0.0197	0.0220	0.0271	0.0667	0.1110	0.1386	0.1607	0.1801	0.1977	0.2143	0.2301
	h_g	179.02	380.91	487.95	605.03	749.21	1069.9	1264.4	1373.5	1459.7	1535.8	1606.7	1674.4	1740.4
5200	$\bar{V}$	0.016366	0.0181	0.0196	0.0219	0.0265	0.0533	0.0974	0.1245	0.1458	0.1643	0.1810	0.1966	0.2115
	h_g	179.93	381.48	488.13	604.27	744.21	1015.3	1239.5	1357.5	1447.9	1526.5	1599.1	1668.1	1735
5600	$\bar{V}$	0.016346	0.0181	0.0195	0.0217	0.0260	0.0446	0.0858	0.1125	0.1331	0.1508	0.1667	0.1815	0.1955
	h_g	180.85	382.05	488.33	603.59	740.07	971.3	1213.9	1341.3	1436	1517.2	1591.5	1661.8	1729.6
6000	$\bar{V}$	0.016327	0.0181	0.0195	0.0216	0.0256	0.0395	0.0759	0.1021	0.1221	0.1391	0.1544	0.1684	0.1817
	h_g	181.76	382.64	488.55	602.99	736.54	940.83	1187.7	1324.8	1424	1507.9	1583.9	1655.5	1724.3
6500	$\bar{V}$	0.016304	0.0180	0.0194	0.0215	0.0252	0.0358	0.0656	0.0910	0.1104	0.1266	0.1411	0.1544	0.1669
	h_g	182.91	383.38	488.86	602.34	732.8	915.6	1155.3	1304.1	1409	1496.2	1574.5	1647.6	1717.6
7000	$\bar{V}$	0.01628	0.0180	0.0193	0.0213	0.0248	0.0334	0.0576	0.0817	0.1004	0.1160	0.1298	0.1424	0.1542
	h_g	184.05	384.13	489.2	601.8	729.62	898.43	1124.8	1283.4	1394	1484.6	1565.1	1639.8	1711
7500	$\bar{V}$	0.016257	0.0179	0.0193	0.0212	0.0245	0.0318	0.0513	0.0739	0.0918	0.1068	0.1200	0.1320	0.1433
	h_g	185.2	384.89	489.56	601.34	726.9	885.81	1097.1	1262.9	1379.1	1473	1555.7	1632	1704.5
8000	$\bar{V}$	0.016234	0.0179	0.0192	0.0211	0.0242	0.0306	0.0465	0.0672	0.0844	0.0988	0.1115	0.1230	0.1337
	h_g	186.34	385.65	489.95	600.96	724.53	876.02	1073.2	1243	1364.3	1461.5	1546.4	1624.3	1697.9
9000	$\bar{V}$	0.016189	0.0178	0.0191	0.0209	0.0237	0.0289	0.0401	0.0568	0.0725	0.0858	0.0975	0.1081	0.1179
	h_g	188.64	387.22	490.81	600.42	720.64	861.6	1036.2	1205.8	1335.6	1438.9	1528.1	1609.1	1685.1
10000	$\bar{V}$	0.016145	0.0177	0.0190	0.0207	0.0232	0.0276	0.0362	0.0495	0.0633	0.0756	0.0864	0.0962	0.1053
	h_g	190.93	388.82	491.75	600.13	717.62	851.32	1010.1	1173.6	1308.5	1417.2	1510.4	1594.3	1672.6
11000	$\bar{V}$	0.016101	0.0177	0.0188	0.0205	0.0229	0.0267	0.0336	0.0442	0.0563	0.0675	0.0776	0.0867	0.0952
	h_g	193.22	390.45	492.76	600.03	715.24	843.58	991.27	1146.8	1283.5	1396.5	1493.3	1579.9	1660.4
12000	$\bar{V}$	0.016059	0.0176	0.0187	0.0203	0.0225	0.0260	0.0317	0.0404	0.0509	0.0611	0.0704	0.0789	0.0868
	h_g	195.51	392.11	493.85	600.12	713.38	837.54	977.11	1125.1	1261.2	1377.1	1477	1566.1	1648.6
13000	$\bar{V}$	0.016017	0.0175	0.0186	0.0201	0.0222	0.0253	0.0303	0.0376	0.0467	0.0559	0.0645	0.0724	0.0798
	h_g	197.81	393.79	494.99	600.36	711.93	832.72	966.08	1107.5	1241.6	1359.2	1461.5	1552.9	1637.2
14000	$\bar{V}$	0.015977	0.0174	0.0185	0.0200	0.0219	0.0248	0.0291	0.0354	0.0433	0.0516	0.0596	0.0670	0.0739
	h_g	200.1	395.5	496.2	600.73	710.81	828.83	957.27	1093.1	1224.6	1342.8	1446.9	1540.3	1626.3
15000	$\bar{V}$	0.015936	0.0174	0.0184	0.0198	0.0217	0.0243	0.0282	0.0337	0.0406	0.0481	0.0554	0.0624	0.0689
	h_g	202.39	397.23	497.45	601.21	709.98	825.65	950.11	1081.2	1209.8	1327.9	1433.3	1528.4	1615.8

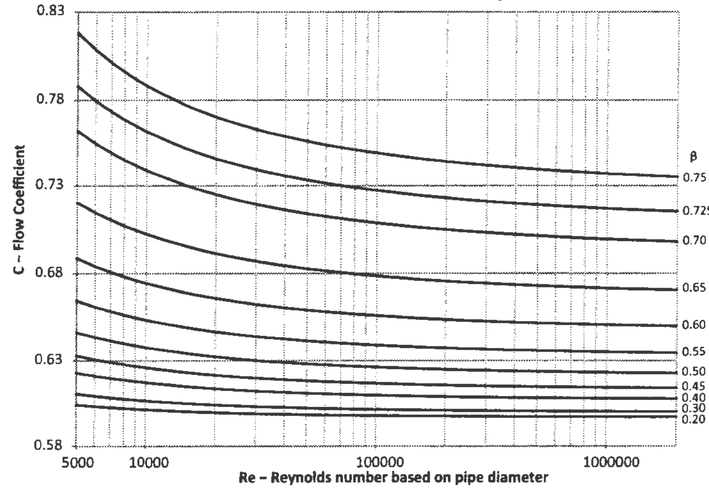
Flow Coefficient C for Square Edge Orifices and Nozzles[27]

$$C = \frac{C_d}{\sqrt{1 - \beta^4}}$$

$$K_{orifice} = \left[\frac{\sqrt{1 - \beta^4(1 - C_d^2)}}{C_d \beta^2} - 1 \right]^2$$

$$\beta = \frac{d_1}{d_2}$$

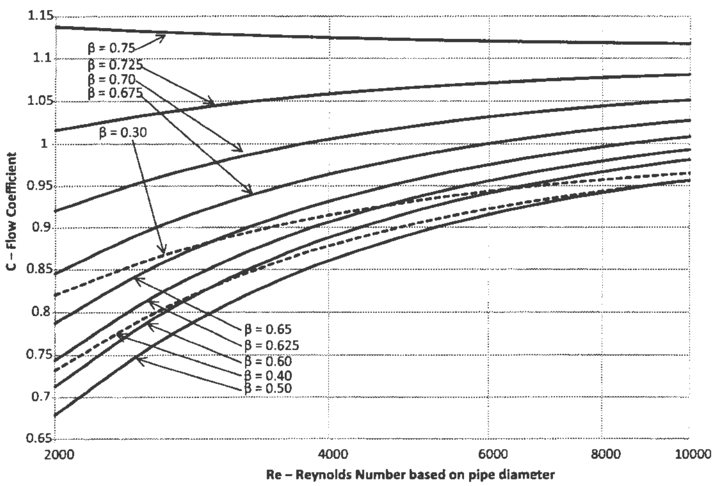
Flow Coefficient C for Square-Edge Orifices



Calculated from equations presented in ASME MFC-3Ma-2007

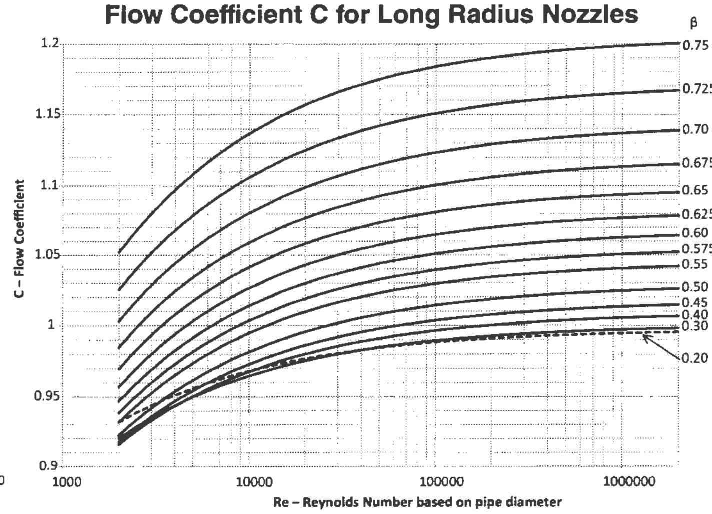
***Note:** Most accurate when pipe diameter is greater than 2.8 inches.*

Flow Coefficient C for ISA 1932 Nozzles



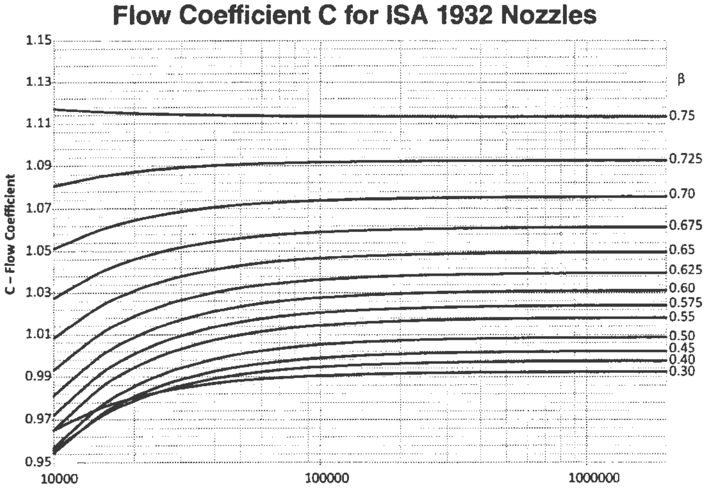
Calculated from equations presented in ASME MFC-3Ma-2007

Flow Coefficient C for Long Radius Nozzles



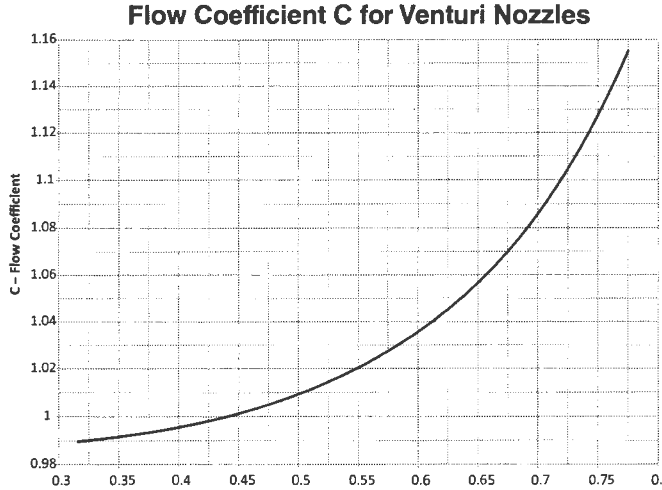
Calculated from equations presented in ASME MFC-3Ma-2007

Flow Coefficient C for ISA 1932 Nozzles



Calculated from equations presented in ASME MFC-3Ma-2007

Flow Coefficient C for Venturi Nozzles



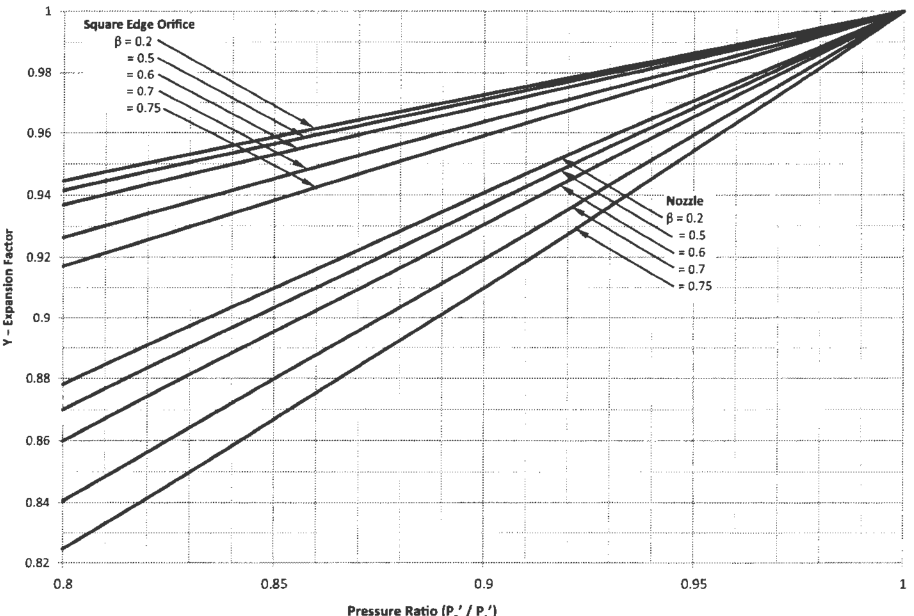
Note: *Flow coefficient is independent of Reynolds Number*

Calculated from equations presented in ASME MFC-3M-2004

www.

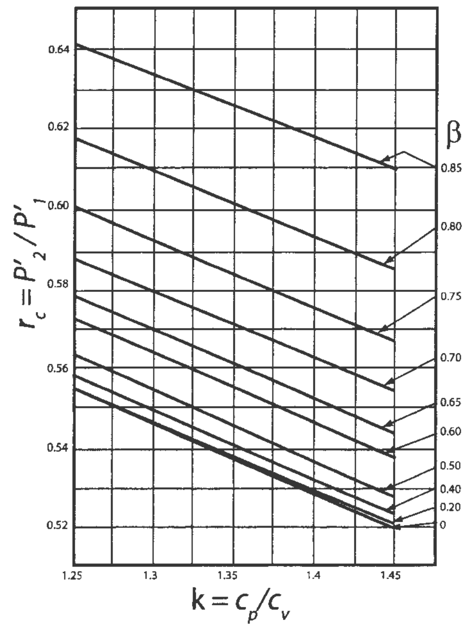
Net Expansion Factor, Y and Critical Pressure Ratio, r_c[27]

Net Expansion Factor, Y for k = 1.3

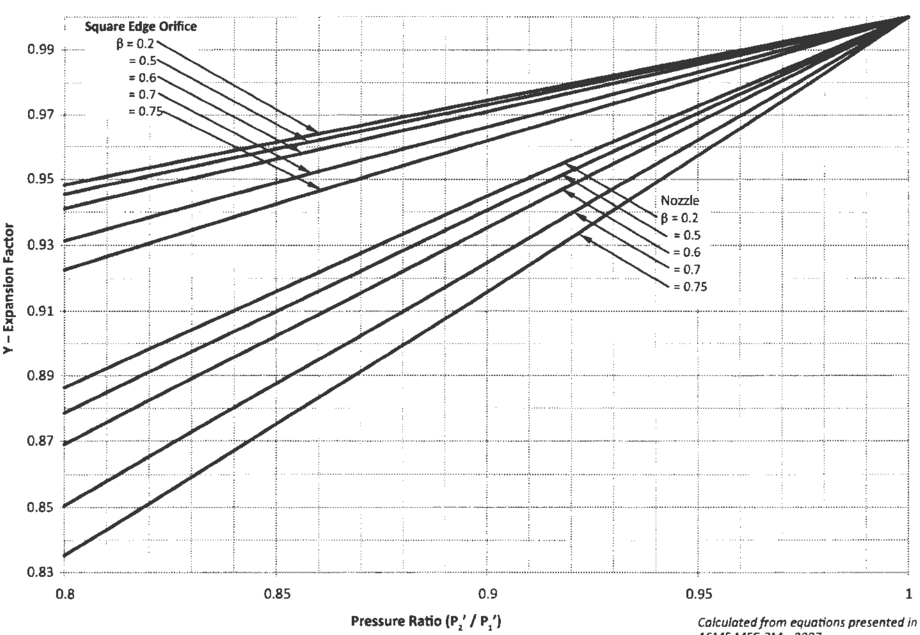


Calculated from equations presented in ASME MFC-3Ma-2007

Critical Pressure Ratio, r_c[46] For Compressible Flow Through Nozzles and Venturi Tubes



Net Expansion Factor, Y for k = 1.4[27]

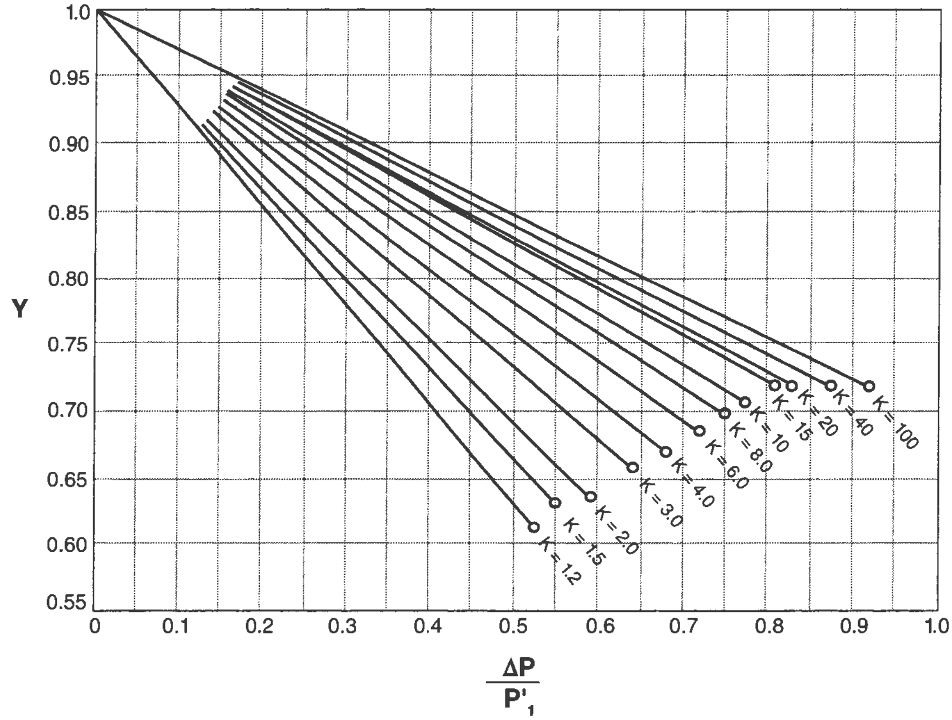


Calculated from equations presented in ASME MFC-3Ma-2007

Net Expansion Factor Y for Compressible Flow Through Pipe to a Larger Flow Area

k = 1.3

(**k** = approximately 1.3 for CO_2, SO_2, H_2O, H_2S, NH_3, N_2O, Cl_2, CH_4, C_2H_2 and C_2H_4)

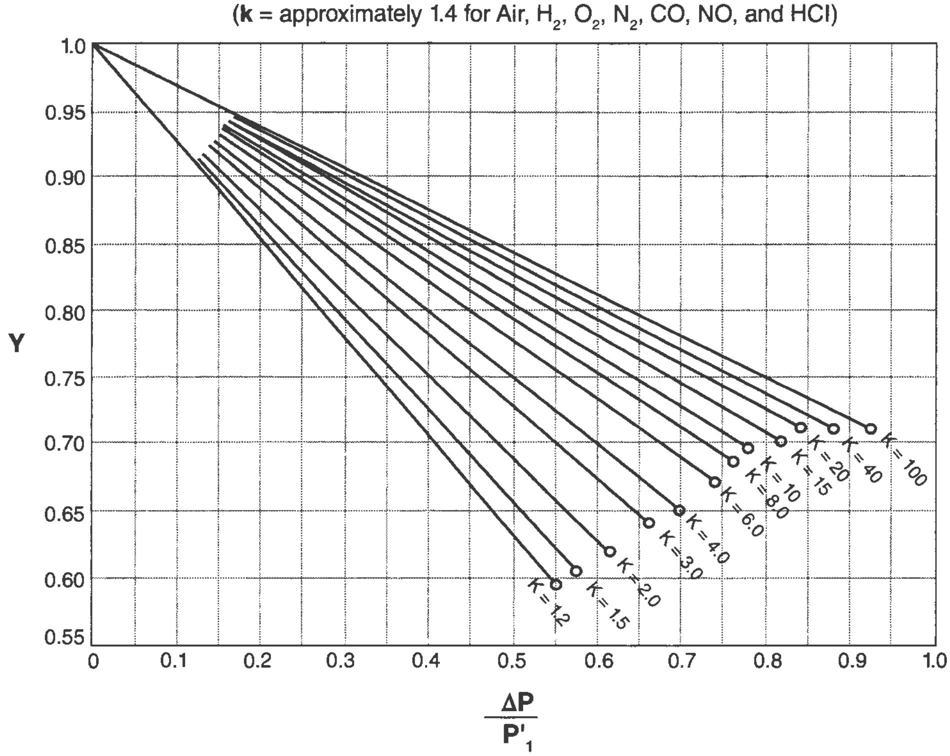


Limiting Factors For Sonic Velocity
k = 1.3

K	$\frac{\Delta P}{P'_1}$	Y
1.2	.525	.612
1.5	.550	.631
2	.593	.635
3	.642	.658
4	.678	.670
6	.722	.685
8	.750	.698
10	.773	.705
15	.807	.718
20	.831	.718
40	.877	.718
100	.920	.718

k = 1.4

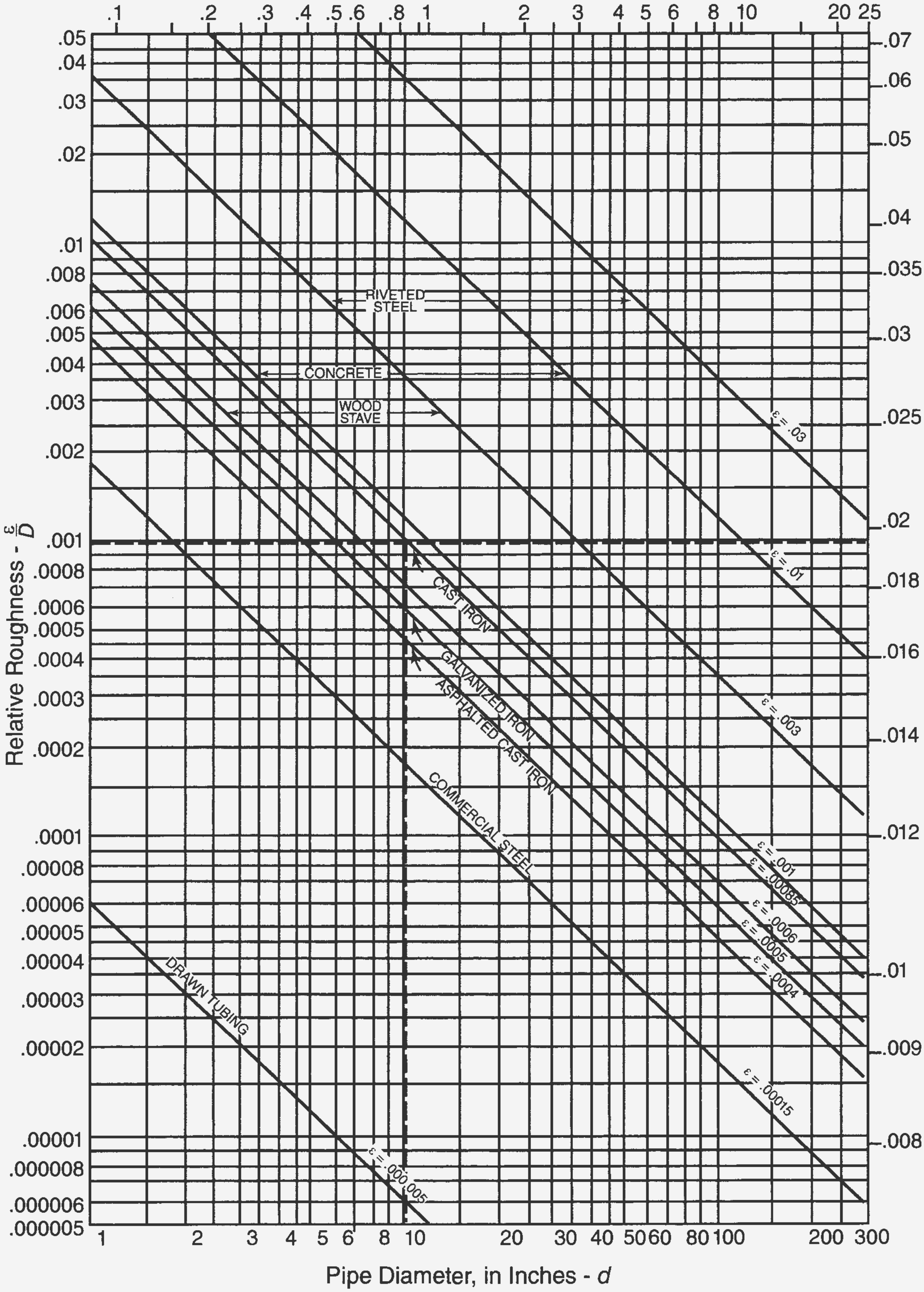
(**k** = approximately 1.4 for Air, H_2, O_2, N_2, CO, NO, and HCl)



Limiting Factors For Sonic Velocity
k = 1.3

K	$\frac{\Delta P}{P'_1}$	Y
1.2	.552	.588
1.5	.576	.606
2	.612	.622
3	.662	.639
4	.697	.649
6	.737	.671
8	.762	.685
10	.784	.695
15	.818	.702
20	.839	.710
40	.883	.710
100	.926	.710

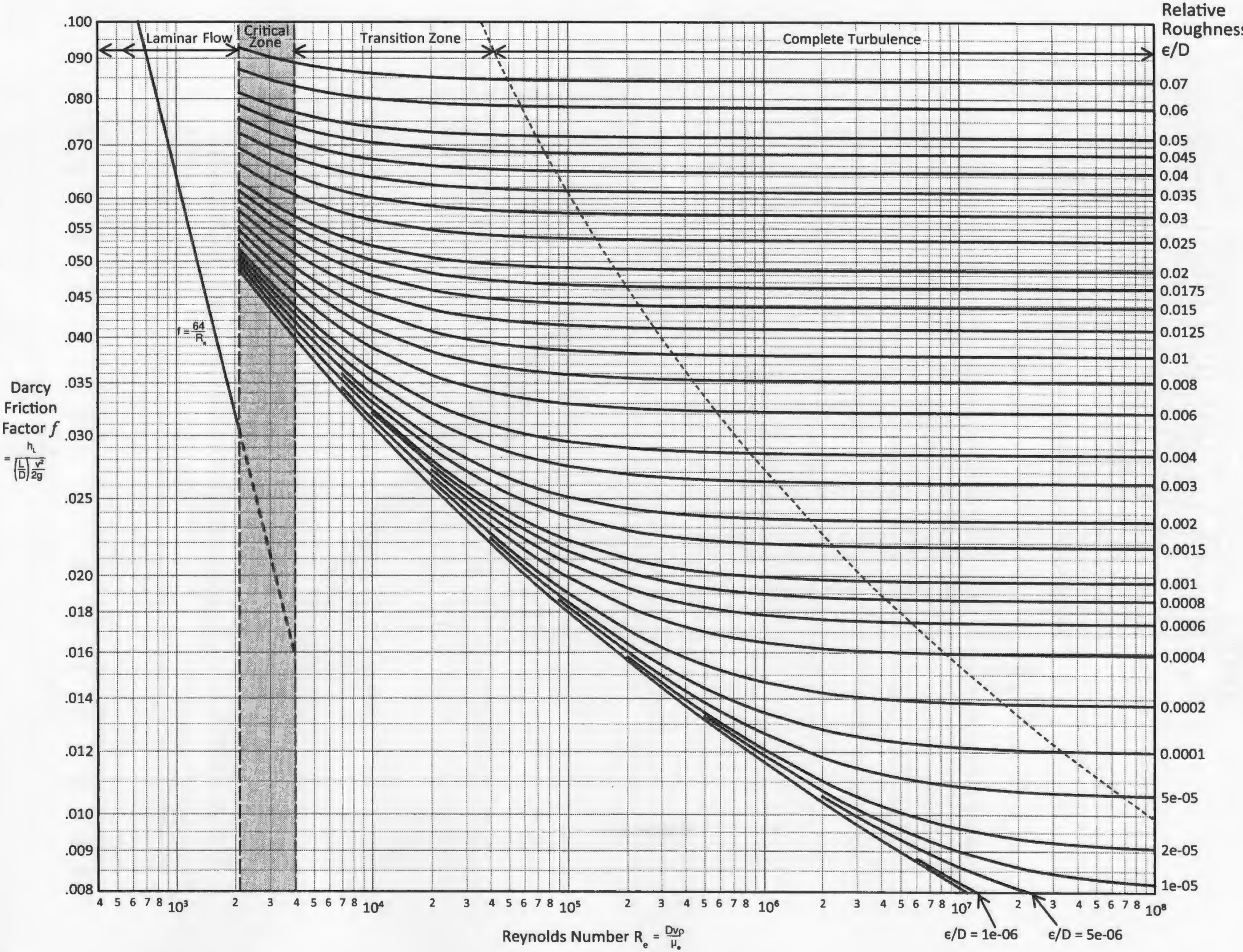
Relative Roughness of Pipe Materials and Friction Factors for Complete Turbulence[2]



Absolute Roughness (ε) is in feet.

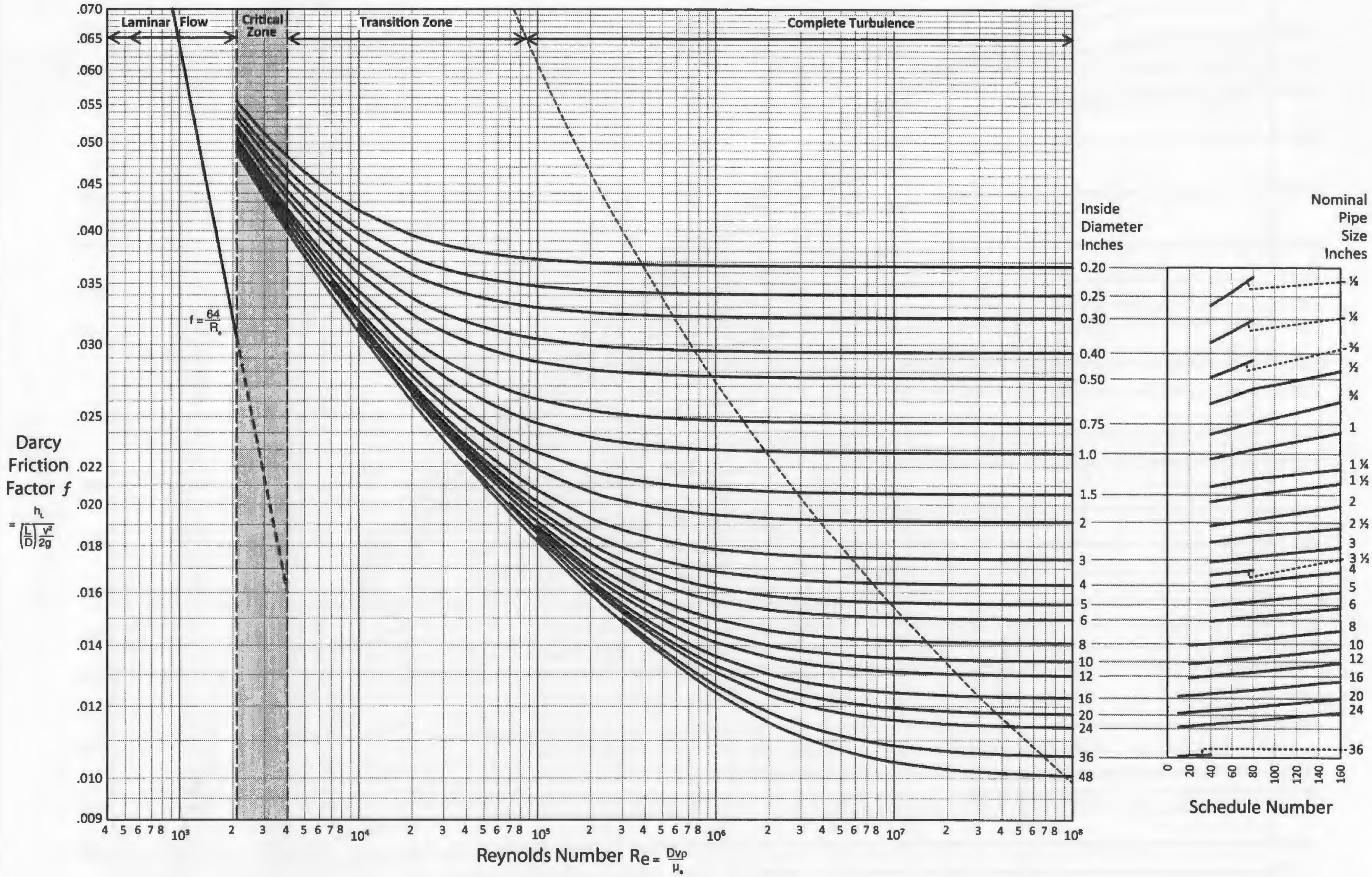
Data extracted from Friction Factors for Pipe Flow by L.F. Moody, with permission of the publisher, The American Society of Mechanical Engineers.

Friction Factors for Any Type of Commercial Pipe[2]



CRANE.

Friction Factors for Clean Commercial Steel Pipe[2]



Representative Resistance Coefficient K for Valves and Fittings

(K is based on use of schedule pipe as listed on page 2-9.)

Pipe Friction Data for Schedule 40 Clean Commercial Steel Pipe with Flow in Zone of Complete Turbulence

Nominal Size	½"	¾"	1"	1¼"	1 ½"	2"	2 ½"	3"	4"	5, 6"	8"	10-14"	16-22"	24-36"
Friction Factor (f_T)	.026	.024	.022	.021	.020	.019	.018	.017	.016	.015	.014	.013	.012	.011

$$f_T = \frac{0.25}{\left[\log\left(\frac{\varepsilon/D}{3.7}\right)\right]^2}$$

Formulas For Calculating K Factors* For Valves and Fittings with Reduced Port

(Refer to page 2-11)

Formula 1

$$K_2 = \frac{0.8\left(\sin\frac{\theta}{2}\right)\left(1-\beta^2\right)}{\beta^4} = \frac{K_1}{\beta^4}$$

Formula 2

$$K_2 = \frac{0.5\left(1-\beta^2\right)\sqrt{\sin\frac{\theta}{2}}}{\beta^4} = \frac{K_1}{\beta^4}$$

Formula 3

$$K_2 = \frac{2.6\left(\sin\frac{\theta}{2}\right)\left(1-\beta^2\right)^2}{\beta^4} = \frac{K_1}{\beta^4}$$

Formula 4

$$K_2 = \frac{\left(1-\beta^2\right)^2}{\beta^4} = \frac{K_1}{\beta^4}$$

Formula 5

$$K_2 = \frac{K_1}{\beta^4} + \text{Formula 1} + \text{Formula 3}$$

$$K_2 = \frac{K_1 + \sin\frac{\theta}{2}\left[0.8\left(1-\beta^2\right) + 2.6\left(1-\beta^2\right)^2\right]}{\beta^4}$$

*Use K furnished by valve or fitting supplier when available.

Formula 6

$$K_2 = \frac{K_1}{\beta^4} + \text{Formula 2} + \text{Formula 4}$$

$$K_2 = \frac{K_1 + 0.5\sqrt{\sin\frac{\theta}{2}}\left(1-\beta^2\right) + \left(1-\beta^2\right)^2}{\beta^4}$$

Formula 7

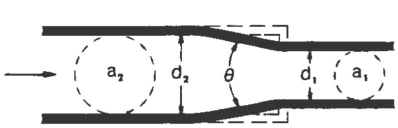
$$K_2 = \frac{K_1}{\beta^4} + \beta\,(\text{Formula 2} + \text{Formula 4}) \quad \text{When} \quad \theta = 180^\circ$$

$$K_2 = \frac{K_1 + \beta\left[0.5\left(1-\beta^2\right) + \left(1-\beta^2\right)^2\right]}{\beta^4}$$

$$\beta = \frac{d_1}{d_2} \qquad \beta^2 = \left(\frac{d_1}{d_2}\right)^2 = \frac{a_1}{a_2}$$

Subscript 1 defines dimensions and coefficients with reference to the smaller diameter. Subscript 2 refers to the larger diameter.

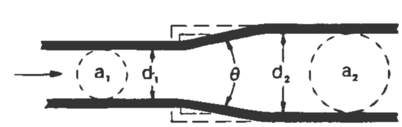
Sudden and Gradual Contraction



If: $\theta \lesssim 45^\circ$ .... K_2 = Formula 1

$45^\circ < \theta \lesssim 180^\circ$ .... K_2 = Formula 2

Sudden and Gradual Enlargement



If: $\theta \lesssim 45^\circ$ .... K_2 = Formula 3

$45^\circ < \theta \lesssim 180^\circ$ .... K_2 = Formula 4

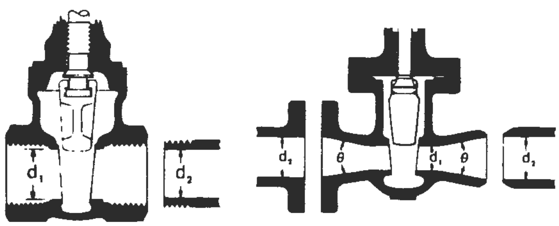
CRANE

Representative Resistance Coefficient K for Valves and Fittings

For formulas and friction data, see page A-27. K is based on use of schedule pipe as listed on page 2-9.

GATE VALVES

Wedge Disc, Double Disc, or Plug Type

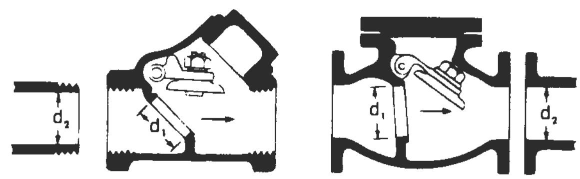


If: $\beta = 1,\ \theta = 0$ .... $K_1 = 8\,f_T$

$\beta < 1$ and $\theta \lessapprox 45°$ .... K_2 = Formula 5

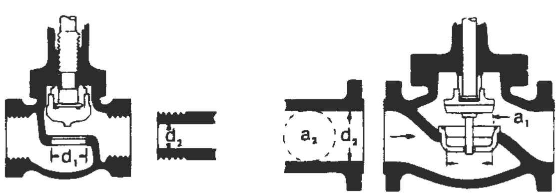
$\beta < 1$ and $45° < \theta \lessapprox 180°$......... K_2 = Formula 6

SWING CHECK VALVES

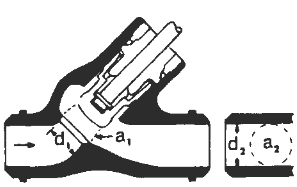


$K = 100\,f_T$	$K = 50\,f_T$
Minimum pipe velocity (fps) for full disc lift	Minimum pipe velocity (fps) for full disc lift
$= 35\sqrt{V}$	$= 60\sqrt{V}$ except U/L listed $= 100\sqrt{V}$

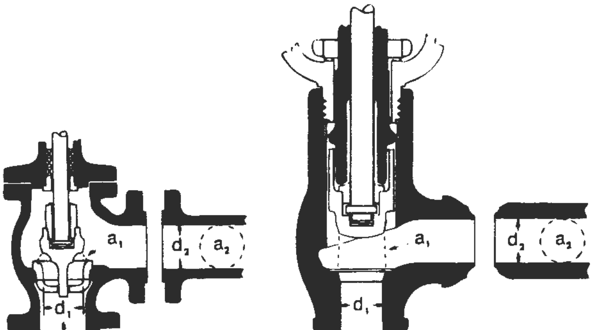
GLOBE AND ANGLE VALVES



If: $\beta = 1 \ldots K_1 = 340\,f_T$



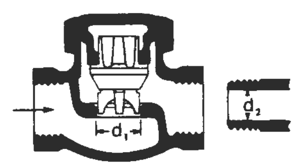
If: $\beta = 1 \ldots K_1 = 55\,f_T$



If: $\beta = 1 \ldots K_1 = 150 f_T$ If: $\beta = 1 \ldots K_1 = 55 f_T$

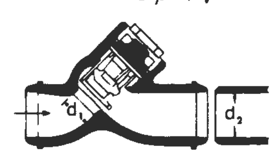
All globe and angle valves,
whether reduced seat or throttled,
If: $\beta < 1 \ldots K_2$ = Formula 7

LIFT CHECK VALVES



If: $\beta = 1 \ldots K_1 = 600\,f_T$
$\beta < 1 \ldots K_2$ = Formula 7

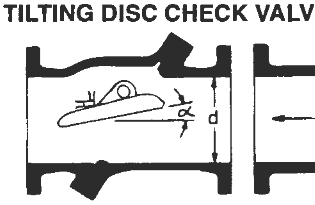
Minimum pipe velocity (fps) for full disc lift
$= 40\,\beta^2\sqrt{V}$



If: $\beta = 1 \ldots K_1 = 55\,f_T$
$\beta < 1 \ldots K_2$ = Formula 7

Minimum pipe velocity (fps) for full disc lift
$= 140\,\beta^2\sqrt{V}$

TILTING DISC CHECK VALVES

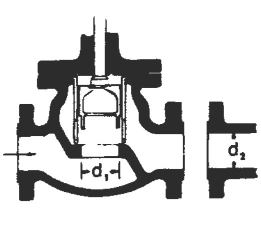


	$\alpha = 5°$	$\alpha = 15°$
Sizes 2 to 8" . . . K =	$40\,f_T$	$120\,f_T$
Sizes 10 to 14" . . . K =	$30\,f_T$	$90\,f_T$
Sizes 16 to 48". . . K =	$20\,f_T$	$60\,f_T$
Minimum pipe velocity (fps) for full disc lift =	$80\sqrt{V}$	$30\sqrt{V}$

Representative Resistance Coefficient K for Valves and Fittings

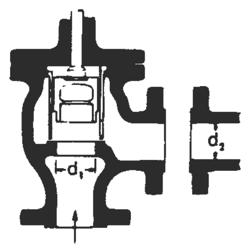
For formulas and friction data, see page A-27. K is based on use of schedule pipe as listed on page 2-9.

STOP-CHECK VALVES
(Globe and Angle Types)



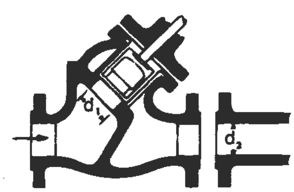
If:
$\beta = 1 \ldots K_1 = 400\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity
for full disc lift
$= 55\, \beta^2 \sqrt{V}$

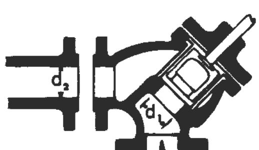


If:
$\beta = 1 \ldots K_1 = 200\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity
for full disc lift
$= 75\, \beta^2 \sqrt{V}$

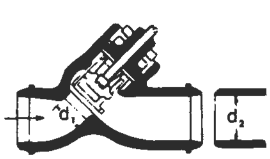


If:
$\beta = 1 \ldots K_1 = 300\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

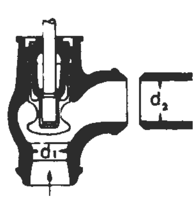


If:
$\beta = 1 \ldots K_1 = 350\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity (fps) for full disc lift
$= 60\, \beta^2 \sqrt{V}$



If:
$\beta = 1 \ldots K_1 = 55\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7



If:
$\beta = 1 \ldots K_1 = 55\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity (fps) for full disc lift
$= 140\, \beta^2 \sqrt{V}$

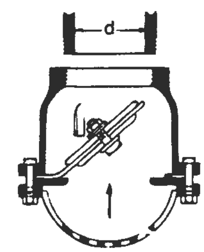
FOOT VALVES WITH STRAINER

Poppet Disc

$K = 420\, f_T$

Minimum pipe velocity
(fps) for full disc lift
$= 15 \sqrt{V}$

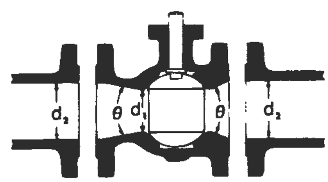
Hinged Disc



$K = 75\, f_T$

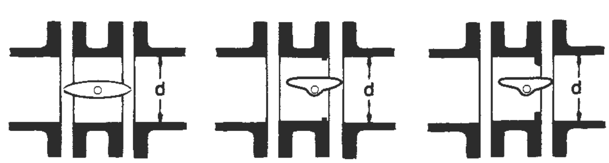
Minimum pipe velocity
(fps) for full disc lift
$= 35 \sqrt{V}$

BALL VALVES



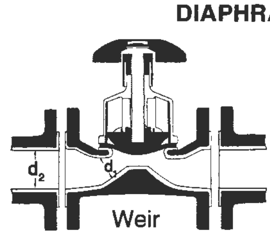
If: $\beta = 1,\ \theta = 0 \ldots\ldots\ldots\ldots\ldots\ldots K_1 = 3\, f_T$
$\beta < 1$ and $\theta \lessapprox 45° \ldots\ldots\ldots\ldots K_2 =$ Formula 5
$\beta < 1$ and $45° < \theta \lessapprox 180° \ldots\ldots\ldots K_2 =$ Formula 6

BUTTERFLY VALVES

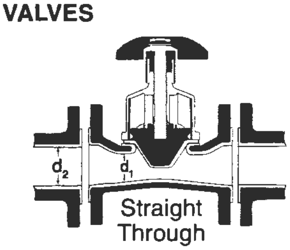


SIZE RANGE	CENTRIC	DOUBLE OFFSET	TRIPLE OFFSET
2" - 8"	$K = 45\, f_T$	$K = 74\, f_T$	$K = 218\, f_T$
10" - 14"	$K = 35\, f_T$	$K = 52\, f_T$	$K = 96\, f_T$
16" - 24"	$K = 25\, f_T$	$K = 43\, f_T$	$K = 55\, f_T$

DIAPHRAGM VALVES



$\beta = 1 \ldots K = 149\, f_T$



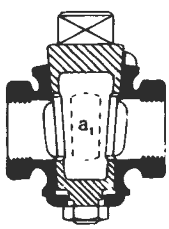
$\beta = 1 \ldots K = 39\, f_T$

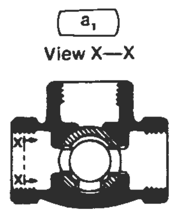
Representative Resistance Coefficient K for Valves and Fittings

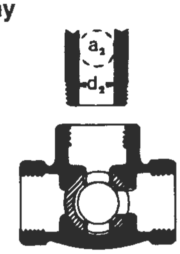
For formulas and friction data, see page A-27. K is based on use of schedule pipe as listed on page 2-9.

PLUG VALVES AND COCKS

Straight Way **3-Way**







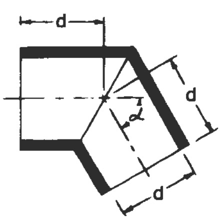
If: $\beta = 1$, $K_1 = 18 f_T$

If: $\beta = 1$, $K_1 = 30 f_T$

If: $\beta = 1$, $K_1 = 90 f_T$

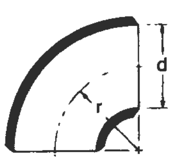
If: $\beta < 1 \ldots K_2$ = Formula 6

MITRE BENDS



α	K
0°	$2 f_T$
15°	$4 f_T$
30°	$8 f_T$
45°	$15 f_T$
60°	$25 f_T$
75°	$40 f_T$
90°	$60 f_T$

90° PIPE BENDS AND FLANGED OR BUTT-WELDING 90° ELBOWS



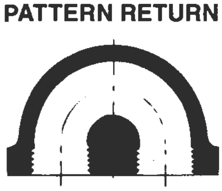
r/d	K	r/d	K
1	$20 f_T$	8	$24 f_T$
1.5	$14 f_T$	10	$30 f_T$
2	$12 f_T$	12	$34 f_T$
3	$12 f_T$	14	$38 f_T$
4	$14 f_T$	16	$42 f_T$
6	$17 f_T$	20	$50 f_T$

The resistance coefficient, K_B, for pipe bends other than 90° may be determined as follows:

$$K_B = (n - 1)\left(0.25\ \pi f_T \frac{r}{d} + 0.5\ K\right) + K$$

n = number of 90° bends
K = resistance coefficient for one 90° bend (per table)

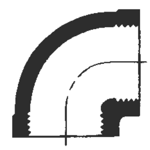
CLOSE PATTERN RETURN BENDS



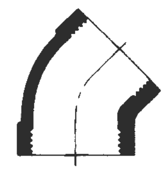
$K = 50 f_T$

STANDARD ELBOWS

90° **45°**



$K = 30 f_T$



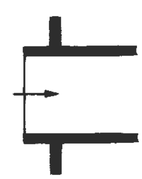
$K = 16 f_T$

STANDARD TEES AND WYES

Refer to Chapter 2, pages 2-14 through 2-16

PIPE ENTRANCE

Inward Projecting

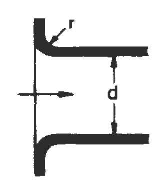


$K = 0.78$

r/d	K
0.00*	0.5
0.02	0.28
0.04	0.24
0.06	0.15
0.10	0.09
0.15 & up	0.04

*Sharp-edged

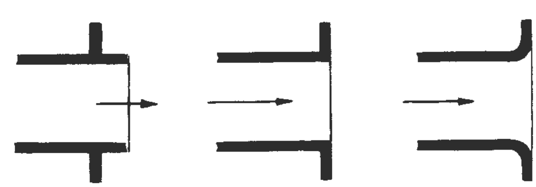
Flush



For K, see table

PIPE EXIT

Projecting **Sharp-Edged** **Rounded**



$K = 1.0$ $K = 1.0$ $K = 1.0$

Appendix B

Engineering Data

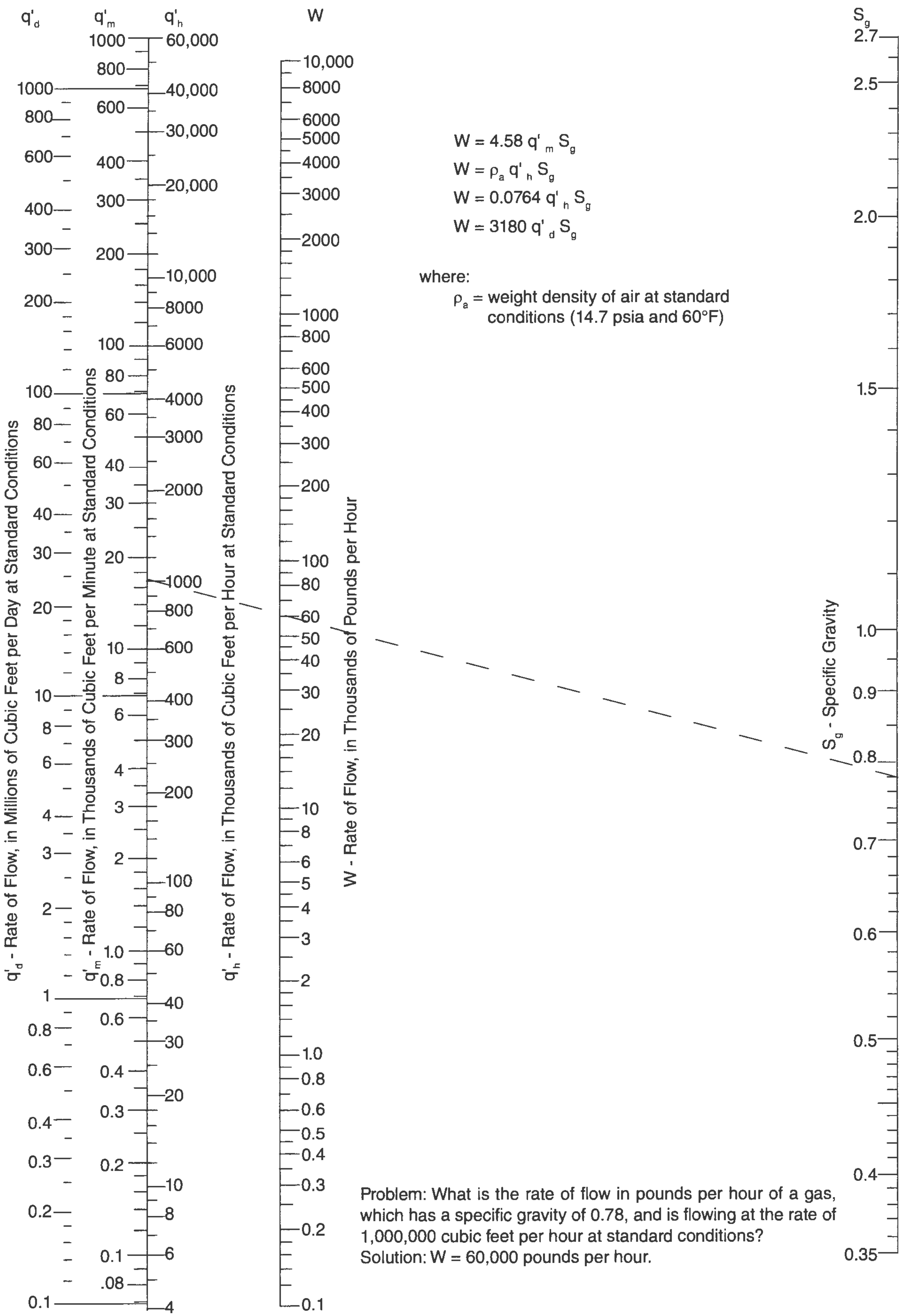
Flow problems are encountered in many fields of engineering; therefore, a wide choice of terminology prevails. Terms most widely accepted in the fluid dynamics field have been employed in this paper. In the event problems are expressed in units other than used in this paper, tables are provided for conversion.

Other useful engineering data are presented to provide direct solutions to frequently recurring factors appearing in flow formulas, as well as complete solutions to water and air flow pressure drop problems.

www. This symbol = online calculators are available at www.flowoffluids.com.

CRANE

Equivalent Volume and Weight - Flow Rates of Compressible Fluids



$W = 4.58\, q'_m\, S_g$

$W = \rho_a\, q'_h\, S_g$

$W = 0.0764\, q'_h\, S_g$

$W = 3180\, q'_d\, S_g$

where:

ρ_a = weight density of air at standard conditions (14.7 psia and 60°F)

Problem: What is the rate of flow in pounds per hour of a gas, which has a specific gravity of 0.78, and is flowing at the rate of 1,000,000 cubic feet per hour at standard conditions?
Solution: W = 60,000 pounds per hour.

www.

Equivalents of Absolute (Dynamic) Viscosity

CONVERT TO → FROM ↓ MULTIPLY BY	pascal second Pa·s	poise P	**Centipoise cP**	pound mass per foot second lbm/(ft·s)	slugs/ft·s or lbf·s/ft^2
Pa·s	1	10	1000	0.67197	0.020885
P	0.1	1	100	0.067197	0.0020885
cP	0.001	0.01	1	6.7197E-04	2.08854E-05
lbm/(ft·s)	1.4882	14.882	1488.2	1	0.031081
slugs/ft·s or lbf·s/ft^2	47.88	478.8	47880	32.174	1

To convert absolute or dynamic viscosity from one set of units to another, locate the given set of units in the left hand column and multiply the numerical value by the factor shown horizontally to the right under the set of units desired.

As an example, suppose a given absolute viscosity of 2 poise is to be converted to slugs/foot second. By referring to the table, we find the conversion factor to be 2.0885 (10^{-3}). Then, 2 (poise) times 2.0885 (10^{-3}) = 4.177 (10^{-3}) = 0.004177 slugs/ foot second.

Equivalents of Kinematic Viscosity

CONVERT TO → FROM ↓ MULTIPLY BY	meter squared per second m^2/s	stokes St	**centistokes cSt**	square foot per second ft^2/s	square inch per second in^2/s
m^2/s	1	10000	1E+06	10.764	1550
St	1E-04	1	100	0.0010764	0.155
cSt	1E-06	0.01	1	1.0764E-05	0.00155
ft^2/s	0.092903	929.03	92903	1	144
in^2/s	6.4516E-04	6.4516	645.16	0.0069444	1

To convert kinematic viscosity from one set of units to another, locate the given set of units in the left hand column and multiply the numerical value by the factor shown horizontally to the right, under the set of units desired.

As an example, suppose a given kinematic viscosity of 0.5 square foot/second is to be converted to centistokes. By referring to the table, we find the conversion factor to be 92,903. Then, 0.5 (sq.ft/sec) times 92,903 = 46,451.5 centistokes.

For conversion from kinematic to absolute viscosity, see page B-5.

Equivalents of Kinematic and Saybolt Universal Viscosity

Kinematic Viscosity, Centistokes ν	Equivalent Saybolt Universal Viscosity, SUS	
	At 100°F Basic Values	At 210°F
1.83	32.0	32.2
2.0	32.6	32.8
4.0	39.2	39.5
6.0	45.6	45.9
8.0	52.1	52.4
10.0	58.8	59.2
15.0	77.4	77.9
20.0	97.8	98.5
25.0	119.4	120.2
30.0	141.5	142.5
35.0	164.0	165.1
40.0	186.8	188.0
45.0	210	211
50.0	233	234
55.0	256	257
60.0	279	280
65.0	302	304
70.0	325	327
75.0	348	350
80.0	371	373
85.0	394	397
90.0	417	420
95.0	440	443
100.0	463	467
120.0	556	560
140.0	649	Over 120: $SUS_{210} = 4.664 \times \nu$
160.0	741	
180.0	834	
200.0	927	
220.0	1019	
240.0	1112	
260.0	1204	
280.0	1297	
300.0	1390	
320.0	1482	
340.0	1575	
360.0	1668	
380.0	1760	
400.0	1853	
420.0	1946	
440.0	2038	
460.0	2131	
480.0	2224	
500.0	2316	
Over 500	$SUS_{100} = 4.632 \times \nu$	

To convert kinematic viscosity (ν) in cSt or mm²/s to SUS at any temperature (t): $SUS_t = SUS_{100} \times [0.000061(t) + 0.9939]$

$$SUS_{100} = 4.6324v + \frac{(1.0 + 0.03264v) \times 10^5}{\left(3930.2 + 262.7v + 23.97v^2 + 1.646v^3\right)}$$

These tables are reprinted from ASTM Standard D2161 – 05 with permission. The table on the left was abstracted from Table 1 and the table on the right was abstracted from Table 3. To convert kinematic viscosity (ν) in cSt or mm²/s to SFS at 122°F or 210°F, use the table on the right or:

$$SFS_{122} = 0.4717v + \left(\frac{13924}{v^2 - 72.59v + 6816}\right)$$

$$SFS_{210} = 0.4792v + \left[\frac{5610}{\left(v^2 + 2130\right)}\right]$$

Equivalents of Kinematic and Saybolt Furol Viscosity

Kinematic Viscosity, Centistokes ν	Equivalent Saybolt Furol Viscosity, SFS	
	At 122°F	At 210°F
48	25.1	
50	26.0	25.2
60	30.6	29.7
70	35.1	34.3
80	39.6	39.0
90	44.1	43.7
100	48.6	48.4
125	60.0	60.2
150	71.5	72.1
175	83.1	84.0
200	94.8	96.0
225	106.5	107.9
250	118.2	119.9
275	129.9	131.9
300	141.7	143.8
325	153.5	155.8
350	165.2	167.8
375	177.0	179.7
400	188.8	191.7
425	201	204
450	212	216
475	224	228
500	236	240
525	248	252
550	259	264
575	271	276
600	283	288
625	295	300
650	307	311
675	318	323
700	330	335
725	342	347
750	354	359
775	366	371
800	377	383
825	389	395
850	401	407
875	413	419
900	425	431
925	436	443
950	448	455
975	460	467
1000	472	479
1025	484	491
1050	495	503
1075	507	515
1100	519	527
1125	531	539
1150	542	551
1175	554	563
1200	566	575
1225	578	587
1250	590	599
1275	601	611
1300	613	623
Over 1300	*	†

*Over 1300 Centistokes at 122°F: $SFS_{122} = 0.4717 \times \nu$ (cSt at 122°F)

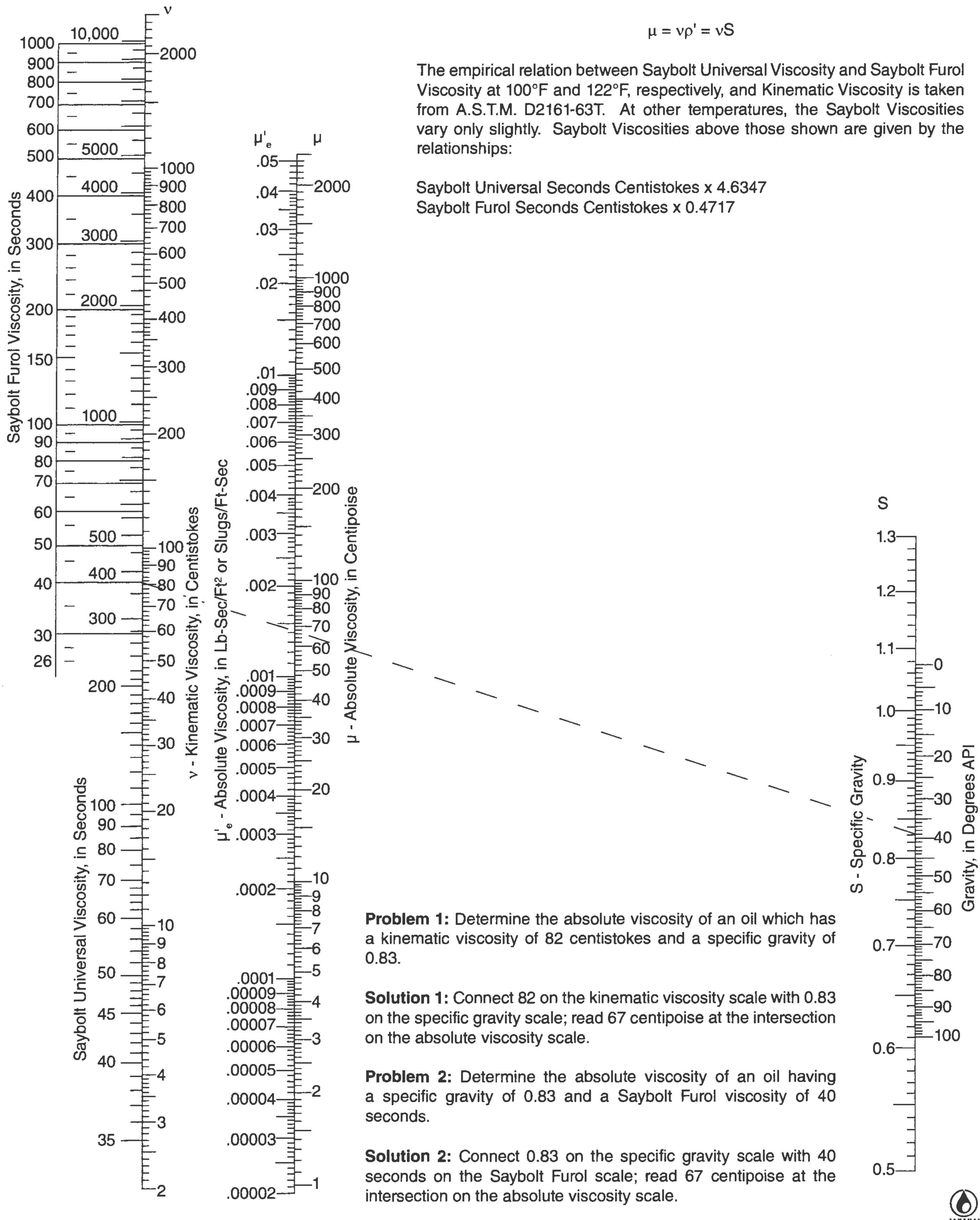
†Over 1300 Centistokes at 210°F: $SFS_{210} = 0.4792 \times \nu$ (cSt at 210°F)

Equivalents of Kinematic, Saybolt Universal, Saybolt Furol & Absolute Viscosity

$$\mu = \nu\rho' = \nu S$$

The empirical relation between Saybolt Universal Viscosity and Saybolt Furol Viscosity at 100°F and 122°F, respectively, and Kinematic Viscosity is taken from A.S.T.M. D2161-63T. At other temperatures, the Saybolt Viscosities vary only slightly. Saybolt Viscosities above those shown are given by the relationships:

Saybolt Universal Seconds Centistokes x 4.6347
Saybolt Furol Seconds Centistokes x 0.4717



Problem 1: Determine the absolute viscosity of an oil which has a kinematic viscosity of 82 centistokes and a specific gravity of 0.83.

Solution 1: Connect 82 on the kinematic viscosity scale with 0.83 on the specific gravity scale; read 67 centipoise at the intersection on the absolute viscosity scale.

Problem 2: Determine the absolute viscosity of an oil having a specific gravity of 0.83 and a Saybolt Furol viscosity of 40 seconds.

Solution 2: Connect 0.83 on the specific gravity scale with 40 seconds on the Saybolt Furol scale; read 67 centipoise at the intersection on the absolute viscosity scale.

Equivalents of Degrees API, Degrees Baumé, Specific Gravity, Weight Density, and Pounds Per Gallon at 60°F/60°F

Degrees on API or Baumé Scale	Values for API Scale			Values for Baumé Scale					
	Values for Baumé Scale			Liquids Lighter Than Water			Liquids Heavier Than Water		
	Specific Gravity S	Weight Density, Lb/ Ft³ ρ	Pounds per Gallon	Specific Gravity S	Weight Density, Lb/ Ft³ ρ	Pounds per Gallon	Specific Gravity S	Weight Density, Lb/ Ft³ ρ	Pounds per Gallon
0	...	...	...	...	...	...	1.0000	62.36	8.337
2	...	...	...	...	...	...	1.0140	63.24	8.454
4	...	...	...	...	...	...	1.0284	64.14	8.574
6	...	...	...	...	...	...	1.0432	65.06	8.697
8	...	...	...	...	...	...	1.0584	66.01	8.824
10	1.0000	62.36	8.337	1.0000	62.36	8.337	1.0741	66.99	8.955
12	0.9861	61.50	8.221	0.9859	61.49	8.219	1.0902	67.99	9.089
14	0.9725	60.65	8.108	0.9722	60.63	8.105	1.1069	69.03	9.228
16	0.9593	59.83	7.998	0.9589	59.80	7.994	1.1240	70.10	9.371
18	0.9465	59.03	7.891	0.9459	58.99	7.886	1.1417	71.20	9.518
20	0.9340	58.25	7.787	0.9333	58.20	7.781	1.1600	72.34	9.671
22	0.9218	57.87	7.736	0.9211	57.44	7.679	1.1789	73.52	9.828
24	0.9100	56.75	7.587	0.9091	56.70	7.579	1.1983	74.73	9.990
26	0.8984	56.03	7.490	0.8974	55.97	7.482	1.2185	75.99	10.159
28	0.8871	55.32	7.396	0.8861	55.26	7.387	1.2393	77.29	10.332
30	0.8762	54.64	7.305	0.8750	54.57	7.295	1.2609	78.64	10.512
32	0.8654	53.97	7.215	0.8642	53.90	7.205	1.2832	80.03	10.698
34	0.8550	53.32	7.128	0.8537	53.24	7.117	1.3063	81.47	10.891
36	0.8448	52.69	7.043	0.8434	52.60	7.031	1.3303	82.96	11.091
38	0.8348	52.06	6.960	0.8333	51.97	6.947	1.3551	84.51	11.297
40	0.8251	51.46	6.879	0.8235	51.36	6.865	1.3810	86.13	11.513
42	0.8155	50.86	6.799	0.8140	50.76	6.786	1.4078	87.80	11.737
44	0.8063	50.28	6.722	0.8046	50.18	6.708	1.4356	89.53	11.969
46	0.7972	49.72	6.646	0.7955	49.61	6.632	1.4646	91.34	12.210
48	0.7883	49.16	6.572	0.7865	49.05	6.557	1.4948	93.22	12.462
50	0.7796	48.62	6.499	0.7778	48.51	6.484	1.5263	95.19	12.725
52	0.7711	48.09	6.429	0.7692	47.97	6.413	1.5591	97.23	12.998
54	0.7628	47.57	6.359	0.7609	47.45	6.344	1.5934	99.37	13.284
56	0.7547	47.07	6.292	0.7527	46.94	6.275	1.6292	101.60	13.583
58	0.7467	46.57	6.225	0.7447	46.44	6.209	1.6667	103.94	13.895
60	0.7389	46.08	6.160	0.7368	45.95	6.143	1.7059	106.39	14.222
62	0.7313	45.61	6.097	0.7292	45.48	6.079	1.7470	108.95	14.565
64	0.7238	45.14	6.034	0.7216	45.00	6.016	1.7901	111.64	14.924
66	0.7165	44.68	5.973	0.7143	44.55	5.955	1.8354	114.46	15.302
68	0.7093	44.23	5.913	0.7071	44.10	5.895	1.8831	117.44	15.699
70	0.7022	43.79	5.854	0.7000	43.66	5.836	1.9333	120.57	16.118
72	0.6953	43.36	5.797	0.6931	43.22	5.778	...	...	...
74	0.6886	42.94	5.741	0.6863	42.80	5.722	...	...	...
76	0.6819	42.53	5.685	0.6796	42.38	5.666	...	...	...
78	0.6754	42.12	5.631	0.6731	41.98	5.612	...	...	...
80	0.6690	41.72	5.577	0.6667	41.58	5.558	...	...	...
82	0.6628	41.33	5.526	0.6604	41.19	5.506	...	...	...
84	0.6566	40.95	5.474	0.6542	40.80	5.454	...	...	...
86	0.6506	40.57	5.424	0.6482	40.42	5.404	...	...	...
88	0.6446	40.20	5.374	0.6422	40.05	5.354	...	...	...
90	0.6388	39.84	5.326	0.6364	39.69	5.306	...	...	...
92	0.6331	39.48	5.278	0.6306	39.33	5.257	...	...	...
94	0.6275	39.13	5.231	0.6250	38.98	5.211	...	...	...
96	0.6220	38.79	5.186	0.6195	38.63	5.165	...	...	...
98	0.6166	38.45	5.141	0.6140	38.29	5.119	...	...	...
100	0.6112	38.12	5.096	0.6087	37.96	5.075	...	...	...

For Formulas, see page 1-3.

Power Required for Pumping

Gals. per Min.	Theoretical Horsepower Required to Raise Water (at 60°F) to Different Heights														
	5 feet	10 feet	15 feet	20 feet	25 feet	30 feet	35 feet	40 feet	45 feet	50 feet	60 feet	70 feet	80 feet	90 feet	100 feet
5	0.006	0.013	0.019	0.025	0.032	0.038	0.044	0.051	0.057	0.063	0.076	0.088	0.101	0.114	0.126
10	0.013	0.025	0.038	0.051	0.063	0.076	0.088	0.101	0.114	0.126	0.152	0.177	0.202	0.227	0.253
15	0.019	0.038	0.057	0.076	0.095	0.114	0.133	0.152	0.171	0.190	0.227	0.265	0.303	0.341	0.379
20	0.025	0.051	0.076	0.101	0.126	0.152	0.177	0.202	0.227	0.253	0.303	0.354	0.404	0.455	0.505
25	0.032	0.063	0.095	0.126	0.158	0.190	0.221	0.253	0.284	0.316	0.379	0.442	0.505	0.568	0.632
30	0.038	0.076	0.114	0.152	0.190	0.227	0.265	0.303	0.341	0.379	0.455	0.531	0.606	0.682	0.758
35	0.044	0.088	0.133	0.177	0.221	0.265	0.310	0.354	0.398	0.442	0.531	0.619	0.707	0.796	0.884
40	0.051	0.101	0.152	0.202	0.253	0.303	0.354	0.404	0.455	0.505	0.606	0.707	0.808	0.910	1.011
45	0.057	0.114	0.171	0.227	0.284	0.341	0.398	0.455	0.512	0.568	0.682	0.796	0.910	1.023	1.137
50	0.063	0.126	0.190	0.253	0.316	0.379	0.442	0.505	0.568	0.632	0.758	0.884	1.011	1.137	1.263
60	0.076	0.152	0.227	0.303	0.379	0.455	0.531	0.606	0.682	0.758	0.910	1.061	1.213	1.364	1.516
70	0.088	0.177	0.265	0.354	0.442	0.531	0.619	0.707	0.796	0.884	1.061	1.238	1.415	1.592	1.768
80	0.101	0.202	0.303	0.404	0.505	0.606	0.707	0.808	0.910	1.011	1.213	1.415	1.617	1.819	2.021
90	0.114	0.227	0.341	0.455	0.568	0.682	0.796	0.910	1.023	1.137	1.364	1.592	1.819	2.046	2.274
100	0.126	0.253	0.379	0.505	0.632	0.758	0.884	1.011	1.137	1.263	1.516	1.768	2.021	2.274	2.526
125	0.158	0.316	0.474	0.632	0.790	0.947	1.105	1.263	1.421	1.579	1.895	2.211	2.526	2.842	3.158
150	0.190	0.379	0.568	0.758	0.947	1.137	1.326	1.516	1.705	1.895	2.274	2.653	3.032	3.411	3.790
175	0.221	0.442	0.663	0.884	1.105	1.326	1.547	1.768	1.990	2.211	2.653	3.095	3.537	3.979	4.421
200	0.253	0.505	0.758	1.011	1.263	1.516	1.768	2.021	2.274	2.526	3.032	3.537	4.042	4.548	5.053
250	0.316	0.632	0.947	1.263	1.579	1.895	2.211	2.526	2.842	3.158	3.790	4.421	5.053	5.684	6.316
300	0.379	0.758	1.137	1.516	1.895	2.274	2.653	3.032	3.411	3.790	4.548	5.305	6.063	6.821	7.579
350	0.442	0.884	1.326	1.768	2.211	2.653	3.095	3.537	3.979	4.421	5.305	6.190	7.074	7.958	8.842
400	0.505	1.011	1.516	2.021	2.526	3.032	3.537	4.042	4.548	5.053	6.063	7.074	8.084	9.095	10.11
500	0.632	1.263	1.895	2.526	3.158	3.790	4.421	5.053	5.684	6.316	7.579	8.842	10.11	11.37	12.63

Gals. per Min.	125 feet	150 feet	175 feet	200 feet	250 feet	300 feet	350 feet	400 feet
5	0.158	0.190	0.221	0.253	0.316	0.379	0.442	0.505
10	0.316	0.379	0.442	0.505	0.632	0.758	0.884	1.011
15	0.474	0.568	0.663	0.758	0.947	1.137	1.326	1.516
20	0.632	0.758	0.884	1.011	1.263	1.516	1.768	2.021
25	0.790	0.947	1.105	1.263	1.579	1.895	2.211	2.526
30	0.947	1.137	1.326	1.516	1.895	2.274	2.653	3.032
35	1.105	1.326	1.547	1.768	2.211	2.653	3.095	3.537
40	1.263	1.516	1.768	2.021	2.526	3.032	3.537	4.042
45	1.421	1.705	1.990	2.274	2.842	3.411	3.979	4.548
50	1.579	1.895	2.211	2.526	3.158	3.790	4.421	5.053
60	1.895	2.274	2.653	3.032	3.790	4.548	5.305	6.063
70	2.211	2.653	3.095	3.537	4.421	5.305	6.190	7.074
80	2.526	3.032	3.537	4.042	5.053	6.063	7.074	8.084
90	2.842	3.411	3.979	4.548	5.684	6.821	7.958	9.095
100	3.158	3.790	4.421	5.053	6.316	7.579	8.842	10.11
125	3.948	4.737	5.527	6.316	7.895	9.474	11.05	12.63
150	4.737	5.684	6.632	7.579	9.474	11.37	13.26	15.16
175	5.527	6.632	7.737	8.842	11.05	13.26	15.47	17.68
200	6.316	7.579	8.842	10.11	12.63	15.16	17.68	20.21
250	7.895	9.474	11.05	12.63	15.79	18.95	22.11	25.26
300	9.474	11.37	13.26	15.16	18.95	22.74	26.53	30.32
350	11.05	13.26	15.47	17.68	22.11	26.53	30.95	35.37
400	12.63	15.16	17.68	20.21	25.26	30.32	35.37	40.42
500	15.79	18.95	22.11	25.26	31.58	37.90	44.21	50.53

HORSEPOWER = 33,000 ft-lb/min
= 550 ft-lb/sec
= 2544.48 Btu/hr
= 745.7 watts

$(whp) = QH\rho \div 247{,}000 = QP \div 1{,}714$

$(bhp) = (whp) \div \eta_p = QH\rho \div 247{,}000\ \eta_p$

$(\eta_p) = QH\rho \div [247{,}000\ (bhp)]$

where : (whp) = water horsepower
H = pump head in feet
(bhp) = brake horsepower
η_p = pump efficiency

Overall efficiency (η_o) takes into account all losses in the pump and driver.

$\eta_o = \eta_p\, \eta_D\, \eta_T$

where : η_D = driver efficiency
η_T = transmission efficiency
η_v = volumetric efficiency

$$\eta_v\ (\%) = \frac{\text{actual pump displacement (Q) (100)}}{\text{theoretical pump displacement (Q)}}$$

Note: For fluids other than water, multiply table values by specific gravity. In pumping liquids with a viscosity considerably higher than that of water, the pump capacity and head are reduced. To calculate the horsepower for such fluids, pipe friction head must be added to the elevation head to obtain the total head; this value is inserted in the first horsepower equation given above.

Specific gravity of water page A-7.
Specific gravity of liquid, other than water page A-8.

CRANE

US Conversion Tables

Length

CONVERT TO → FROM ↓ MULTIPLY BY	millimeter mm	centimeter cm	meter m	kilometer km	inch in	**foot ft**	mile mi
mm	1	0.1	0.001	1E-06	0.03937	0.0032808	6.2137E-07
cm	10	1	0.01	1E-05	0.3937	0.032808	6.2137E-06
m	1000	100	1	0.001	39.37	3.2808	6.2137E-04
km	1E+06	1E+05	1000	1	39370	3280.8	0.62137
in	25.4	2.54	0.0254	2.54E-05	1	0.083332	1.5783E-05
ft	304.8	30.48	0.3048	3.048E-04	12	1	1.8939E-04
mi	1.6093E+06	1.6093E+05	1609.3	1.6093	63360	5280	1

Area

CONVERT TO → FROM ↓ MULTIPLY BY	square millimeter mm^2	square centimeter cm^2	square meter m^2	square kilometer km^2	square inch in^2	**square foot ft^2**	square mile mi^2
mm^2	1	0.01	1E-06	1E-12	0.00155	1.0764E-05	3.861E-13
cm^2	100	1	0.0001	1E-10	0.155	0.0010764	3.861E-11
m^2	1E+06	1E+04	1	1E-06	1550	10.764	3.861E-07
km^2	1E+12	1E+10	1E+06	1	1.55E+09	1.0764E+07	3.861E-01
in^2	645.16	6.4516	6.4516E-04	6.4516E-10	1	0.0069444	2.491E-10
ft^2	92903	929.03	0.092903	9.2903E-08	144	1	3.587E-08
mi^2	2.59E+12	2.59E+10	2.59E+06	2.59	4.0145E+09	2.7878E+07	1

Volume

CONVERT TO → FROM ↓ MULTIPLY BY	cubic millimeter mm^3	cubic centimeter cm^3(mL)	cubic meter m^3	liter L	cubic inch in^3	**cubic foot ft^3**	gallon (U.S.) gal
mm^3	1	0.001	1E-09	1E-06	6.1024E-05	3.5315E-08	2.6417E-07
cm^3(mL)	1000	1	1E-06	0.001	0.061024	3.5315E-05	2.6417E-04
m^3	1E+09	1E+06	1	1000	61024	35.315	264.17
L	1E+06	1000	0.001	1	61.024	0.035315	0.26417
in^3	16387	16.387	1.6387E-05	0.016387	1	5.787E-04	0.004329
ft^3	2.832E+07	28317	0.028	28.317	1728	1	7.4805
gal	3.7854E+06	3785.4	0.0037854	3.7854	231	0.13368	1

1 Barrel (U.S.)(bbl)= 42 Gallons (U.S.) = 0.15899 m^3
1 gallon (Imperial) = 1.201 Gallons (U.S.) = 0.0045461 m^3

Velocity

CONVERT TO → FROM ↓ MULTIPLY BY	meter per second m/s	meter per minute m/min	kilometer per hour kph	**foot per second ft/s**	foot per minute ft/min	mile per hour mi/h
m/s	1	60	3.6	3.2808	196.85	2.2369
m/min	0.016667	1	0.06	0.054681	3.2808	0.037282
kph	0.27778	16.667	1	0.91134	54.681	0.62137
ft/s	0.3048	18.288	1.0973	1	60	0.68182
ft/min	0.00508	0.3048	0.018288	0.016667	1	0.011364
mi/h	0.44704	26.822	1.6093	1.4667	88	1

US Conversion Tables

Mass

CONVERT TO → FROM ↓ MULTIPLY BY	gram g	kilogram kg	tonne t	**pound (avoirdupois) lbm**	slug slug	ton, short sh ton
g	1	0.001	1E-06	0.0022046	6.8522E-05	1.1023E-06
kg	1000	1	0.001	2.2046	0.068522	0.0011023
t	1E+06	1000	1	2204.6	68.522	1.1023
lbm	453.59	0.45359	4.5359E-04	1	0.031081	5E-04
slug	14594	14.594	0.014594	32.174	1	0.016087
sh ton	9.0718E+05	907.185	0.90718	2000	62.162	1

Mass Flow Rate

CONVERT TO → FROM ↓ MULTIPLY BY	kilogram per second kg/s	kilogram per hour kg/h	tonne per hour t/h	pound per second lbm/s	**pound per hour lbm/h**	ton, short per hour sh ton/h
kg/s	1	3600	3.6	2.2046	7936.6	3.9683
kg/h	2.7778E-04	1	0.001	6.124E-04	2.2046	0.0011023
t/h	0.27778	1000	1	0.6124	2204.6	1.1023
lbm/s	0.45359	1632.9	1.6329	1	3600	1.8
lbm/h	1.26E-04	0.45359	4.5359E-04	2.7778E-04	1	5E-04
sh ton/h	0.252	907.18	0.90718	0.55556	2000	1

Volumetric Flow Rate

CONVERT TO → FROM ↓ MULTIPLY BY	liter per second L/s	liter per minute L/min	cubic meter per second m³/s	cubic meter per hour m³/h	cubic foot per second ft³/s	cubic foot per hour ft³/h	**gallon(U.S.) per minute gpm**	U.S. barrels per day bbl/day
L/s	1	60	0.001	3.6	0.035315	127.13	15.85	543.44
L/min	0.016667	1	1.6667E-05	0.06	5.8858E-04	2.1189	0.26417	9.0573
m³/s	1000	6E+04	1	3600	35.315	1.2713E+05	1.585E+04	5.4344E+05
m³/h	0.27778	16.667	2.7778E-04	1	0.0098096	35.315	4.4029	150.96
ft³/s	28.317	1699	0.028317	101.94	1	3600	448.83	15388
ft³/h	0.0078658	0.47195	7.8658E-06	0.028317	2.7778E-04	1	0.12468	4.2746
gpm	0.06309	3.7854	6.309E-05	0.22712	0.002228	8.0208	1	34.286
bbl/day	0.0018401	0.11041	1.8401E-06	0.0066245	6.4984E-05	0.23394	0.029167	1

Force

CONVERT TO → FROM ↓ MULTIPLY BY	dyne dyn	newton N	kilonewton kN	kilogram force kgf	ounce force ozf	**pound force lbf**	poundal poundal	ton force ton force
dyn	1	1E-05	1E-08	1.0197E-06	3.5969E-05	2.2481E-06	7.233E-05	1.124E-09
N	1E+05	1	0.001	0.10197	3.5969	0.22481	7.233	1.124E-04
kN	1E+08	1000	1	101.97	3596.9	224.81	7233	0.1124
kgf	980670	9.8067	0.0098067	1	35.274	2.2046	70.932	0.0011023
ozf	27801	0.27801	2.7801E-04	0.02835	1	0.0625	2.0109	3.125E-05
lbf	4.4482E+05	4.4482	0.0044482	0.45359	16	1	32.174	5E-04
poundal	13826	0.13826	1.3826E-04	0.014098	0.4973	0.031081	1	1.554E-05
ton force	8.8964E+08	8896.443	8.8964	907.18	32000	2000	64348	1

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US Conversion Tables

Pressure and Liquid Head

CONVERT TO → FROM ↓ MULTIPLY BY	pascal Pa	kilopascal kPa	bar bar	meter of water, conventional $m\ H_2O$	millimeter of mercury, conventional mm Hg	**pounds per square inch psi (lbf/in²)**	inch of water, conventional in H_2O	foot of water, conventional ft H_2O	inch of mercury, conventional in Hg	atmosphere atm
Pa	1	0.001	1E-05	1.0197E-04	0.0075006	1.4504E-04	0.0040146	3.3455E-04	2.953E-04	9.8692E-06
kPa	1000	1	0.01	0.10197	7.5006	0.14504	4.0146	0.33455	0.2953	0.0098692
bar	1E+05	100	1	10.197	750.06	14.504	401.46	33.455	29.53	0.98692
$m\ H_2O$	9806.7	9.8067	0.098067	1	73.556	1.4223	39.37	3.2808	2.8959	0.096784
mm Hg	133.32	0.13332	0.0013332	0.013595	1	0.019337	0.53524	0.044603	0.03937	0.0013158
psi (lbf/in²)	6894.8	6.8948	0.068948	0.70307	51.715	1	27.68	2.3067	2.036	0.068046
in H_2O	249.09	0.24909	0.0024909	0.0254	1.8683	0.036127	1	0.083333	0.073556	0.0024583
ft H_2O	2989.1	2.9891	0.029891	0.3048	22.42	0.43353	12	1	0.88267	0.0295
in Hg	3386.4	3.3864	0.033864	0.34531	25.4	0.49115	13.595	1.1329	1	0.033421
atm	101325	101.325	1.01325	10.332	760	14.696	406.78	33.899	29.921	1

Energy, Work, Heat

CONVERT TO → FROM ↓ MULTIPLY BY	erg erg	joule J	kilojoule kJ	calorie cal	kilowatt hour kWh	**British thermal unit Btu**	therm thm
erg	1	1E-07	1E-10	2.3885E-08	2.7778E-14	9.4782E-11	9.4782E-16
J	1E+07	1	0.001	0.23885	2.7778E-07	9.4782E-04	9.4782E-09
kJ	1E+10	1000	1	238.85	2.7778E-04	0.94782	9.4782E-06
cal	4.1868E+07	4.1868	0.0041868	1	1.163E-06	0.0039683	3.9683E-08
kWh	4E+13	4E+06	3600	8.5985E+05	1	3412.1	0.034121
Btu	1.0551E+10	1055.1	1.0551	252	2.9307E-04	1	1E-05
thm	1.0551E+15	1.0551E+08	1.0551E+05	2.52E+07	29.307	1E+05	1

Power

CONVERT TO → FROM ↓ MULTIPLY BY	Watt W	kiloWatt kW	metric horsepower hp (metric)	foot pound force per second ft lbf/s	**horsepower hp**
W	1	0.001	0.0013596	0.73756	0.001341
kW	1000	1	1.3596	737.56	1.341
hp (metric)	735.5	0.7355	1	542.48	0.98632
ft lbf/s	1.3558	0.0013558	0.0018434	1	0.0018182
hp	745.7	0.7457	1.0139	550	1

Density

CONVERT TO → FROM ↓ MULTIPLY BY	gram per cubic centimeter g/cm³ or kg/L	kilogram per cubic meter kg/m³	pound per gallon lb/gal	**pound per cubic foot lb/ft³**	slug per cubic foot slug/ft³
g/cm³ or kg/L	1	1000	8.3454	62.428	1.9403
kg/m³	0.001	1	0.0083454	0.062428	0.0019403
lb/gal	0.11983	119.83	1	7.4805	0.2325
lb/ft³	0.016018	16.018	0.13368	1	0.031081
slug/ft³	0.51538	515.38	4.301	32.174	1

Temperature Equivalents

To convert degrees Celsius to degrees Fahrenheit:

$$t = \frac{9}{5}t_c + 32$$

To convert degrees Kelvin to degrees Fahrenheit:

$$t = \frac{9}{5}(T_K - 273.15) + 32$$

To convert degrees Fahrenheit to Rankine:

$$T = t + 459.67$$

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Flow of Water Through Schedule 40 Steel Pipe*

Pressure Drop (dP) per 100 feet** and Velocity (Vel) in Schedule 40 Pipe for Water at 60°F																			
Flow Rate		Vel	dP	Vel	dP	Vel	dP	Vel	dP	Vel	dP	Vel	dP	Vel	dP	Vel	dP	Vel	dP
		ft/sec	psi	ft/sec	psi	ft/sec	psi	ft/sec	psi	ft/sec	psi	ft/sec	psi	ft/sec	psi	ft/sec	psi	ft/sec	psi
gpm	ft³/sec	1/8"		1/4"		3/8"		1/2"		3/4"		1"		1 1/4"		1 1/2"		2"	
0.2	0.000446	1.13	2.13	0.617	0.342	0.336	0.102	0.211	0.040										
0.3	0.000668	1.69	4.31	0.925	0.999	0.504	0.153	0.317	0.060										
0.4	0.000891	2.26	7.18	1.23	1.64	0.672	0.383	0.422	0.080	0.241	0.026								
0.5	0.00111	2.82	10.7	1.54	2.43	0.840	0.561	0.528	0.185	0.301	0.033								
0.6	0.00134	3.39	14.9	1.85	3.35	1.01	0.77	0.634	0.253	0.361	0.039								
0.8	0.00178	4.52	25.3	2.47	5.60	1.34	1.27	0.845	0.414	0.481	0.108	0.297	0.035	0.172	0.007				
1	0.00223	5.65	38.3	3.08	8.40	1.68	1.89	1.06	0.610	0.602	0.159	0.371	0.050	0.215	0.008				
2	0.00446	11.29	142.0	6.17	30.27	3.36	6.59	2.11	2.09	1.20	0.530	0.742	0.166	0.429	0.045	0.315	0.022		
3	0.00668	2 1/2"		9.25	65.15	5.04	13.96	3.17	4.37	1.80	1.09	1.11	0.337	0.644	0.090	0.473	0.043		
4	0.00891			12.33	112.94	6.72	23.95	4.22	7.42	2.41	1.84	1.48	0.563	0.858	0.149	0.630	0.071	0.382	0.022
5	0.01114	0.335	0.014	3"		8.40	36.55	5.28	11.25	3.01	2.76	1.86	0.840	1.07	0.221	0.788	0.105	0.478	0.032
6	0.01337	0.402	0.019			10.08	51.73	6.34	15.85	3.61	3.86	2.23	1.17	1.29	0.306	0.946	0.145	0.574	0.044
8	0.01782	0.536	0.031	0.347	0.011	13.45	89.86	8.45	27.33	4.81	6.59	2.97	1.98	1.72	0.513	1.26	0.242	0.765	0.072
10	0.02228	0.670	0.046	0.434	0.016	3 1/2"		10.56	41.84	6.02	10.02	3.71	2.99	2.15	0.769	1.58	0.361	0.956	0.107
15	0.03342	1.01	0.094	0.651	0.033			15.84	91.37	9.02	21.62	5.57	6.37	3.22	1.62	2.36	0.755	1.43	0.222
20	0.04456	1.34	0.157	0.868	0.055	0.649	0.027	4"		12.03	37.52	7.42	10.97	4.29	2.76	3.15	1.28	1.91	0.374
25	0.05570	1.68	0.235	1.08	0.082	0.811	0.040			15.04	57.71	9.28	16.78	5.36	4.19	3.94	1.94	2.39	0.562
30	0.06684	2.01	0.327	1.30	0.113	0.974	0.056	0.756	0.030	5"		11.14	23.80	6.44	5.91	4.73	2.73	2.87	0.786
35	0.07798	2.35	0.434	1.52	0.150	1.14	0.074	0.882	0.040			12.99	32.02	7.51	7.92	5.52	3.64	3.35	1.05
40	0.08912	2.68	0.555	1.74	0.191	1.30	0.094	1.01	0.051	0.641	0.017	14.85	41.45	8.58	10.22	6.30	4.69	3.82	1.34
45	0.1003	3.02	0.690	1.95	0.237	1.46	0.116	1.13	0.063	0.722	0.021	16.71	52.07	9.65	12.80	7.09	5.86	4.30	1.67
50	0.1114	3.35	0.839	2.17	0.287	1.62	0.141	1.26	0.076	0.802	0.025	18.56	63.90	10.73	15.67	7.88	7.16	4.78	2.03
60	0.1337	4.02	1.18	2.60	0.401	1.95	0.196	1.51	0.106	0.962	0.035	6"		12.87	22.25	9.46	10.14	5.74	2.87
70	0.1560	4.69	1.57	3.04	0.534	2.27	0.261	1.76	0.140	1.12	0.047			15.02	29.98	11.03	13.63	6.69	3.84
80	0.1782	5.36	2.02	3.47	0.684	2.60	0.333	2.02	0.179	1.28	0.059	0.89	0.024	17.16	38.84	12.61	17.63	7.65	4.95
90	0.2005	6.03	2.53	3.91	0.853	2.92	0.415	2.27	0.222	1.44	0.073	1.00	0.030	19.31	48.84	14.18	22.13	8.61	6.20
100	0.2228	6.70	3.09	4.34	1.04	3.25	0.504	2.52	0.270	1.60	0.089	1.11	0.036	8"		15.76	27.15	9.56	7.58
125	0.2785	8.38	4.72	5.42	1.58	4.06	0.765	3.15	0.408	2.00	0.134	1.39	0.055			19.70	41.89	11.95	11.65
150	0.3342	10.05	6.70	6.51	2.23	4.87	1.08	3.78	0.573	2.41	0.187	1.67	0.076	0.96	0.020	23.64	59.79	14.34	16.56
175	0.3899	11.73	9.01	7.59	2.99	5.68	1.44	4.41	0.766	2.81	0.249	1.94	0.101	1.12	0.026	27.58	80.84	16.73	22.34
200	0.4456	13.40	11.66	8.68	3.86	6.49	1.86	5.04	0.985	3.21	0.320	2.22	0.129	1.28	0.034	10"		19.12	28.96
225	0.5013	15.08	14.64	9.76	4.84	7.30	2.32	5.67	1.23	3.61	0.398	2.50	0.160	1.44	0.042			21.51	36.44
250	0.5570	16.75	17.96	10.85	5.93	8.11	2.84	6.30	1.50	4.01	0.485	2.78	0.195	1.60	0.051	1.02	0.017	23.90	44.77
275	0.6127	18.43	21.62	11.93	7.13	8.92	3.41	6.93	1.80	4.41	0.580	3.05	0.233	1.76	0.060	1.12	0.020	26.29	53.95
300	0.6684	20.10	25.62	13.02	8.43	9.74	4.03	7.56	2.13	4.81	0.683	3.33	0.274	1.92	0.071	1.22	0.023	28.68	63.98
325	0.7241	21.78	29.95	14.10	9.84	10.55	4.70	8.19	2.48	5.21	0.795	3.61	0.318	2.08	0.082	1.32	0.027	12"	
350	0.7798	23.45	34.61	15.19	11.36	11.36	5.42	8.82	2.85	5.61	0.915	3.89	0.366	2.24	0.094	1.42	0.031		
375	0.8355	25.13	39.62	16.27	12.99	12.17	6.19	9.45	3.26	6.01	1.04	4.16	0.416	2.40	0.107	1.53	0.035	1.07	0.015
400	0.8912	26.80	44.95	17.36	14.73	12.98	7.01	10.08	3.69	6.41	1.18	4.44	0.470	2.57	0.120	1.63	0.039	1.15	0.017
425	0.9469	28.48	50.63	18.44	16.57	13.79	7.88	10.71	4.14	6.82	1.32	4.72	0.527	2.73	0.135	1.73	0.044	1.22	0.019
450	1.003	30.16	56.64	19.53	18.52	14.60	8.80	11.34	4.63	7.22	1.47	5.00	0.587	2.89	0.150	1.83	0.049	1.29	0.021
475	1.058	14"		20.61	20.58	15.41	9.78	11.97	5.13	7.62	1.63	5.28	0.650	3.05	0.166	1.93	0.054	1.36	0.023
500	1.114			21.70	22.75	16.23	10.80	12.60	5.67	8.02	1.80	5.55	0.716	3.21	0.183	2.03	0.059	1.43	0.025
550	1.225	1.30	0.019	23.87	27.41	17.85	13.00	13.86	6.82	8.82	2.16	6.11	0.858	3.53	0.218	2.24	0.071	1.58	0.030
600	1.337	1.42	0.022	26.04	32.50	19.47	15.40	15.12	8.07	9.62	2.56	6.66	1.01	3.85	0.257	2.44	0.083	1.72	0.035
650	1.448	1.54	0.026	28.21	38.02	21.09	18.00	16.38	9.43	10.42	2.98	7.22	1.18	4.17	0.299	2.64	0.097	1.86	0.041
700	1.560	1.66	0.029	30.38	43.97	22.72	20.81	17.64	10.89	11.23	3.44	7.77	1.36	4.49	0.343	2.85	0.111	2.01	0.047
750	1.671	1.78	0.033	32.55	50.35	24.34	23.82	18.90	12.45	12.03	3.93	8.33	1.55	4.81	0.391	3.05	0.126	2.15	0.053
800	1.782	1.90	0.038	16"		25.96	27.02	20.16	14.12	12.83	4.45	8.88	1.75	5.13	0.442	3.25	0.142	2.29	0.060
850	1.894	2.02	0.042			27.58	30.43	21.42	15.90	13.63	5.01	9.44	1.97	5.45	0.495	3.46	0.160	2.44	0.067
900	2.005	2.13	0.047	1.63	0.024	29.21	34.04	22.68	17.77	14.43	5.59	9.99	2.20	5.77	0.552	3.66	0.178	2.58	0.075
950	2.117	2.25	0.052	1.72	0.027	30.83	37.86	23.94	19.76	15.24	6.21	10.55	2.44	6.09	0.612	3.87	0.197	2.72	0.083
1,000	2.228	2.37	0.057	1.82	0.030	32.45	41.87	25.20	21.84	16.04	6.86	11.11	2.69	6.41	0.675	4.07	0.217	2.87	0.091
1,100	2.451	2.61	0.068	2.00	0.035	18"		27.72	26.33	17.64	8.26	12.22	3.24	7.05	0.809	4.48	0.259	3.15	0.109
1,200	2.674	2.85	0.080	2.18	0.041			30.24	31.23	19.24	9.79	13.33	3.83	7.70	0.955	4.88	0.305	3.44	0.128
1,300	2.896	3.08	0.093	2.36	0.048	1.86	0.027	32.76	36.55	20.85	11.44	14.44	4.48	8.34	1.11	5.29	0.355	3.73	0.149
1,400	3.119	3.32	0.107	2.54	0.055	2.01	0.031	35.28	42.29	22.45	13.22	15.55	5.17	8.98	1.28	5.70	0.409	4.01	0.171
1,500	3.342	3.56	0.122	2.72	0.063	2.15	0.035	20"		24.06	15.13	16.66	5.91	9.62	1.47	6.10	0.466	4.30	0.195
1,600	3.565	3.79	0.137	2.90	0.071	2.29	0.040			25.66	17.17	17.77	6.70	10.26	1.66	6.51	0.527	4.59	0.220
1,800	4.010	4.27	0.171	3.27	0.088	2.58	0.049	2.08	0.029	28.87	21.63	19.99	8.43	11.54	2.08	7.32	0.660	5.16	0.275
2,000	4.456	4.74	0.209	3.63	0.108	2.87	0.060	2.31	0.035	32.07	26.61	22.21	10.36	12.83	2.55	8.14	0.808	5.73	0.335
2,500	5.570	5.93	0.319	4.54	0.164	3.59	0.091	2.89	0.053	24"		27.76	16.03	16.03	3.94	10.17	1.24	7.17	0.513
3,000	6.684	7.12	0.452	5.45	0.231	4.30	0.129	3.46	0.075			33.32	22.94	19.24	5.61	12.21	1.76	8.60	0.727
3,500	7.798	8.30	0.607	6.35	0.310	5.02	0.172	4.04	0.100	2.79	0.040	38.87	31.07	22.45	7.58	14.24	2.38	10.03	0.978
4,000	8.912	9.49	0.784	7.26	0.400	5.74	0.222	4.62	0.129	3.19	0.052	32"		25.65	9.85	16.27	3.08	11.47	1.27
4,500	10.03	10.67	0.984	8.17	0.502	6.45	0.278	5.19	0.161	3.59	0.064			28.86	12.41	18.31	3.87	12.90	1.59
5,000	11.14	11.86	1.21	9.08	0.614	7.17	0.339	5.77	0.197	3.99	0.079	2.18	0.018	32.07	15.26	20.34	4.76	14.33	1.95
6,000	13.37	14.23	1.72	10.89	0.872	8.61	0.481	6.93	0.279	4.79	0.111	2.61	0.025	36"		24.41	6.80	17.20	2.78
7,000	15.60	16.60	2.32	12.71	1.18	10.04	0.648	8.08	0.375	5.59	0.149	3.05	0.033			28.48	9.19	20.06	3.75
8,000	17.82	18.97	3.00	14.52	1.52	11.47	0.838	9.23	0.484	6.38	0.192	3.48	0.042	2.75	0.024	32.55	11.95	22.93	4.87
9,000	20.05	21.35	3.78	16.34	1.91	12.91	1.05	10.39	0.608	7.18	0.240	3.92	0.053	3.09	0.029	36.62	15.07	25.80	6.14
10,000	22.28	23.72	4.65	18.16	2.35	14.34	1.29	11.54	0.745	7.98	0.294	4.36	0.065	3.43	0.036			28.66	7.55
12,000	26.74	28.46	6.64	21.79	3.35	17.21	1.84	13.85	1.06	9.58	0.417	5.23	0.091	4.12	0.051			34.40	10.80
14,000	31.19	33.20	8.99	25.42	4.54	20.08	2.49	16.16	1.43	11.17	0.562	6.10	0.123	4.80	0.068			40.13	14.63
16,000	35.65	37.95	11.70	29.05	5.89	22.95	3.23	18.47	1.85	12.77	0.727	6.97	0.158	5.49	0.087				
18,000	40.10	42.69	14.75	32.68	7.43	25.82	4.06	20.78	2.33	14.37	0.914	7.84	0.198	6.18	0.109				
20,000	44.56	47.43	18.16	36.31	9.14	28.69	5.00	23.09	2.87	15.96	1.12	8.71	0.243	6.86	0.134				

* For pipe other than Schedule 40, see explanation on page B-12.

** For length other than 100 ft: pressure drop is proportional to length. For 50 ft of pipe, pressure drop is one-half the value given in the table; for 300 ft, three times the given value, etc. Velocity is a function of flow area and is constant for a given flow rate and independent of pipe length.

Flow of Air Through Schedule 40 Steel Pipe

For lengths of pipe other than 100 feet, the pressure drop is proportional to the length. Thus, for 50 feet of pipe, the pressure drop is approximately one-half the value given in the table; for 300 feet, three times the given value, etc.

The pressure drop is also inversely proportional to the absolute pressure and directly proportional to the absolute temperature.

Therefore, to determine the pressure drop for inlet or average pressures other than 100 psi and at temperatures other than 60°F, multiply the values given in the table by the ratio:

$$\left(\frac{100 + 14.7}{P + 14.7}\right)\left(\frac{460 + t}{520}\right)$$

where:
"P" is the inlet or average gauge pressure in pounds per square inch, and, "t" is the temperature in degrees Fahrenheit under consideration.

The cubic feet per minute of compressed air at any pressure is inversely proportional to the absolute pressure and directly proportional to the absolute temperature.

To determine the cubic feet per minute of compressed air at any temperature and pressure other than standard conditions, multiply the value of cubic feet per minute of free air by the ratio:

$$\left(\frac{14.7}{14.7+P}\right)\left(\frac{460 + t}{(520)}\right)$$

Calculations for Pipe Other than Schedule 40

To determine the velocity of water, or the pressure drop of water or air, through pipe other than Schedule 40, use the following formulas:

$$v_a = v_{40}\left(\frac{d_{40}}{d_a}\right)^2$$

$$\Delta P_a = \Delta P_{40}\left(\frac{d_{40}}{d_a}\right)^5$$

Subscript "a" refers to the Schedule of pipe through which velocity or pressure drop is desired.

Subscript "40" refers to the velocity or pressure drop through Schedule 40 pipe, as given in the tables on these facing pages.

Free Air q'_m Cubic Feet Per Minute at 60°F and 14.7 psia	Compressed Air Cubic Feet Per Minute at 60°F and 100 psig	Pressure Drop of Air In Pounds per Square Inch Per 100 Feet of Schedule 40 Pipe For Air at 100 Pounds per Square Inch Gauge Pressure and 60°F Temperature								
		1/8"	1/4"	3/8"	1/2"					
1	0.128	0.361	0.083	0.018						
2	0.256	1.31	0.285	0.064	0.020					
3	0.384	3.06	0.605	0.133	0.042	3/4"				
4	0.513	4.83	1.04	0.226	0.071					
5	0.641	7.45	1.58	0.343	0.106	0.027	1"			
6	0.769	10.6	2.23	0.408	0.148	0.037				
8	1.025	18.6	3.89	0.848	0.255	0.062	0.019			
10	1.282	28.7	5.96	1.26	0.356	0.094	0.029	1 1/4"		
15	1.922	...	13.0	2.73	0.834	0.201	0.062		1 1/2"	
20	2.563	...	22.8	4.76	1.43	0.345	0.102	0.026		
25	3.204	...	35.6	7.34	2.21	0.526	0.156	0.039	0.019	
30	3.845	...	...	10.5	3.15	0.748	0.219	0.055	0.026	
35	4.486	...	...	14.2	4.24	1.00	0.293	0.073	0.035	
40	5.126	...	...	18.4	5.49	1.30	0.379	0.095	0.044	2"
45	5.767	...	...	23.1	6.90	1.62	0.474	0.116	0.055	
50	6.408			28.5	8.49	1.99	0.578	0.149	0.067	0.019
60	7.690	2 1/2"		40.7	12.2	2.85	0.819	0.200	0.094	0.027
70	8.971			...	16.5	3.83	1.10	0.270	0.126	0.036
80	10.25	0.019		...	21.4	4.96	1.43	0.350	0.162	0.046
90	11.53	0.023		...	27.0	6.25	1.80	0.437	0.203	0.058
100	12.82	0.029	3"		33.2	7.69	2.21	0.534	0.247	0.070
125	16.02	0.044			...	11.9	3.39	0.825	0.380	0.107
150	19.22	0.062	0.021		...	17.0	4.87	1.17	0.537	0.151
175	22.43	0.083	0.028	3 1/2"	...	23.1	6.60	1.58	0.727	0.205
200	25.63	0.107	0.036		...	30.0	8.54	2.05	0.937	0.264
225	28.84	0.134	0.045	0.022		37.9	10.8	2.59	1.19	0.331
250	32.04	0.164	0.055	0.027		...	13.3	3.18	1.45	0.404
275	35.24	0.191	0.066	0.032		...	16.0	3.83	1.75	0.484
300	38.45	0.232	0.078	0.037		...	19.0	4.56	2.07	0.573
325	41.65	0.270	0.090	0.043	4"	...	22.3	5.32	2.42	0.673
350	44.87	0.313	0.104	0.050		...	25.8	6.17	2.80	0.776
375	48.06	0.356	0.119	0.057	0.030	...	29.6	7.05	3.20	0.887
400	51.26	0.402	0.134	0.064	0.034	...	33.6	8.02	3.64	1.00
425	54.47	0.452	0.151	0.072	0.038	...	37.9	9.01	4.09	1.13
450	57.67	0.507	0.168	0.081	0.042	...	...	10.2	4.59	1.26
475	60.88	0.562	0.187	0.089	0.047		...	11.3	5.09	1.40
500	64.08	0.623	0.206	0.099	0.052		...	12.5	5.61	1.55
550	70.49	0.749	0.248	0.118	0.062		...	15.1	6.79	1.87
600	76.90	0.887	0.293	0.139	0.073	5"	...	18.0	8.04	2.21
650	83.30	1.04	0.342	0.163	0.086		...	21.1	9.43	2.60
700	89.71	1.19	0.395	0.188	0.099	0.032		24.3	10.9	3.00
750	96.12	1.36	0.451	0.214	0.113	0.036		27.9	12.6	3.44
800	102.5	1.55	0.513	0.244	0.127	0.041		31.8	14.2	3.90
850	108.9	1.74	0.576	0.274	0.144	0.04	6"	35.9	16.0	4.40
900	115.3	1.95	0.642	0.305	0.160	0.051		40.2	18.0	4.91
950	121.8	2.18	0.715	0.340	0.178	0.057	0.023	...	20.0	5.47
1,000	128.2	2.40	0.788	0.375	0.197	0.063	0.025	...	22.1	6.06
1,100	141.0	2.89	0.948	0.451	0.236	0.075	0.030	...	26.7	7.29
1,200	153.8	3.44	1.13	0.533	0.279	0.089	0.035	...	31.8	8.63
1,300	166.6	4.01	1.32	0.626	0.327	0.103	0.041	...	37.3	10.1
1,400	179.4	4.65	1.52	0.718	0.377	0.119	0.047			11.8
1,500	192.2	5.31	1.74	0.824	0.431	0.136	0.054			13.5
1,600	205.1	6.04	1.97	0.932	0.490	0.154	0.061	8"		15.3
1,800	230.7	7.65	2.50	1.18	0.616	0.193	0.075			19.3
2,000	256.3	9.44	3.06	1.45	0.757	0.237	0.094	0.023	10"	23.9
2,500	320.4	14.7	4.76	2.25	1.17	0.366	0.143	0.035		37.3
3,000	384.5	21.1	6.82	3.20	1.67	0.524	0.204	0.051	0.016	
3,500	448.6	28.8	9.23	4.33	2.26	0.709	0.276	0.068	0.022	
4,000	512.6	37.6	12.1	5.66	2.94	0.919	0.358	0.088	0.028	12"
4,500	576.7	47.6	15.3	7.16	3.69	1.16	0.450	0.111	0.035	
5,000	640.8	...	18.8	8.85	4.56	1.42	0.552	0.136	0.043	0.018
6,000	769.0	...	27.1	12.7	6.57	2.03	0.794	0.195	0.061	0.025
7,000	897.1	...	36.9	17.2	8.94	2.76	1.07	0.262	0.082	0.034
8,000	1025	...	...	22.5	11.7	3.59	1.39	0.339	0.107	0.044
9,000	1153	...	...	28.5	14.9	4.54	1.76	0.427	0.134	0.055
10,000	1282	...	...	35.2	18.4	5.60	2.16	0.526	0.164	0.067
11,000	1410	...	...	...	22.2	6.78	2.62	0.633	0.197	0.081
12,000	1538	...	...	...	26.4	8.07	3.09	0.753	0.234	0.096
13,000	1666	...	...	...	31.0	9.47	3.63	0.884	0.273	0.112
14,000	1794	...	...	...	36.0	11.0	4.21	1.02	0.316	0.129
15,000	1922	...	...	...	...	12.6	4.84	1.17	0.364	0.148
16,000	2051	...	...	...	...	14.3	5.50	1.33	0.411	0.167
18,000	2307	...	...	...	...	18.2	6.96	1.68	0.520	0.213
20,000	2563	...	...	...	...	22.4	8.60	2.01	0.642	0.260
22,000	2820	...	...	...	...	27.1	10.4	2.50	0.771	0.314
24,000	3076	...	...	...	...	32.3	12.4	2.97	0.918	0.371
26,000	3332	...	...	...	...	37.9	14.5	3.49	1.12	0.435
28,000	3588	...	...	...	...	...	16.9	4.04	1.25	0.505
30,000	3845	...	...	...	...	...	19.3	4.64	1.42	0.520

Pipe Data - Carbon and Alloy Steel - Stainless Steel

Nominal Pipe Size Inches	Outside Diameter O.D. Inches	Identification: Steel: Iron Pipe Size	Identification: Steel: Sched. No.	Identification: Stainless Steel Sched. No.	Wall Thickness (t) Inches	Inside Diameter (d) Inches	Area of Metal Square Inches	Transverse Internal Area: (a) Square Inches	Transverse Internal Area: (A) Square Feet	Moment of Inertia (I) Inches	Weight Pipe Pounds per foot	Weight Water Pounds per foot pipe	External Surface Sq. Ft. per foot of Pipe	Section Modulus $\left(2 \frac{I}{O.D.}\right)$
		...	...	10S	.049	.307	.0548	.0740	.00051	.00088	.19	.032	.106	.00437
1/8	0.405	STD	40	40S	.068	.269	.0720	.0568	.00040	.00106	.24	.025	.106	.00523
		XS	80	80S	.095	.215	.0925	.0364	.00025	.00122	.31	.016	.106	.00602
		...	...	10S	.065	.410	.0970	.1320	.00091	.00279	.33	.057	.141	.01032
1/4	0.540	STD	40	40S	.088	.364	.1250	.1041	.00072	.00331	.42	.045	.141	.01227
		XS	80	80S	.119	.302	.1574	.0716	.00050	.00377	.54	.031	.141	.01395
		...	...	10S	.065	.545	.1246	.2333	.00162	.00586	.42	.101	.178	.01736
3/8	0.675	STD	40	40S	.091	.493	.1670	.1910	.00133	.00729	.57	.083	.178	.02160
		XS	80	80S	.126	.423	.2173	.1405	.00098	.00862	.74	.061	.178	.02554
		...	...	5S	.065	.710	.1583	.3959	.00275	.01197	.54	.172	.220	.02849
		...	...	10S	.083	.674	.1974	.3568	.00248	.01431	.67	.155	.220	.03407
1/2		STD	40	40S	.109	.622	.2503	.3040	.00211	.01709	.85	.132	.220	.04069
	0.840	XS	80	80S	.147	.546	.3200	.2340	.00163	.02008	1.09	.102	.220	.04780
		...	160	...	.187	.466	.3836	.1706	.00118	.02212	1.31	.074	.220	.05267
		XXS	...	...	.294	.252	.5043	.050	.00035	.02424	1.71	.022	.220	.05772
		...	...	5S	.065	.920	.2011	.6648	.00462	.02450	.69	.288	.275	.04667
		...	...	10S	.083	.884	.2521	.6138	.00426	.02969	.86	.266	.275	.05655
		STD	40	40S	.113	.824	.3326	.5330	.00371	.03704	1.13	.231	.275	.07055
3/4	1.050	XS	80	80S	.154	.742	.4335	.4330	.00300	.04479	1.47	.188	.275	.08531
		...	160	...	.219	.612	.5698	.2961	.00206	.05269	1.94	.128	.275	.10036
		XXS	...	...	.308	.434	.7180	.148	.00103	.05792	2.44	.064	.275	.11032
		...	...	5S	.065	1.185	.2553	1.1029	.00766	.04999	.87	.478	.344	.07603
		...	...	10S	.109	1.097	.4130	.9452	.00656	.07569	1.40	.409	.344	.11512
		STD	40	40S	.133	1.049	.4939	.8640	.00600	.08734	1.68	.375	.344	.1328
1	1.315	XS	80	80S	.179	.957	.6388	.7190	.00499	.1056	2.17	.312	.344	.1606
		...	160	...	.250	.815	.8365	.5217	.00362	.1251	2.84	.230	.344	.1903
		XXS	...	...	.358	.599	1.0760	.282	.00196	.1405	3.66	.122	.344	.2136
		...	...	5S	.065	1.530	.3257	1.839	.01277	.1038	1.11	.797	.435	.1250
		...	...	10S	.109	1.442	.4717	1.633	.01134	.1605	1.81	.708	.435	.1934
		STD	40	40S	.140	1.380	.6685	1.495	.01040	.1947	2.27	.649	.435	.2346
1 1/4	1.660	XS	80	80S	.191	1.278	.8815	1.283	.00891	.2418	3.00	.555	.435	.2913
		...	160	...	.250	1.160	1.1070	1.057	.00734	.2839	3.76	.458	.435	.3421
		XXS	...	...	.382	.896	1.534	.630	.00438	.3411	5.21	.273	.435	.4110
		...	...	5S	.065	1.770	.3747	2.461	.01709	.1579	1.28	1.066	.497	.1662
		...	...	10S	.109	1.682	.6133	2.222	.01543	.2468	2.09	.963	.497	.2598
		STD	40	40S	.145	1.610	.7995	2.036	.01414	.3099	2.72	.882	.497	.3262
1 1/2	1.900	XS	80	80S	.200	1.500	1.068	1.767	.01225	.3912	3.63	.765	.497	.4118
		...	160	...	.281	1.338	1.429	1.406	.00976	.4824	4.86	.608	.497	.5078
		XXS	...	...	.400	1.100	1.885	.950	.00660	.5678	6.41	.42	.497	.5977
		...	...	5S	.065	2.245	.4717	3.958	.02749	.3149	1.61	1.72	.622	.2652
		...	...	10S	.109	2.157	.7760	3.654	.02538	.4992	2.64	1.58	.622	.4204
		STD	40	40S	.154	2.067	1.075	3.355	.02330	.6657	3.65	1.45	.622	.5606
2	2.375	XS	80	80S	.218	1.939	1.477	2.953	.02050	.8679	5.02	1.28	.622	.7309
		...	160	...	.344	1.687	2. 190	2.241	.01556	1.162	7.46	.97	.622	.979
		XXS	...	...	.436	1.503	2.656	1.774	.01232	1.311	9.03	.77	.622	1.104
		...	...	5S	.083	2.709	.7280	5.764	.04002	.7100	2.48	2.50	.753	.4939
		...	...	10S	.120	2.635	1.039	5.453	.03787	.9873	3.53	2.36	.753	.6868
		STD	40	40S	.203	2.469	1.704	4.788	.03322	1.530	5.79	2.07	.753	1.064
2 1/2	2.875	XS	80	80S	.276	2. 323	2.254	4.238	.02942	1.924	7.66	1.87	.753	1.339
		...	160	...	.375	2.125	2.945	3.546	.02463	2.353	10.01	1.54	.753	1.638
		XXS	...	...	.552	1.771	4.028	2.464	.01710	2.871	13.69	1.07	.753	1.997
		...	...	5S	.083	3.334	.8910	8.730	.06063	1.301	3.03	3.78	.916	.7435
		...	...	10S	.120	3.260	1.274	8.347	.05796	1.822	4.33	3.62	.916	1.041
		STD	40	40S	.216	3.068	2.228	7.393	.05130	3.017	7.58	3.20	.916	1.724
3	3.500	XS	80	80S	.300	2.900	3.016	6.605	.04587	3.894	10.25	2.86	.916	2.225
		...	160	...	.438	2.624	4.205	5.408	.037055	5.032	14.32	2.35	.916	2.876
		XXS	...	...	.600	2.300	5.466	4.155	.02885	5.993	18.58	1.80	.916	3.424

Identification, wall thickness and weights are extracted from ANSI B36.10 and B36.19. The notations STD, XS, and XXS indicate Standard, Extra Strong, and Double Extra Strong pipe respectively.

Transverse Internal Area values listed in "square feet" also represent volume in cubic feet per foot of pipe length.

CRANE®

Pipe Data - Carbon and Alloy Steel - Stainless Steel

Nominal Pipe Size Inches	Outside Diameter O.D. Inches	Identification: Steel: Iron Pipe Size	Identification: Steel: Sched. No.	Identification: Stainless Steel Sched. No.	Wall Thickness (t) Inches	Inside Diameter (d) Inches	Area of Metal Square Inches	Transverse Internal Area (a) Square Inches	Transverse Internal Area (A) Square Feet	Moment of Inertia (I) Inches	Weight Pipe Pounds per foot	Weight Water Pounds per foot pipe	External Surface Sq. Ft. per foot of Pipe	Section Modulus $\left(2\frac{I}{O.D.}\right)$
3 ½	4.000	...	...	5S	.083	3.834	1.021	11.545	.08017	1.960	3.48	5.00	1.047	.9799
		...	...	10S	.120	3.760	1.463	11.104	.07711	2.755	4.97	4.81	1.047	1.378
		STD	40	40S	.226	3.548	2.680	9.886	.06870	4.788	9.11	4.29	1.047	2.394
		XS	80	80S	.318	3.364	3.678	8.888	.06170	6.280	12.50	3.84	1.047	3.140
4	4.500	...	...	5S	.083	4.334	1.152	14.75	.10245	2.810	3.92	6.39	1.178	1.249
		...	...	10S	.120	4.260	1.651	14.25	.09898	3.963	5.61	6.18	1.178	1.761
		STD	40	40S	.237	4.026	3.174	12.73	.08840	7.233	10.79	5.50	1.178	3.214
		XS	80	80S	.337	3.826	4.407	11.50	.07986	9.610	14.98	4.98	1.178	4.271
		...	120	...	.438	3.624	5.595	10.31	.0716	11.65	19.00	4.47	1.178	5.178
		...	160	...	.531	3.438	6.621	9.28	.0645	13.27	22.51	4.02	1.178	5.898
		XXS	...	...	.674	3.152	8.101	7.80	.0542	15.28	27.54	3.38	1.178	6.791
5	5.563	...	...	5S	.109	5.345	1.868	22.44	.1558	6.947	6.36	9.72	1.456	2.498
		...	...	10S	.134	5.295	2.285	22.02	.1529	8.425	7.77	9.54	1.456	3.029
		STD	40	40S	.258	5.047	4.300	20.01	.1390	15.16	14.62	8.67	1.456	5.451
		XS	80	80S	.375	4.813	6.112	18.19	.1263	20.67	20.78	7.88	1.456	7.431
		...	120	...	.500	4.563	7.953	16.35	.1136	25.73	27.04	7.09	1.456	9.250
		...	160	...	.625	4.313	9.696	14.61	.1015	30.03	32.96	6.33	1.456	10.796
		XXS	...	...	.750	4.063	11.340	12.97	.0901	33.63	38.55	5.61	1.456	12.090
6	6.625	...	...	5S	.109	6.407	2.231	32.24	.2239	11.85	7.60	13.97	1.734	3.576
		...	...	10S	.134	6.357	2.733	31.74	.2204	14.40	9.29	13.75	1.734	4.346
		STD	40	40S	.280	6.065	5.581	28.89	.2006	28.14	18.97	12.51	1.734	8.496
		XS	80	80S	.432	5.761	8.405	26.07	.1810	40.49	28.57	11.29	1.734	12.22
		...	120	...	.562	5.501	10.70	23.77	.1650	49.61	36.39	10.30	1.734	14.98
		...	160	...	.719	5.187	13.32	21.15	.1469	58.97	45.35	9.16	1.734	17.81
		XXS	...	...	.864	4.897	15.64	18.84	.1308	66.33	53.16	8.16	1.734	20.02
8	8.625	...	...	5S	.109	8.407	2.916	55.51	.3855	26.44	9.93	24.06	2.258	6.131
		...	...	10S	.148	8.329	3.941	54.48	.3784	35.41	13.40	23.61	2.258	8.212
		...	20	...	.250	8.125	6.57	51.85	.3601	57.72	22.36	22.47	2.258	13.39
		...	30	...	.277	8.071	7.26	51.16	.3553	63.35	24.70	22.17	2.258	14.69
		STD	40	40S	.322	7.981	8.40	50.03	.3474	72.49	28.55	21.70	2.258	16.81
		...	60	...	.406	7.813	10.48	47.94	.3329	88.73	35.64	20.77	2.258	20.58
		XS	80	80S	.500	7.625	12.76	45.66	.3171	105.7	43.39	19.78	2.258	24.51
		...	100	...	.594	7.437	14.96	43.46	.3018	121.3	50.95	18.83	2.258	28.14
		...	120	...	.719	7.187	17.84	40.59	.2819	140.5	60.71	17.59	2.258	32.58
		...	140	...	.812	7.001	19.93	38.50	.2673	153.7	67.76	16.68	2.258	35.65
		XXS	...	...	.875	6.875	21.30	37.12	.2578	162.0	72.42	16.10	2.258	37.56
		...	160	...	.906	6.813	21.97	36.46	.2532	165.9	74.69	15.80	2.258	38.48
10	10.750	...	...	5S	.134	10.482	4.36	86.29	.5992	63.0	15.19	37.39	2.814	11.71
		...	...	10S	.165	10.420	5.49	85.28	.5922	76.9	18.65	36.95	2.814	14.30
		...	20	...	.250	10.250	8.24	82.52	.5731	113.7	28.04	35.76	2.814	21.15
		...	30	...	.307	10.136	10.07	80.69	.5603	137.4	34.24	34.96	2.814	25.57
		STD	40	40S	.365	10.020	11.90	78.86	.5475	160.7	40.48	34.20	2.814	29.90
		XS	60	80S	.500	9.750	16.10	74.66	.5185	212.0	54.74	32.35	2.814	39.43
		...	80	...	.594	9.562	18.92	71.84	.4989	244.8	64.43	31.13	2.814	45.54
		...	100	...	.719	9.312	22.63	68.13	.4732	286.1	77.03	29.53	2.814	53.22
		...	120	...	.844	9.062	26.24	64.53	.4481	324.2	89.29	27.96	2.814	60.32
		XXS	140	...	1.000	8.750	30.63	60.13	.4176	367.8	104.13	26.06	2.814	68.43
		...	160	...	1.125	8.500	34.02	56.75	.3941	399.3	115.64	24.59	2.814	74.29
12	12.75	...	...	5S	.156	12.438	6.17	121.50	.8438	122.4	20.98	52.65	3.338	19.2
		...	...	10S	.180	12.390	7.11	120.57	.8373	140.4	24.17	52.25	3.338	22.0
		...	20	...	.250	12.250	9.82	117.86	.8185	191.8	33.38	51.07	3.338	30.2
		...	30	...	.330	12.090	12.87	114.80	.7972	248.4	43.77	49.74	3.338	39.0
		STD	...	40S	.375	12.000	14.58	113.10	.7854	279.3	49.56	49.00	3.338	43.8
		...	40	...	.406	11.938	15.77	111.93	.7773	300.3	53.52	48.50	3.338	47.1
		XS	...	80S	.500	11.750	19.24	108.43	.7528	361.5	65.42	46.92	3.338	56.7
		...	60	...	.562	11.626	21.52	106.16	.7372	400.4	73.15	46.00	3.338	62.8
		...	80	...	.688	11.374	26.03	101.64	.7058	475.1	88.63	44.04	3.338	74.6
		...	100	...	.844	11.062	31.53	96.14	.6677	561.6	107.32	41.66	3.338	88.1
		XXS	120	...	1.000	10.750	36.91	90.76	.6303	641.6	125.49	39.33	3.338	100.7
		...	140	...	1.125	10.500	41.08	86.59	.6013	700.5	139.67	37.52	3.338	109.9
		...	160	...	1.312	10.126	47.14	80.53	.5592	781.1	160.27	34.89	3.338	122.6

Identification, wall thickness and weights are extracted from ANSI B36.10 and B36.19. The notations STD, XS, and XXS indicate Standard, Extra Strong, and Double Extra Strong pipe respectively.

Transverse Internal Area values listed in "square feet" also represent volume in cubic feet per foot of pipe length.

Pipe Data - Carbon and Alloy Steel - Stainless Steel

Nominal Pipe Size Inches	Outside Diameter O.D. Inches	Identification: Steel: Iron Pipe Size	Identification: Steel: Sched. No.	Identification: Stainless Steel Sched. No.	Wall Thickness (t) Inches	Inside Diameter (d) Inches	Area of Metal Square Inches	Transverse Internal Area (a) Square Inches	Transverse Internal Area (A) Square Feet	Moment of Inertia (I) Inches	Weight Pipe Pounds per foot	Weight Water Pounds per foot pipe	External Surface Sq. Ft. per foot of Pipe	Section Modulus $\left(2\frac{I}{O.D.}\right)$
		...	...	5S	.156	13.688	6.78	147.15	1.0219	162.6	23.07	63.77	3.665	23.2
		...	...	10S	.188	13.624	8.16	145.78	1.0124	194.6	27.73	63.17	3.665	27.8
		...	10	...	.250	13.500	10.80	143.14	.9940	255.3	36.71	62.03	3.665	36.6
		...	20	...	.312	13.376	13.42	140.52	.9758	314.4	45.61	60.89	3.665	45.0
		STD	30	...	.375	13.250	16.05	137.88	.9575	372.8	54.57	59.75	3.665	53.2
		...	40	...	.438	13.124	18.66	135.28	.9394	429.1	63.44	58.64	3.665	61.3
14	14.00	XS	...	...	.500	13.000	21.21	132.73	.9217	483.8	72.09	57.46	3.665	69.1
		...	60	...	.594	12.812	24.98	128.96	.8956	562.3	85.05	55.86	3.665	80.3
		...	80	...	.750	12.500	31.22	122.72	.8522	678.3	106.13	53.18	3.665	98.2
		...	100	...	.938	12.124	38.45	115.49	.8020	824.4	130.85	50.04	3.665	117.8
		...	120	...	1.094	11.812	44.32	109.62	.7612	929.6	150.79	47.45	3.665	132.8
		...	140	...	1.250	11.500	50.07	103.87	.7213	1027.0	170.28	45.01	3.665	146.8
		...	160	...	1.406	11.188	55.63	98.31	.6827	1117.0	189.11	42.60	3.665	159.6
		...	...	5S	.165	15.670	8.21	192.85	1.3393	257.3	27.90	83.57	4.189	32.2
		...	...	10S	.188	15.624	9.34	191.72	1.3314	291.9	31.75	83.08	4.189	36.5
		...	10	...	.250	15.500	12.37	188.69	1.3103	383.7	42.05	81.74	4.189	48.0
		...	20	...	.312	15.376	15.38	185.69	1.2895	473.2	52.27	80.50	4.189	59.2
		STD	30	...	.375	15.250	18.41	182.65	1.2684	562.1	62.58	79.12	4.189	70.3
16	16.00	XS	40	...	.500	15.000	24.35	176.72	1.2272	731.9	82.77	76.58	4.189	91.5
		...	60	...	.656	14.688	31.62	169.44	1.1766	932.4	107.50	73.42	4.189	116.6
		...	80	...	.844	14.312	40.14	160.92	1.1175	1155.8	136.61	69.73	4.189	144.5
		...	100	...	1.031	13.938	48.48	152.58	1.0596	1364.5	164.82	66.12	4.189	170.5
		...	120	...	1.219	13.562	56.56	144.50	1.0035	1555.8	192.43	62.62	4.189	194.5
		...	140	...	1.438	13.124	65.78	135.28	.9394	1760.3	223.64	58.64	4.189	220.0
		...	160	...	1.594	12.812	72.10	128.96	.8956	1893.5	245.25	55.83	4.189	236.7
		...	...	5S	.165	17.670	9.25	245.22	1.7029	367.6	31.43	106.26	4.712	40.8
		...	...	10S	.188	17.624	10.52	243.95	1.6941	417.3	35.76	105.71	4.712	46.4
		...	10	...	.250	17.500	13.94	240.53	1.6703	549.1	47.39	104.21	4.712	61.1
		...	20	...	.312	17.376	17.34	237.13	1.6467	678.2	58.94	102.77	4.712	75.5
		STD	...	...	.375	17.250	20.76	233.71	1.6230	806.7	70.59	101.18	4.712	89.6
		...	30	...	.438	17.124	24.17	230.30	1.5990	930.3	82.15	99.84	4.712	103.4
18	18.00	XS	...	...	.500	17.000	27.49	226.98	1.5763	1053.2	93.45	98.27	4.712	117.0
		...	40	...	.562	16.876	30.79	223.68	1.5533	1171.5	104.67	96.93	4.712	130.1
		...	60	...	.750	16.500	40.64	213.83	1.4849	1514.7	138.17	92.57	4.712	168.3
		...	80	...	.938	16.124	50.23	204.24	1.4183	1833.0	170.92	88.50	4.712	203.8
		...	100	...	1.156	15.688	61.17	193.30	1.3423	2180.0	207.96	83.76	4.712	242.3
		...	120	...	1.375	15.250	71.81	182.66	1.2684	2498.1	244.14	79.07	4.712	277.6
		...	140	...	1.562	14.876	80.66	173.80	1.2070	2749.0	274.22	75.32	4.712	305.5
		...	160	...	1.781	14.438	90.75	163.72	1.1369	3020.0	308.50	70.88	4.712	335.6
		...	...	5S	.188	19.624	11.70	302.46	2.1004	574.2	39.78	131.06	5.236	57.4
		...	...	10S	.218	19.564	13.55	300.61	2.0876	662.8	46.06	130.27	5.236	66.3
		...	10	...	.250	19.500	15.51	298.65	2.0740	765.4	52.73	129.42	5.236	75.6
		STD	20	...	.375	19.250	23.12	290.04	2.0142	1113.0	78.60	125.67	5.236	111.3
		XS	30	...	.500	19.000	30.63	283.53	1.9690	1457.0	104.13	122.87	5.236	145.7
20	20.00	...	40	...	.594	18.812	36.15	278.00	1.9305	1703.0	123.11	120.46	5.236	170.4
		...	60	...	.812	18.376	48.95	265.21	1.8417	2257.0	166.40	114.92	5.236	225.7
		...	80	...	1.031	17.938	61.44	252.72	1.7550	2772.0	208.87	109.51	5.236	277.1
		...	100	...	1.281	17.438	75.33	238.83	1.6585	3315.2	256.10	103.39	5.236	331.5
		...	120	...	1.500	17.000	87.18	226.98	1.5762	3754.0	296.37	98.35	5.236	375.5
		...	140	...	1.750	16.500	100.33	213.82	1.4849	4216.0	341.09	92.66	5.236	421.7
		...	160	...	1.969	16.062	111.49	202.67	1.4074	4585.5	379.17	87.74	5.236	458.5
		...	...	5S	.188	21.624	12.88	367.25	2.5503	766.2	43.80	159.14	5.760	69.7
		...	...	10S	.218	21.564	14.92	365.21	2.5362	884.8	50.71	158.26	5.760	80.4
		...	10	...	.250	21.500	17.08	363.05	2.5212	1010.3	58.07	157.32	5.760	91.8
		STD	20	...	.375	21.250	25.48	354.66	2.4629	1489.7	86.61	153.68	5.760	135.4
		XS	30	...	.500	21.000	33.77	346.36	2.4053	1952.5	114.81	150.09	5.760	117.5
22	22.00	...	60	...	.875	20.250	58.07	322.06	2.2365	3244.9	197.41	139.56	5.760	295.0
		...	80	...	1.125	19.75	73.78	306.35	2.1275	4030.4	250.81	132.76	5.760	366.4
		...	100	...	1.375	19.25	89.09	291.04	2.0211	4758.5	302.88	126.12	5.760	432.6
		...	120	...	1.625	18.75	104.02	276.12	1.9175	5432.0	353.61	119.65	5.760	493.8
		...	140	...	1.875	18.25	118.55	261.59	1.8166	6053.7	403.00	113.36	5.760	550.3
		...	160	...	2.125	17.75	132.68	247.45	1.7184	6626.4	451.06	107.23	5.760	602.4

Identification, wall thickness and weights are extracted from ANSI B36.10 and B36.19. The notations STD, XS, and XXS indicate Standard, Extra Strong, and Double Extra Strong pipe respectively.

Transverse Internal Area values listed in "square feet" also represent volume in cubic feet per foot of pipe length.

CRANE

Pipe Data - Carbon and Alloy Steel - Stainless Steel

Nominal Pipe Size Inches	Outside Diameter O.D. Inches	Identification: Steel: Iron Pipe Size	Identification: Steel: Sched. No.	Identification: Stainless Steel Sched. No.	Wall Thickness (t) Inches	Inside Diameter (d) Inches	Area of Metal Square Inches	Transverse Internal Area (a) Square Inches	Transverse Internal Area (A) Square Feet	Moment of Inertia (I) Inches	Weight Pipe Pounds per foot	Weight Water Pounds per foot pipe	External Surface Sq. Ft. per foot of Pipe	Section Modulus $\left(2\frac{I}{O.D.}\right)$
24	24.00	...	...	5S	.218	23.564	16.29	436.10	3.0285	1151.6	55.37	188.98	6.283	96.0
		...	10	10S	.250	23.500	18.65	433.74	3.0121	1315.4	63.41	187.95	6.283	109.6
		STD	20	...	.375	23.250	27.83	424.56	2.9483	1942.0	94.62	183.95	6.283	161.9
		XS	...	...	.500	23.000	36.91	415.48	2.8853	2549.5	125.49	179.87	6.283	212.5
		...	30	...	.562	22.876	41.39	411.00	2.8542	2843.0	140.68	178.09	6.283	237.0
		...	40	...	.688	22.624	50.31	402.07	2.7921	3421.3	171.29	174.23	6.283	285.1
		...	60	...	.969	22.062	70.04	382.35	2.6552	4652.8	238.35	165.52	6.283	387.7
		...	80	...	1.219	21.562	87.17	365.22	2.5362	5672.0	296.58	158.26	6.283	472.8
		...	100	...	1.531	20.938	108.07	344.32	2.3911	6849.9	367.39	149.06	6.283	570.8
		...	120	...	1.812	20.376	126.31	326.08	2.2645	7825.0	429.39	141.17	6.283	652.1
		...	140	...	2.062	19.876	142.11	310.28	2.1547	8625.0	483.12	134.45	6.283	718.9
		...	160	...	2.344	19.312	159.41	292.98	2.0346	9455.9	542.13	126.84	6.283	787.9
26	26.00	...	10	...	.312	25.376	25.18	505.75	3.5122	2077.2	85.60	219.16	6.806	159.8
		STD	...	...	.375	25.250	30.19	500.74	3.4774	2478.4	102.63	216.99	6.806	190.6
		XS	20	...	.500	25.000	40.06	490.87	3.4088	3257.0	136.17	212.71	6.806	250.5
28	28.00	...	10	...	.312	27.376	27.14	588.61	4.0876	2601.0	92.26	255.07	7.330	185.8
		STD	...	...	.375	27.250	32.54	583.21	4.0501	3105.1	110.64	252.73	7.330	221.8
		XS	20	...	.500	27.000	43.20	572.56	3.9761	4084.8	146.85	248.11	7.330	291.8
		...	30	...	.625	26.750	53.75	562.00	3.9028	5037.7	182.73	243.53	7.330	359.8
30	30.00	...	...	5S	.250	29.500	23.37	683.49	4.7465	2585.2	79.43	296.18	7.854	172.3
		...	10	10S	.312	29.376	29.10	677.76	4.7067	3206.3	98.93	293.70	7.854	213.8
		STD	...	...	.375	29.250	34.90	671.96	4.6664	3829.4	118.65	291.18	7.854	255.3
		XS	20	...	.500	29.000	46.34	660.52	4.5869	5042.2	157.53	286.22	7.854	336.1
		...	30	...	.625	28.750	57.68	649.18	4.5082	6224.0	196.08	281.31	7.854	414.9
32	32.00	...	10	...	.312	31.376	31.06	773.19	5.3694	3898.9	105.59	335.05	8.378	243.7
		STD	...	...	.375	31.250	37.26	766.99	5.3263	4658.5	126.66	332.36	8.378	291.2
		XS	20	...	.500	31.000	49.48	754.77	5.2414	6138.6	168.21	327.06	8.378	383.7
		...	30	...	.625	30.750	61.60	742.64	5.1572	7583.4	209.43	321.81	8.378	474.0
		...	40	...	.688	30.624	67.68	736.57	5.1151	8298.3	230.08	319.18	8.378	518.6
34	34.00	...	10	...	.312	33.376	33.02	874.9	6.075695	4684.65	112.36	378.9	8.901	275.6
		STD	...	...	.375	33.250	39.61	868.31	6.0299	5599.3	134.67	376.27	8.901	329.4
		XS	20	...	.500	33.000	52.62	855.30	5.9396	7383.5	178.89	370.63	8.901	434.3
		...	30	...	.625	32.750	65.53	842.39	5.8499	9127.6	222.78	365.03	8.901	536.9
		...	40	...	.688	32.624	72.00	835.92	5.8050	9991.6	244.77	362.23	8.901	587.7
36	36.00	...	10	...	.312	35.376	34.98	982.90	6.8257	5569.5	118.92	425.92	9.425	309.4
		STD	...	...	.375	35.250	41.97	975.91	6.7771	6658.9	142.68	422.89	9.425	369.9
		XS	20	...	.500	35.000	55.76	962.11	6.6813	8786.2	189.57	416.91	9.425	488.1
		...	30	...	.625	34.750	69.46	948.42	6.5862	10868.4	236.13	417.22	9.425	603.8
		...	40	...	.750	34.500	83.06	934.82	6.4918	12906.1	282.35	405.09	9.425	717.0

Identification, wall thickness and weights are extracted from ANSI B36.10 and B36.19. The notations STD, XS, and XXS indicate Standard, Extra Strong, and Double Extra Strong pipe respectively.

Transverse Internal Area values listed in "square feet" also represent volume in cubic feet per foot of pipe length.

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VG-AG-TB-EN-L13-00-0911