

PLAINTIFF'S EXHIBIT

CRA-51

FLOW OF FLUIDS THROUGH VALVES, FITTINGS, AND PIPE

FLOW OF FLUIDS

THROUGH

VALVES, FITTINGS, AND PIPE

By the Engineering Division

Crane Co., Chicago

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FOREWORD

The more complex industry becomes, the more vital becomes the role played by fluids in the industrial machine. One hundred years ago 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. The age of atomic energy and rocket power has added fluids such as liquid metals . . . i.e., sodium, potassium, and bismuth, as well as liquid oxygen, nitrogen, etc. . . to the list of more common fluids such as oil, water, gases, acids, and liquors that are being transported in pipe today. Nor is the transportation of fluids the only phase of hydraulics which warrants attention now. Hydraulic and pneumatic mechanisms are used extensively for the controls of modern aircraft, sea-going vessels, automotive equipment, machine tools, earth-moving and road-building machines, and even in scientific laboratory equipment where precise control of fluid flow is required.

So extensive are the applications of hydraulics and fluid mechanics that almost every engineer has found it necessary to familiarize himself with 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. published in 1935, 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. That this work was immediately acclaimed by both practical and scientific engineers was indicated by its world-wide circulation and the many favorable comments received. The appearance of much new and useful data on the subject has made the present revision necessary. In order to obtain the latest information, Crane Co. sponsored a fellowship at Armour Research Foundation under the direction of Dr. V. L. Streeter. Mr. W. G. Kautz, recipient of the fellowship, com-

plied much of the data upon which the present revision is based. Development and arrangement of these data to the form in which they are presented, as well as the analysis of extensive Crane Laboratory and field test data, required the extensive effort of Crane Engineers for a period of several years.

The chief endeavor in the preparation of the present edition has been to present the newest available information on the flow of fluids, in summarized form, and to include all the auxiliary data necessary to the solution of any but the most unusual fluid flow problems. As in the 1942 edition, nomographs are included for the use of those engineers who prefer graphical methods of solving some of the more simple problems. However, the number of nomographs has been considerably reduced by avoiding duplication; at the same time, their range of application has been expanded.

The general arrangement of the book has been changed to present the theory in Chapters 1 and 2 . . . practical application to flow problems in Chapters 3 and 4 . . . physical properties of fluids and flow characteristics of valves, fittings, and pipe in Appendix A . . . and conversion units and other useful engineering data in Appendix B. Furthermore, a page numbering system has been adopted that will facilitate future revisions of individual chapters without affecting others.

Most of the data on flow through valves and fittings were obtained by carefully conducted experiments in the Crane Engineering Laboratories. Liberal use has been made, however, of other reliable sources of data on this subject and due credit has been given these sources in the text. The bibliography of references will provide a source for further study of the subject presented.

CRANE CO.

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Nomenclature

Nomenclature	see next page
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Nomenclature

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

A = cross sectional area of pipe or orifice, in square feet
 a = cross sectional area of pipe or orifice, in square inches
 B = rate of flow in barrels (42 gallons) per hour
 C = flow coefficient for orifices and nozzles = discharge coefficient corrected for velocity of approach = $C_d / \sqrt{1 - (d_0/d_1)^4}$
 C_d = discharge coefficient for orifices and nozzles
 C_v = flow coefficient for valves; expresses flow rate in gallons per minute of 60 F water with 1.0 psi pressure drop across valve = $Q \sqrt{\rho / (62.4 \Delta P)}$
 D = internal diameter of pipe, in feet
 d = internal diameter of pipe, in inches
 e = base of natural logarithm = 2.718
 f = friction factor in formula $h_L = f L v^2 / D 2g$
 g = acceleration of gravity = 32.2 feet per second per second
 H = total head, in feet of fluid
 h = static pressure head existing at a point, in feet of fluid
 h_g = total heat of steam, in Btu per pound
 h_L = loss of static pressure head due to fluid flow, in feet of fluid
 h_w = static pressure head, in inches of water
 K = resistance coefficient or velocity head loss in the formula, $h_L = K v^2 / 2g$
 k = ratio of specific heat at constant pressure to specific heat at constant volume = c_p / c_v
 L = length of pipe, in feet
 L/D = equivalent length of a resistance to flow, in pipe diameters
 L_m = length of pipe, in miles
 M = molecular weight
 MR = universal gas constant = 1544
 n = exponent in equation for polytropic change ($p' V^n = \text{constant}$)
 P = pressure, in pounds per square inch gauge
 P' = pressure, pounds per square inch absolute (see page 1-5 for diagram showing relationship between gauge and absolute pressure)
 p' = pressure, in pounds per square foot absolute
 Q = rate of flow, in gallons per minute
 q = rate of flow, in cubic feet per second at flowing conditions
 q' = rate of flow, in cubic feet per second at standard conditions (14.7 psia and 60F)
 q'_d = rate of flow, in millions of standard cubic feet per day, MMscfd
 q'_h = rate of flow, in cubic feet per hour at standard conditions (14.7 psia and 60F), scfh
 q_m = rate of flow, in cubic feet per minute at flowing conditions
 q'_m = rate of flow, in cubic feet per minute at std. conditions (14.7 psia and 60F), scfm

R = individual gas constant = $MR/M = 1544/M$
 R_e = Reynolds number
 r_c = critical pressure ratio for compressible flow
 S = specific gravity of liquids relative to water, both at standard temperature (60 F)
 S_g = specific gravity of a gas relative to air = the ratio of the molecular weight of the gas to that of air
 T = absolute temperature, in degrees Rankine ($460 + t$)
 t = temperature, in degrees Fahrenheit
 $\bar{V}$ = specific volume of fluid, in cubic feet per pound
 V = mean velocity of flow, in feet per minute
 V_a = volume, in cubic feet
 v = mean velocity of flow, in feet per second
 v_s = sonic (or critical) velocity of flow of a gas, in feet per second
 W = rate of flow, in pounds per hour
 w = rate of flow, in pounds per second
 w_a = weight, in pounds
 x = percent quality of steam = 100 minus per cent of moisture
 Y = net expansion factor for compressible flow through orifices, nozzles, or pipe
 Z = potential head or elevation above reference level, in feet

Subscripts

- (o) . . indicates orifice conditions unless otherwise specified
- (1) . . indicates inlet or upstream conditions unless otherwise specified
- (2) . . indicates outlet or downstream conditions unless otherwise specified
- (100) . refers to 100 feet of pipe

Greek Letters

Delta
 Δ = differential between two points
Epsilon
 ϵ = absolute roughness or effective height of pipe wall irregularities, in feet
Rho
 ρ = weight density of fluid, pounds per cubic ft
 ρ' = density of fluid, grams per cubic centimeter
Mu
 μ = absolute (dynamic) viscosity, in centipoise
 μ_e = absolute viscosity, in pound mass per foot second or poundal seconds per sq foot
 μ'_e = absolute viscosity, in slugs per foot second or pound force seconds per square foot
Nu
 ν = kinematic viscosity, in centistokes
 ν' = kinematic viscosity, square feet per second

Theory of Flow In Pipe

CHAPTER 1

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 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 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 C.G.S. (centimeter, gram, second) or 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:

$$\begin{aligned}\mu &= 1 \text{ centipoise}^* = \begin{cases} 0.01 \text{ poise} \\ 0.01 \text{ gram per cm second} \\ 0.01 \text{ dyne second per sq cm} \end{cases} \\ \mu_c &= \begin{cases} 0.000672 \text{ pound mass per foot second} \\ 0.000672 \text{ poundal second per square foot} \end{cases} \\ \mu'_c &= \begin{cases} 0.0000209 \text{ slug per foot second} \\ 0.0000209 \text{ pound force second per square ft} \end{cases}\end{aligned}$$

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.

$$\nu \text{ (centistokes)} = \frac{\mu \text{ (centipoise)}}{\rho' \text{ (grams per cubic cm)}} = \frac{\mu}{S}$$

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 in this country is the Saybolt Universal Viscosimeter. 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 viscosimeters, somewhat similar to the Saybolt but not used to any extent in this country, are 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 ASTM standard viscosity temperature chart for liquid petroleum products, reproduced on page B-6, is used to determine the Saybolt Universal viscosity of a petroleum product at any temperature when the viscosities at two different temperatures are known. The viscosities of some of the most common fluids are given on pages A-2 to A-5. 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 perfect 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 1.003 centipoise.

Physical Properties of Fluids — continued

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 $\bar{V}$, 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:

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

Computations in the metric system are not commonly referred to in terms of specific volume; however, the number of cubic centimeters per gram of a substance can readily be expressed as the reciprocal of the weight density, that is:

$$\frac{1}{\rho'}$$

The variations in weight density as well as other properties of water with changes in temperature are shown on page A-6. The weight densities of other common liquids are shown on page A-7. 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 the so-called "perfect" gases, the weight density can be computed from the formula:

$$\rho = \frac{144 P'}{RT}$$

The individual gas constant R is equal to the universal gas constant, $MR = 1544$, divided by the molecular weight of the gas,

$$R = \frac{1544}{M}$$

Values of R , as well as other useful gas constants, are given on page A-8. The weight density of air for various conditions of temperature and pressure can be found on page A-10.

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-18. A chart for determining the weight density and specific volume of gases is given on page A-11.

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 its weight density at 60 F (unless otherwise specified) to that of water at standard temperature, 60 F.

$$S = \frac{\rho}{\rho_{\text{water at 60 F}}} \left\{ \begin{array}{l} \text{any liquid at 60 F,} \\ \text{unless otherwise specified} \end{array} \right.$$

A hydrometer can be used to measure the specific gravity of liquids directly. Three hydrometer scales are common in this country . . . 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,

$$S (60 \text{ F}/60 \text{ F}) = \frac{141.5}{131.5 + \text{deg. API}}$$

For liquids lighter than water,

$$S (60 \text{ F}/60 \text{ F}) = \frac{140}{130 + \text{deg. Baumé}}$$

For liquids heavier than water,

$$S (60 \text{ F}/60 \text{ F}) = \frac{145}{145 - \text{deg. Baumé}}$$

For convenience in converting hydrometer readings to more useful units, refer to the table shown on page B-7.

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

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

Nature of Flow in Pipe — Laminar and Turbulent

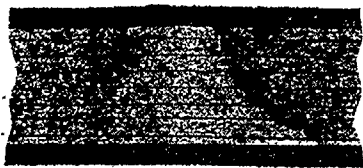


Figure 1-1
Laminar Flow

Actual photograph of colored filaments being carried along undisturbed by a stream of water.



Figure 1-2
Flow in Critical Zone, Between Laminar and Transition Zones

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:

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

(For nomenclature, see page preceding Chapter 1)

"Reasonable" velocities for use in design work are given on pages 3-6 and 3-16.

Reynolds number: The work of Osborne Reynolds has shown that the nature of flow in pipe . . . that is, 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:

$$R_e = \frac{Dv\rho}{\mu_e} \quad \text{Equation 1-2}$$

(For other forms of this equation, see page 3-2.)

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.

Hydraulic radius: Occasionally a conduit of non-circular cross section is encountered. In calculating the Reynolds number for this condition, the equivalent diameter (four times the hydraulic radius) is substituted for the circular diameter. Use friction factors given on pages A-24 and A-25.

$$\text{hydraulic radius} = \frac{\text{area of flowing fluid}}{\text{wetted perimeter}}$$

This applies to any ordinary conduit (circular conduit not flowing full, oval, square, or rectangular) but not to extremely narrow shapes such as annular or elongated openings, where width is small relative to length. In such cases, the hydraulic radius is approximately equal to the width of the passage.

(Velocity and pressure drop calculations are based on actual flow area.)

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:

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

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:

$$Z_1 + \frac{144 P_1}{\rho_1} + \frac{v_1^2}{2g} = Z_2 + \frac{144 P_2}{\rho_2} + \frac{v_2^2}{2g} + h_L \quad \text{Equation 1-3}$$

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

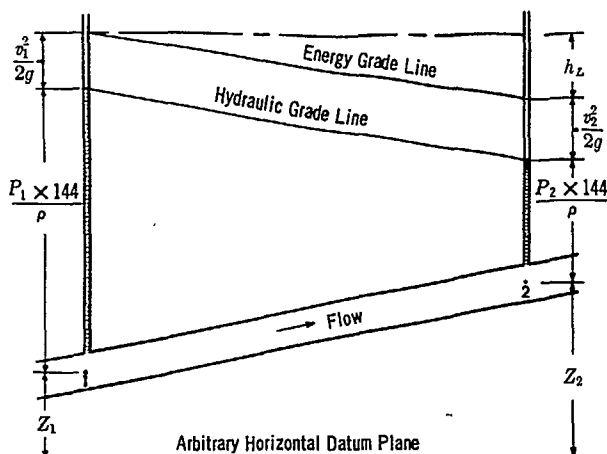


Figure 1-4
Energy Balance for Two Points in a Fluid

By permission, from *Fluid Mechanics** by R. A. Dodge and M. J. Thompson. Copyright 1937; McGraw-Hill Book Company, Inc.

Measurement of Pressure

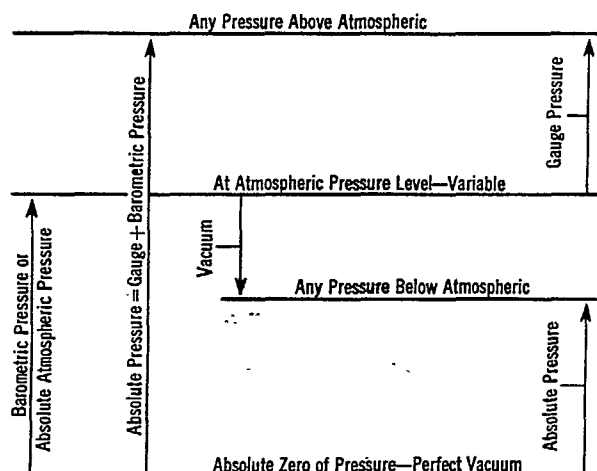


Figure 1-5
Relationship Between
Gauge and Absolute Pressures

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.

*All superior figures used as reference marks refer to the Bibliography; see page C-1.

Darcy's Formula General Equation for Flow of Fluids

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 .

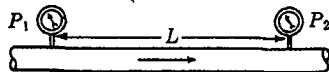


Figure 1-6

The general equation for pressure drop, known as Darcy's formula and expressed in feet of fluid, is $h_L = fLv^2/D 2g$. This equation may be written to express pressure drop in pounds per square inch, by substitution of proper units, as follows:

$$\Delta P = \frac{\rho f L v^2}{144 D 2g} \quad \text{Equation 1-4}$$

(For other forms of this equation, see page 3-2.)

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 1-7.

Equation 1-4 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 (page 1-5). For an example using this theorem, see page 4-8.

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 1-4 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:

$$f = \frac{64}{R_e} = \frac{64 \mu_e}{D v \rho} = \frac{64 \mu}{124 d v \rho}$$

If this quantity is substituted into Equation 1-4, the pressure drop in pounds per square inch is:

$$\Delta P = 0.000668 \frac{\mu L v}{d^2} \quad \text{Equation 1-5}$$

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 (ϵ), as compared to the diameter of the pipe (D). 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. Moody¹⁸ and are reproduced on pages A-23 to A-25. Professor Moody improved upon the well-established Pigott and Kemler friction factor diagram, incorporating more recent investigations and developments of many outstanding scientists.

The friction factor, f , is plotted on page A-24 on the basis of relative roughness obtained from the chart on page A-23 and the Reynolds number. The

Darcy's Formula

General Equation for Flow of Fluids — continued

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-23. Since most calculations involve commercial steel or wrought iron pipe, the chart on page A-25 is furnished for a more direct solution. It should be kept in mind that these figures apply to clean new pipe.

Effect of age and use on pipe friction: Friction loss in pipe is sensitive to changes in diameter and roughness of pipe. For a given rate of flow and a fixed friction factor, the pressure drop per foot of pipe varies inversely with the fifth power of the diameter. Therefore, a 2% reduction of diameter

causes a 10% increase in pressure drop; a 5% reduction of diameter increases pressure drop 23%. In many services, the interior of pipe becomes encrusted with scale, dirt, tubercles or other foreign matter; thus, it is often prudent to make allowance for expected diameter changes.

Authorities² point out that roughness may be expected to increase with use (due to corrosion or incrustation) at a rate determined by the pipe material and nature of the fluid. Ippen¹⁸, in discussing the effect of aging, cites a 4-inch galvanized steel pipe which had its roughness doubled and its friction factor increased 20% after three years of moderate use.

Principles of Compressible Flow in Pipe

An accurate determination of the pressure drop of a compressible fluid flowing through a pipe requires a knowledge of the relationship between pressure and specific volume; this is not easily determined in each particular problem. The usual extremes considered are adiabatic flow ($p'V_a^k = \text{constant}$) and isothermal flow ($p'V_a = \text{constant}$). Adiabatic flow is usually assumed in short, perfectly insulated pipe. 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.

Isothermal flow or flow at constant temperature is often assumed, partly for convenience but more often because it is closer to fact in piping practice. The most outstanding case of isothermal flow occurs in natural gas pipe lines. Dodge and Thompson¹ show that gas flow in insulated pipe is closely approximated by isothermal flow for reasonably high pressures.

Since the relationship between pressure and volume may follow some other relationship ($p'V_a^n = \text{constant}$) called polytropic flow, specific information in each individual case is almost an impossibility.

The density of gases and vapors changes considerably

with changes in pressure; therefore, if the pressure drop between P_1 and P_2 in Figure 1-6 is great, the density and velocity will change appreciably.

When dealing with compressible fluids, such as air, steam, etc., the following restrictions should be observed in applying the Darcy formula:

1. If the calculated pressure drop ($P_1 - P_2$) is less than about 10% of the inlet pressure P_1 , reasonable accuracy will be obtained if the specific volume used in the formula is based upon either the upstream or downstream conditions, whichever are known.
2. If the calculated pressure drop ($P_1 - P_2$) is greater than about 10%, but less than about 40% of inlet pressure P_1 , the Darcy equation may be used with reasonable accuracy by using a specific volume based upon the average of upstream and downstream conditions; otherwise, the method given on page 1-9 may be used.
3. For greater pressure drops, such as are often encountered in long pipe lines, the methods given on the next two pages should be used.

(continued on the next page)

Principles of Compressible Flow in Pipe

(continued)

Complete isothermal equation: The flow of gases in long pipe lines 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:

$$w = \left[\frac{144 \text{ g } A^2}{V_1 \left(\frac{fL}{D} + 2 \log_e \frac{P_1}{P_2} \right)} \right] \left[\frac{(P_1)^2 - (P_2)^2}{P_1} \right] \quad \text{Equation 1-6}$$

The formula is developed on the basis of these assumptions:

1. Isothermal flow.
2. No mechanical work is done on or by the system.
3. Steady flow or discharge unchanged with time.
4. The gas 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 pipe line is straight and horizontal between end points.

Simplified Compressible Flow—Gas Pipe Line Formula: In the practice of gas pipe line engineering, another assumption is added to the foregoing:

8. Acceleration can be neglected because the pipe line is long.

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

$$w^2 = \left[\frac{144 \text{ g } DA^2}{V_1 fL} \right] \left[\frac{(P_1)^2 - (P_2)^2}{P_1} \right] \quad \text{Equation 1-7}$$

This is equivalent to the complete isothermal equation if the pipe line is 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 1-7 as follows:

$$q'_h = 114.2 \sqrt{\left[\frac{(P_1)^2 - (P_2)^2}{f L_m T S_g} \right] d^5} \quad \text{Equation 1-7a}$$

Other commonly used formulas for compressible flow in long pipe lines:

Weymouth formula²⁴:

Equation 1-8

$$q'_h = 28.0 d^{2.667} \sqrt{\left[\frac{(P_1)^2 - (P_2)^2}{S_g L_m} \right] \frac{520}{T}}$$

Panhandle formula³ for natural gas pipe lines 6 to 24-inch diameter, Reynolds numbers 5×10^6 to 14×10^6 , and $S_g = 0.6$:

$$q'_h = 36.8 E d^{2.6182} \left[\frac{(P_1)^2 - (P_2)^2}{L_m} \right]^{0.5394} \quad \text{Equation 1-9}$$

The flow efficiency factor E is defined as an experience factor and is usually assumed to be 0.92 or 92% for average operating conditions. Suggested values for E for other operating conditions are given on page 3-3.

Comparison of formulas for compressible flow in pipe lines: Equations 1-7, 1-8, and 1-9 are derived from the same basic formula, but differ in the selection of data used for the determination of the friction factors.

Friction factors in accordance with the Moody¹⁸ diagram are normally used with the Simplified Compressible Flow formula (Equation 1-7). However, if the same friction factors employed in the Weymouth or Panhandle formulas are used in the Simplified formula, identical answers will be obtained.

The Weymouth friction factor²⁴ is defined as:

$$f = \frac{0.032}{d^{1/3}}$$

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-inch, and smaller for sizes larger than 20-inch.

The Panhandle friction factor³ is defined as:

$$f = 0.1225 \left(\frac{d}{q'_h S_g} \right)^{0.1461}$$

In the flow range to which the Panhandle formula is limited, this results in friction factors that are lower than those obtained from either the Moody data or the Weymouth friction formula. As a result, flow rates obtained by solution of the Panhandle formula are usually greater than those obtained by employing either the Simplified Compressible Flow formula with Moody friction factors, or the Weymouth formula.

An example of the variation in flow rates which may be obtained for a specific condition by employing these formulas is given on page 4-11.

Principles of Compressible Flow in Pipe

(continued)

Limiting flow of gases and vapors: The feature not evident in the preceding formulas (Equations 1-4 and 1-6 to 1-9 inclusive) is that the weight rate of flow (e.g., lbs/sec) of a compressible fluid in a pipe, with a given upstream pressure, will approach a certain maximum rate which it cannot exceed, no matter how much the downstream pressure is further reduced.

The maximum velocity of a compressible fluid in pipe is limited by the velocity of propagation of a pressure wave which travels at the speed of sound in the fluid. Since pressure falls off and velocity increases as fluid proceeds downstream in 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 velocity of sound. Further decrease in the outlet pressure will not be felt upstream because the pressure wave can only travel at sonic velocity, 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 in the pipe is sonic velocity, which is expressed as:

$$v_s = \sqrt{kg RT} = \sqrt{kg 144 P' V}$$

Equation 1-10

The value of k , the ratio of specific heats at constant pressure to constant volume, is 1.4 for most diatomic gases; see pages A-8 and A-9 for values of k for gases and steam respectively. This velocity will occur at the outlet end or in a constricted area, when the pressure drop is sufficiently high. The pressure, temperature, and specific volume 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. This assumption is supported by experimental data on pipe having lengths of 220 and 130 pipe diameters discharging air to atmosphere. Investigation of the complete theoretical analysis of adiabatic flow¹⁹ has led to a basis for establishing correction factors, which may be applied to the Darcy equation for this condition of flow. Since these correction factors compensate for the changes in fluid properties due to expansion of the fluid, they are identified as Y net expansion factors; see page A-22.

The Darcy formula, including the Y factor, is:

$$w = 0.525 Y d^2 \sqrt{\frac{\Delta P}{KV_1}} \quad \text{Equation 1-11}$$

(Resistance coefficient K is defined on page 2-8)

It should be noted that the value of K in this equation is the total resistance coefficient of the pipe line, 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-22, 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 1-11, whenever the Y factor falls within the limits defined by the resistance factor K curves in the charts on page A-22. 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-22, must be used in Equation 1-11.

Application of Equation 1-11 and the determination of values for K , Y , and ΔP in the formula is demonstrated in examples on pages 4-13 and 4-14.

The charts on page A-22 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 approximately follow the perfect gas laws. Steam and vapors deviate from the perfect gas laws, and application of the Y factor obtained from the charts to these flows, will therefore yield flow rates slightly greater (up to about 5%) than those calculated on the basis of sonic velocity at the outlet. However, greater accuracy will be obtained if the charts are used to establish the downstream pressure when sonic velocity occurs, and the fluid properties at this pressure condition are used in the sonic velocity and continuity equations (Equations 3-8 and 3-2 respectively) to determine the flow rate. An example of this type of flow problem is presented on page 4-13.

This condition of flow is comparable to the flow through nozzles and venturi tubes, covered on page 2-15, and the solutions of such problems are similar.

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 180.1 Btu. This quantity of heat is called *heat of the liquid* or *sensible heat*. In order to

change the liquid into a vapor at atmospheric pressure (14.7 psia), 970.3 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*. Consequently, the *total heat* of the vapor, formed when water boils at atmospheric pressure, is the sum of the two quantities . . . 180.1 Btu and 970.3 Btu, or, 1150.4 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.

Flow of Fluids Through Valves and Fittings

CHAPTER 2

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 on pages A-26, A-27, and A-30. For conversion of "equivalent length in pipe diameters", as obtained from page A-27 or A-30, to "equivalent length in feet of pipe" for any size of valve or fitting, see page A-31. The chart on page A-31 also illustrates the correlation of equivalent length, resistance coefficient K , and pipe size. A chart is presented on page A-32 which may be used to readily determine the C_v flow coefficient of any valve for which the resistance coefficient is known or can be determined from page A-30 and page A-31.

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 2-8 and 2-9.

Types of Valves and Fittings Used in Pipe Systems

Valves: Although the great variety of valve designs precludes any thorough classification, most of the designs may be considered as modifications of the two basic types:

1. the gate type
2. the globe type

If valves were classified according to the resistance which they offer to flow, the gate type valves would be put in the low resistance class and the globe type valves in the high resistance class. The classification is not all-inclusive, however, because a large number of modified valve types fall between the two extremes. Some of the most commonly used valve designs are illustrated on pages A-28 and A-29.

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, of course, 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 Chargeable 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 pipe line 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.

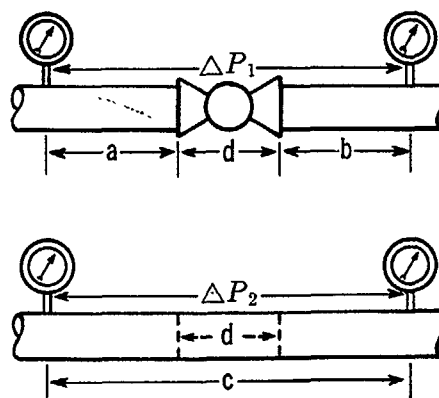


Figure 2-1

Figure 2-1 shows two sections of a pipe line 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 equivalent length in pipe diameters, of various valves and fittings as given on page 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 2-3 is unique in that 6-inch gate, globe, and angle valves or 90 degree ells 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 per 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 2-4 to 2-7 shown on the two pages following.

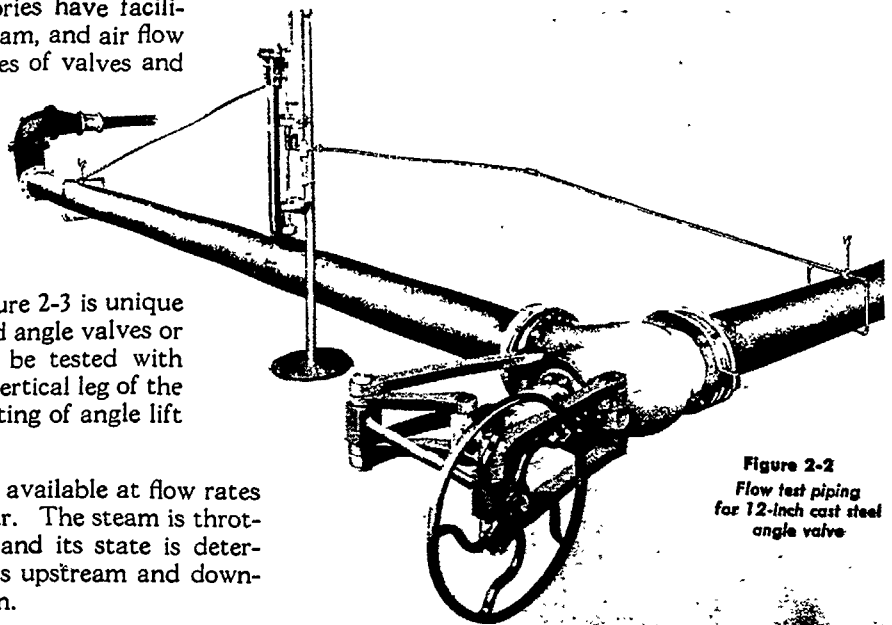


Figure 2-2
Flow test piping
for 12-inch cast steel
angle valve

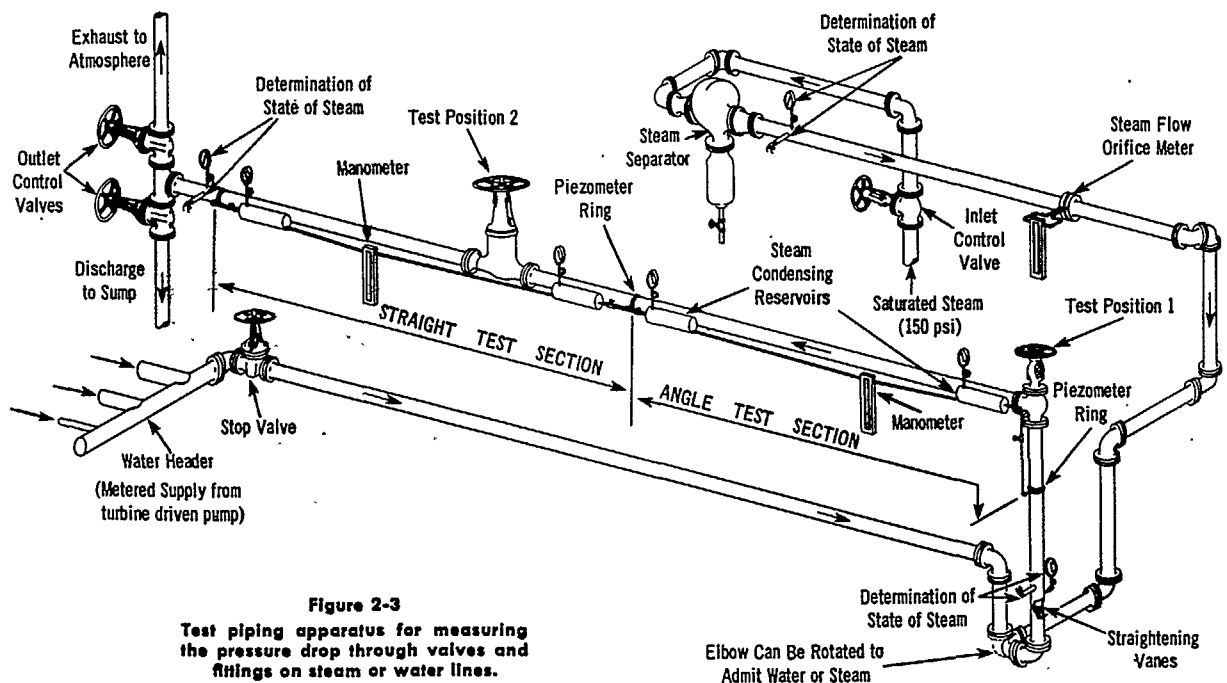


Figure 2-3
Test piping apparatus for measuring
the pressure drop through valves and
fittings on steam or water lines.

Crane Water Flow Tests

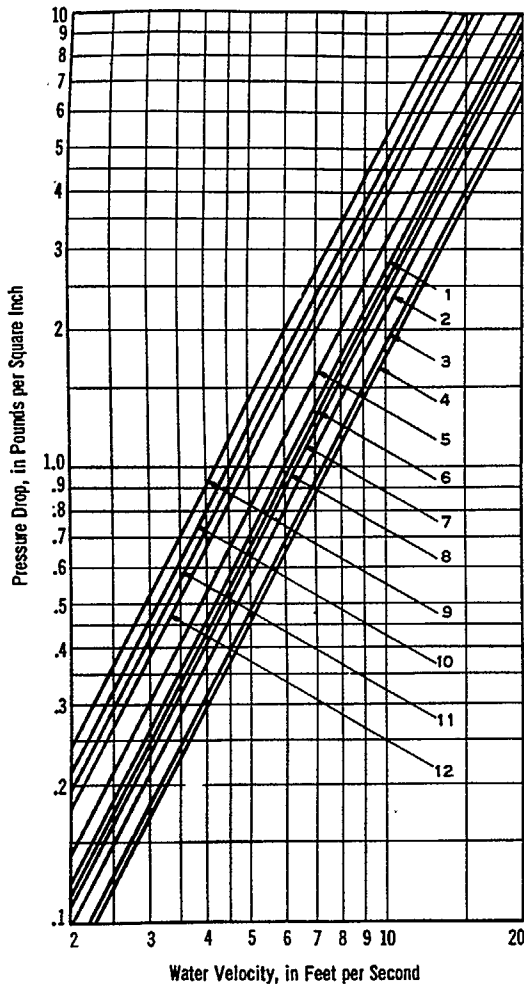


Figure 2-4

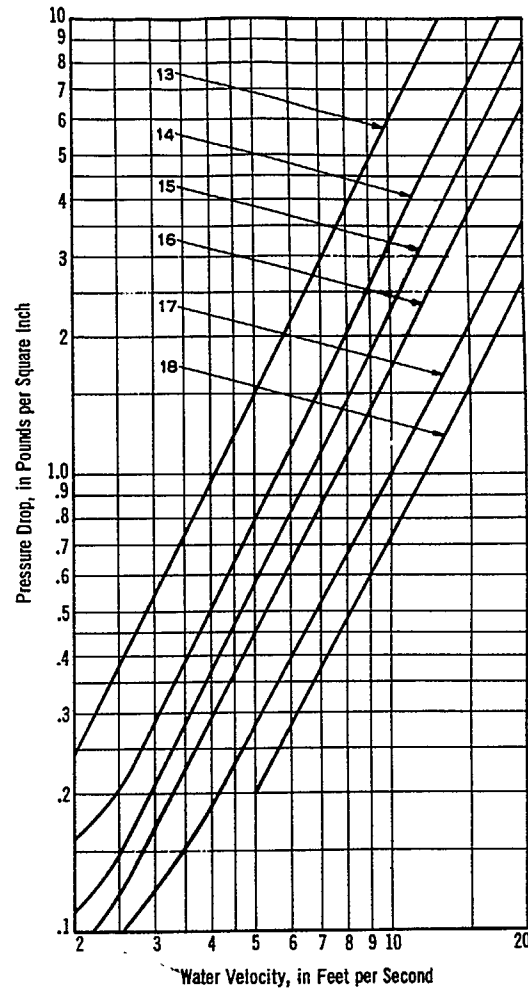


Figure 2-5

Water Flow Tests — Curves 1 to 18

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

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

Crane Steam Flow Tests

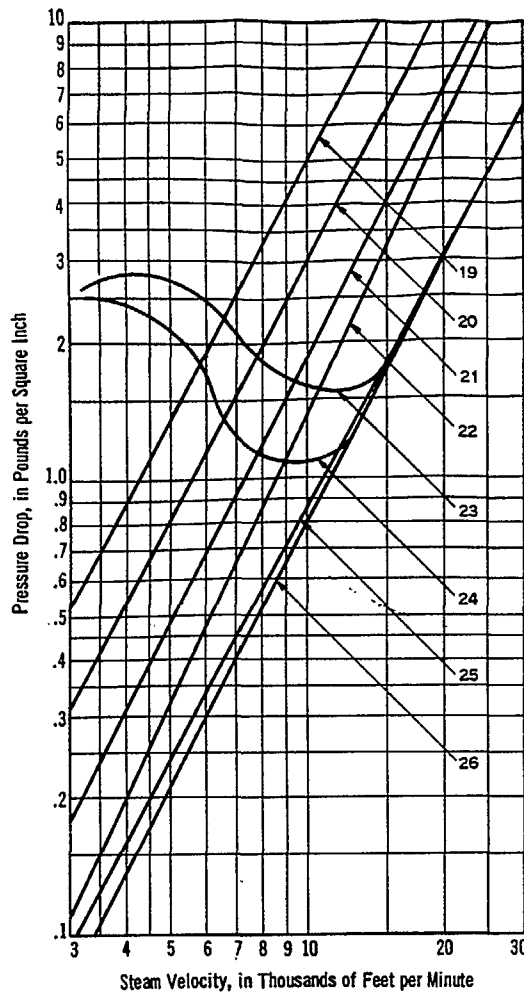


Figure 2-6

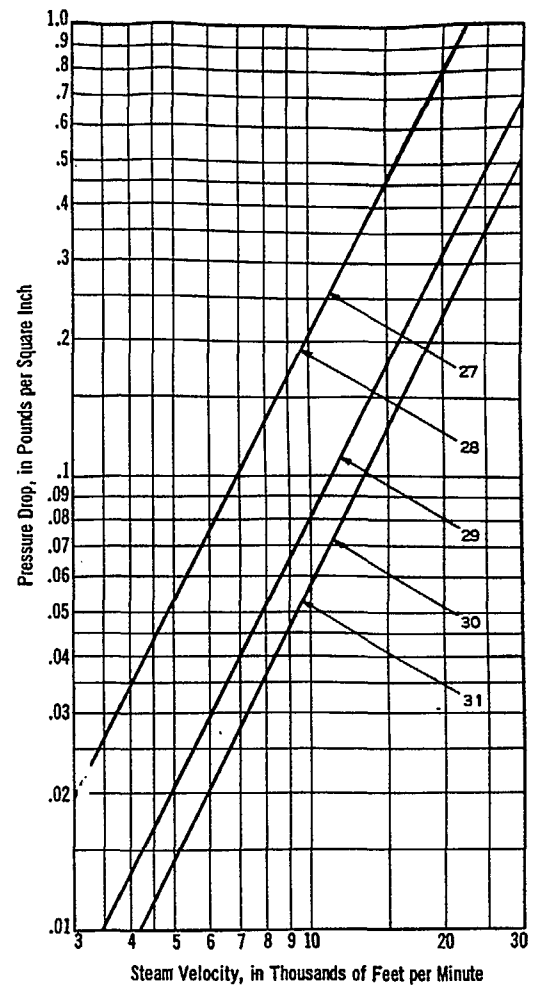


Figure 2-7

Steam Flow Tests — Curves 19 to 31

Fluid	Figure No.	Curve No.	Size, Inches	Valve* or Fitting Type
Saturated Steam 50 psi gauge	Figure 2-6	19	2	300-Pound Brass Conventional Globe Valve..... Plug Type Seat
		20	6	300-Pound Steel Conventional Globe Valve..... Plug Type Seat
		21	6	300-Pound Steel Angle Valve..... Plug Type Seat
		22	6	300-Pound Steel Angle Valve..... Ball to Cone Seat
		23	6	600-Pound Steel Angle Stop-Check Valve
		24	6	600-Pound Steel Y-Pattern Globe Stop-Check Valve
		25	6	600-Pound Steel Angle Valve
		26	6	600-Pound Steel Y-Pattern Globe Valve
	Figure 2-7	27	2	90° Short Radius Elbow for Use with Schedule 40 Pipe
		28	6	250-Pound Cast Iron Flanged Conventional 90° Elbow
		29	6	600-Pound Steel Gate Valve
		30	6	125-Pound Cast Iron Gate Valve
		31	6	150-Pound Steel Gate Valve

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

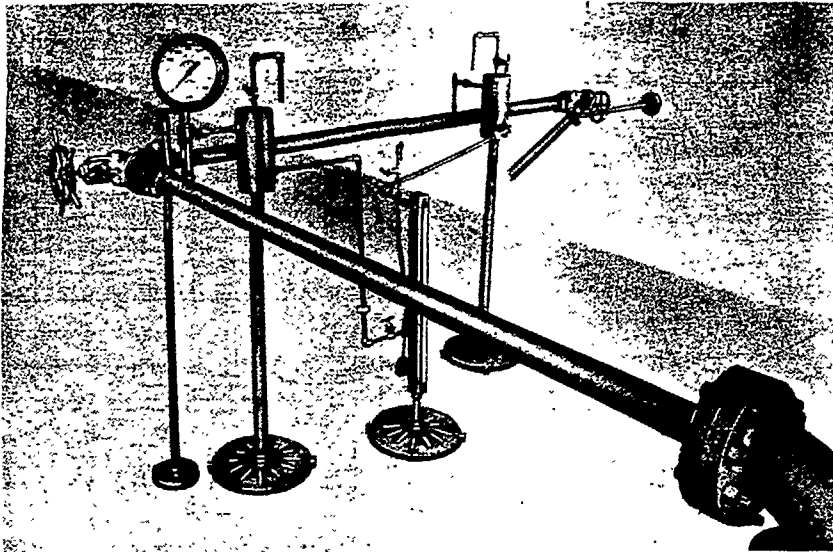


Figure 2-8
Flow test piping
for 2½-inch cast
steel angle valve.

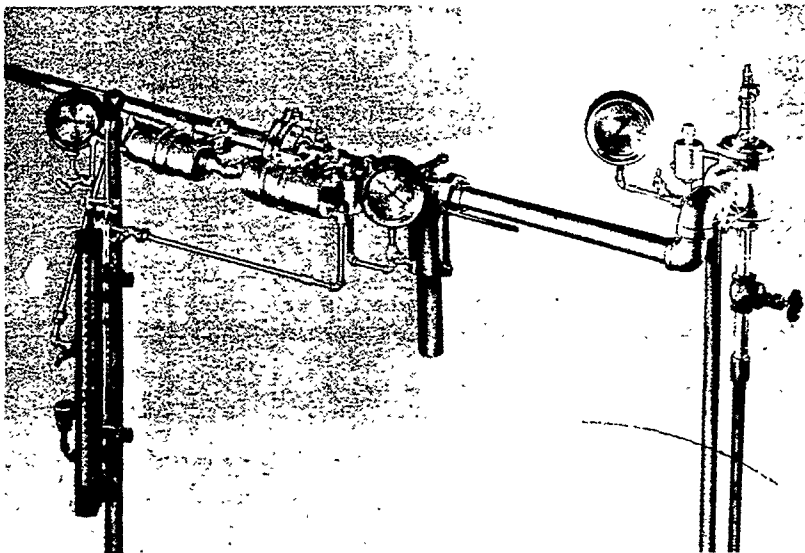


Figure 2-9
Steam capacity test
of a ½-inch brass
relief valve.

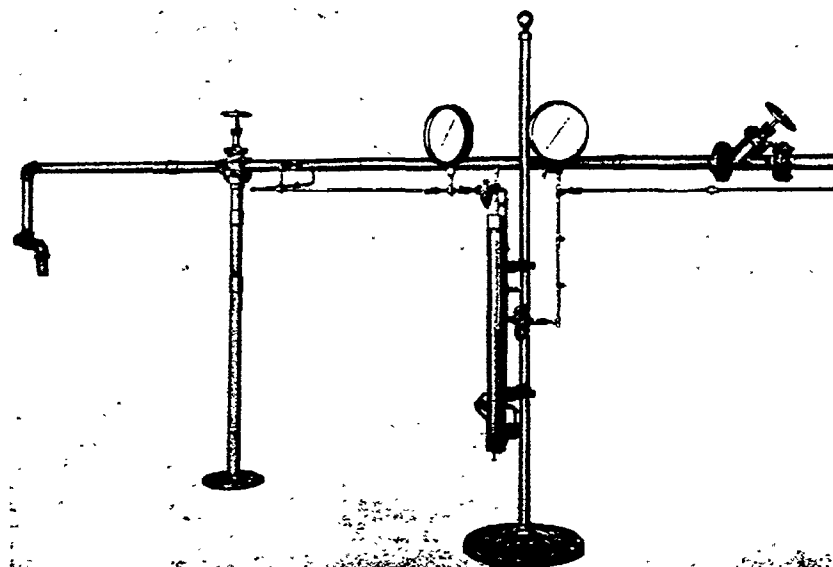


Figure 2-10
Flow test piping for
2-inch fabricated steel
y-pattern globe valve.

Relationship of Pressure Drop to Velocity of Flow

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 v 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 2-5 and 2-6, 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 2-6, it will be noted that the pressure drop, at the point where the two curves representing check valves deviate from a straight line, is about $1\frac{1}{2}$ to 2 pounds per square inch. This value will vary somewhat for different valve designs depending upon the relative weight and size of the disc; however, it has been found to be a good "rule of thumb" to size check valves so that the pressure drop in the fully open position is about 2 psi in lift checks and about $\frac{1}{2}$ psi in swing checks. This rule applies only to check valves designed on the basis of established fundamental considerations which assure a

full disc lift at low flow rates. On some poorly designed lift check valves, tests have shown that the disc will not lift fully even at extremely high flow rates. In many cases, application of this rule will result in check valves smaller in size than the pipe line; however, the actual pressure drop will be little, if any, higher than that of a full size valve which is used in other than a wide open position.

The losses due to sudden contraction and enlargement which will occur in such an installation with bushings or reducing flanges can be readily calculated from the data given on page A-26. If tapered reducers are used, the loss due to gradual contraction at the inlet to the smaller size valve is partially compensated for by the corresponding gradual enlargement on the outlet side, so that the added pressure drop due to these effects is minor.

In-line ball check valves of the design shown in Figure 2-11 should be installed in a horizontal position wherever possible. In this position, the flow required to move the disc to the fully open position is very low and the valves can be full size to match the pipe line; this will result in low pressure drop at all flow rates. If it is necessary to install this type of valve in a vertical line, due to piping arrangement or for other reasons, it should be sized so that the flow rate will be sufficient to cause a pressure drop of about $2\frac{1}{2}$ psi across the valve. This will provide full disc lift and prevent noisy operation and premature wear of parts.

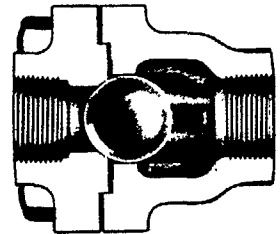


Figure 2-11
In-line ball check valve
in horizontal position

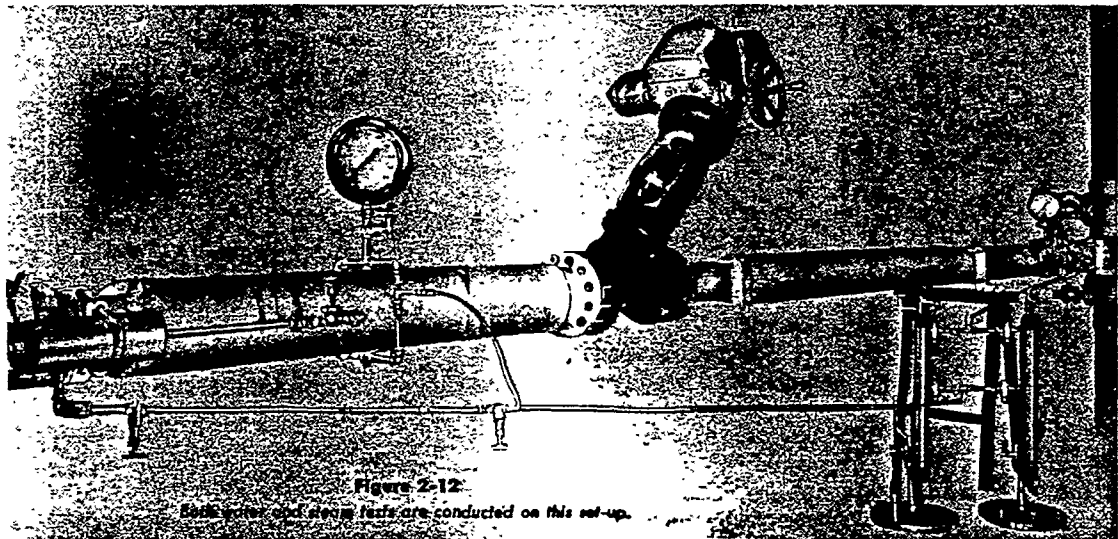


Figure 2-12
Tests are conducted on this set-up.

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

The numerous types of valves and fittings and the great variety of service conditions make it virtually impossible to obtain test data on every size and type of valve and fitting used today. For this reason, it is desirable to find a means for utilizing the limited test data which are available. Several methods of accomplishing this have been devised; the most commonly used are the "equivalent length", "resistance coefficient", and "flow coefficient".

Velocity in a pipe is obtained at the expense of static head, and decrease in static head due to velocity is:

$$h = \frac{v^2}{2g}, \quad \text{Equation 2-1}$$

which is defined as the "velocity head". Flow through a valve or fitting in a pipe line also causes a reduction in static head which may be expressed in terms of velocity head. The resistance coefficient K in the equation

$$h_L = K \frac{v^2}{2g}, \quad \text{Equation 2-2}$$

therefore, is defined as the number of velocity heads lost due to the valve or fitting. Also, the same head loss in straight pipe is expressed by the Darcy equation

$$h_L = \left(f \frac{L}{D}\right) \frac{v^2}{2g} \quad \text{Equation 2-3}$$

It follows that,

$$K = \left(f \frac{L}{D}\right) \quad \text{Equation 2-4}$$

The ratio L/D is the equivalent length in pipe diameters of straight pipe which will cause the same pressure drop as the valve under the same flow conditions.

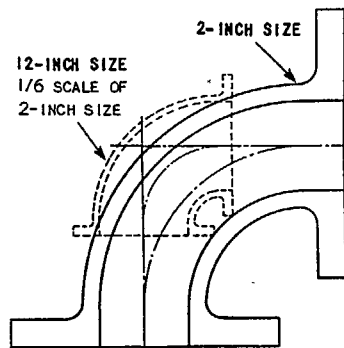


Figure 2-13
Geometrical dissimilarity between 2 and 12-inch standard cast iron flanged elbows

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. An example of geometric dis-

similarity is shown in Figure 2-13 where a 12-inch standard elbow has been drawn to 1/6 scale of 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.

Figure 2-14 on the opposite page is based on the analysis of extensive test data from various sources. The K coefficients for a number of lines of valves and fittings have been plotted against size. It will be noted that the slopes of the K curves show a definite tendency to follow the same slope as the $f(L/D)$ curve for straight pipe. It is probably coincidence that the effect of geometric dissimilarity between different sizes of the same line of valves or fittings upon the resistance coefficient K is similar to that of relative roughness, or size of pipe, upon friction factor.

Based on the evidence presented in Figure 2-14, it can be said that the resistance coefficient K , for a given line of valves or fittings, tends to vary with size as does the friction factor f for straight pipe, and that the equivalent length L/D tends toward a constant for the various sizes of a given line of valves or fittings.

In the flow range of complete turbulence as defined by the Friction Factor Charts, pages A-24 and A-25, the K coefficient for a given size and the L/D value are, of course, constant. In the transition zone, where f for pipe increases with decreasing Reynolds numbers, it is assumed that the value of L/D is constant and that K varies in the same manner as the friction factor. Limited tests have shown that this is not an exact relationship and that it may vary for different types of valves and fittings; however, since the tendency is in this direction, it is believed to provide more accurate solutions than would the assumption that K is constant for all Reynolds numbers.

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 . The C_v coefficient of a valve is defined as the flow of water at 60 F, in gallons per minute, at a pressure drop of one pound per square inch across the valve.

By the substitution of appropriate equivalent units in the Darcy equation, it can be shown that,

$$C_v = \frac{29.9d^2}{\sqrt{f \frac{L}{D}}} = \frac{29.9d^2}{\sqrt{K}} \quad \text{Equation 2-5}$$

(continued on next page)

Resistance Coefficient K , Equivalent Length L/D , And Flow Coefficient C_v — continued

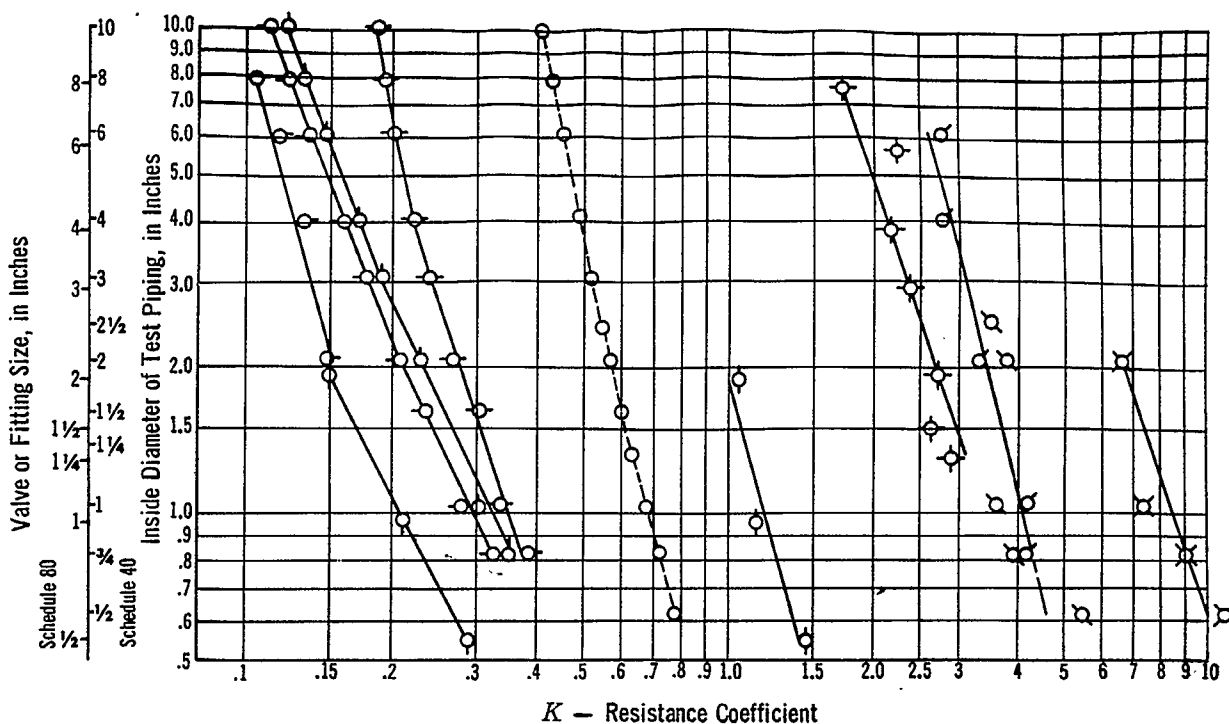


Figure 2-14, Variation of Resistance Coefficient $K (= f L/D)$ with Size

Symbol	Product Tested	Authority
○	Schedule 40 Pipe, 30 Diameters Long ($K = 30 f$)	Moody A.S.M.E. Trans., Nov.-1944 ¹
○	125-Pound Iron Body Wedge Gate Valves	Univ. of Wisc. Exp. Sta. Bull., Vol. 9, No. 1, 1922 ¹⁰
○	600-Pound Steel Wedge Gate Valves	Crane Tests
○	90 Degree Pipe Bends, $R/D = 2$	Pigott A.S.M.E. Trans., 1950 ⁸
○	90 Degree Pipe Bends, $R/D = 3$	Pigott A.S.M.E. Trans., 1950 ⁸
○	90 Degree Pipe Bends, $R/D = 1$	Pigott A.S.M.E. Trans., 1950 ⁸
○	600-Pound Steel Wedge Gate Valves, Seat Reduced	Crane Tests
○	300-Pound Steel Venturi Ball-Cage Gate Valves	Crane-Armour Tests
○	125-Pound Iron Body Y-Pattern Globe Valves	Crane-Armour Tests
○	125-Pound Brass Angle Valves, Composition Disc	Crane Tests
○	125-Pound Brass Globe Valves, Composition Disc	Crane Tests

(continued from the preceding page).

Also, the quantity in gallons per minute of any liquid having a viscosity close to that of water at 60 F that will flow through the valve can be determined from:

$$Q = C_v \sqrt{\Delta P \left(\frac{62.4}{\rho} \right)} \quad \text{Equation 2-6}$$

$$Q = 7.9 C_v \sqrt{\frac{\Delta P}{\rho}}$$

and the pressure drop can be computed from the

same formula arranged as follows:

$$\Delta P = \frac{\rho}{62.4} \left(\frac{Q}{C_v} \right)^2 \quad \text{Equation 2-6}$$

Since Equations 2-2, 2-3, and 2-6 are simply other forms of the Darcy equation, the limitations regarding their use for compressible flow (explained in Chapter 1, Page 1-7) apply. Other convenient forms of Equations 2-2, 2-3, and 2-6 in terms of commonly used units are presented on page 3-4.

Relationship of Equivalent Length L/D and Resistance Coefficient K To Inside Diameter of Connecting Pipe

Tests have shown that the pressure drop due to a given valve or fitting does not change when the product is installed with pipe of the same nominal size but of different thickness. Small variations in entrance and exit losses caused by mating the valve ends to variable pipe thicknesses, within reasonable limits, are insignificant. Since the pressure drop is a function of the square of the velocity, and velocity is a function of the square of the internal diameter, it follows that the equivalent length of a given valve or fitting, expressed in terms of the pipe to which it is connected, varies as the fourth power of the internal diameter of the pipe. For example, if the equivalent length of a 1/2-inch valve is determined by test to be 100 pipe diameters of Schedule 80 pipe, its equivalent length will be 169 diameters of Schedule 40 pipe, since the ratio of the inside diameters of the two pipes to the fourth power is 1.69. This ratio, of course, varies with the different sizes and thicknesses of pipes. The same relationship applies to the resistance coefficient K .

In view of this condition, the Crane Engineering Laboratories have established the practice of making all flow tests with pipe having internal diameters normally used with the particular valve or fitting. For this purpose, pipe normally used with the various pressure classes of valves and fittings has been arbitrarily established in accordance with the table shown at the top of page A-30.

Computation of pressure drop using equivalent length or K coefficient data established on the basis of this table should be made using pipe dimensions specified on pages B-16 to B-18; for installation conditions not in agreement with the table, the equivalent length in pipe diameters or the K resistance coefficient should be multiplied by the ratio of diameters to the fourth power.

$$K_a = K_b \left(\frac{d_a}{d_b} \right)^4 \quad \text{Equation 2-7}$$

$$\left(\frac{L}{D} \right)_a = \left(\frac{L}{D} \right)_b \left(\frac{d_a}{d_b} \right)^4 \quad \text{where,}$$

Subscript "a" defines the resistance coefficients with reference to the internal diameter of the pipe in which the valve will be installed.

Subscript "b" defines the known resistance coefficients and the internal diameters of the pipe for which these coefficients were established.

This procedure of obtaining equivalent length or K coefficient data with respect to pipe of various internal diameters corrects a significant variable that has often been neglected in translating test data and makes possible a more accurate prediction of the flow characteristics of untested valves and fittings by comparison of detail dimensions and shapes with tested items.

Valves with Gradually Increased Ports

Various types of valves are often made with reduced seats and have uniformly tapered ports. Straight-through valves such as gates and plug cocks, when so designed, are sometimes referred to as venturi valves. When the transition from seat size to port size is uniformly tapered with an angle, approximately 20 degrees or less with the center line, the flow characteristics may be determined in the same manner as for valves installed with pipe having an inside diam-

eter other than that with which the valve was tested.

Tests have shown the effect of tapered port sections upon flow characteristics is minor compared to losses across the valve seat. This applies particularly to valves having relatively high flow resistances.

Equation 2-7 may be used to establish approximate resistance coefficients for reduced seat valves.

Effect of End Connections

Analysis of many tests indicates there is little, if any, justification for assigning different resistance coefficients to a given valve or fitting with varying types of end connections. The difference between

flanged, screwed, and welding ends in this respect has been found to be insignificant. The pressure drop due to unions, couplings, and flanged joints is, likewise, insignificant.

Laminar Flow Conditions

One of the problems in flow of fluids confronting engineers from time to time, for which there is very meager information, is the resistance of valves and fittings under laminar flow conditions. Flow through straight pipe is adequately covered by the basic flow equation,

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

which is identical to Poiseuille's law for laminar flow when the equation for f in this flow range, $f = 64/R_e$, is included in the formula.

For solution of these problems, we have developed on the basis of data presented in "Principles of Chemical Engineering" by Walker, Lewis, McAdams and Gilliland²², the empirical relationship between equivalent length in the laminar flow region to that in the turbulent region, namely:

$$\left(\frac{L}{D}\right)_s = \frac{R_e}{1000} \left(\frac{L}{D}\right)_t$$

Equation 2-8

Subscript "s" refers to the equivalent length in pipe diameters under laminar flow conditions where the Reynolds number is less than 1000.

Subscript "t" refers to the equivalent length in pipe diameters determined from tests in the turbulent flow range. Representative values of equivalent length are given in the table on page A-30.

The minimum equivalent length is the length in pipe diameters of the centerline of the actual flow path through the valve or fitting. While laboratory test data supporting this method is meager, reports of field experience indicate that the results obtained agree closely with observed conditions.

Basis for Design of Charts for Determining Equivalent Length, Resistance Coefficient, and Flow Coefficient

The table on page A-30 lists average equivalent length data expressed in pipe diameters abstracted from all available tests. It is not practical to identify all valve and fitting types with the many variations in design which may affect the flow characteristics. The data given for globe and angle valves represent actual tests on the variation of designs indicated. By using the data given in this table along with the principles presented from page 2-8 to this point as a basis, reasonable equivalent length values can be estimated for any valve or fitting upon consideration of design features, such as, the relative area of seat or restricted sections to pipe diameter and the shape of the flow passage.

The chart on page A-31 provides a convenient means of translating equivalent length in pipe diameters as given in the table on page A-30, to equivalent length in feet of pipe for any given size of valve or fitting. Also, the resistance coefficient K can be readily determined from this chart for any size, if the resistance coefficient or equivalent length for any other size of the same item has been established, either by experiment or estimate.

The chart on page A-32 gives a graphical solution of Equation 2-5 and permits a ready determination of C_v if K is known. An example illustrating the use of this chart is given on page 4-2.

Limitations of charts: As explained on page 2-8 and at the top of this page, the value of L/D for a given type of valve or fitting is considered to be constant for flow conditions resulting in Reynolds

numbers of 1000 or greater. Equivalent lengths either in pipe diameters or feet of pipe, as determined from pages A-30 or A-31, are therefore legitimate for all flow conditions except in the laminar flow range where the Reynolds number is less than 1000. For Reynolds numbers less than 1000, values of L/D must be determined in accordance with Equation 2-8.

On the other hand, values of K , as determined from pages A-31 and A-32, are legitimate only for flow conditions resulting in Reynolds numbers falling in the completely turbulent flow range, as defined by the friction factor on pages A-24 and A-25. At lower flow rates, values of K vary in approximately the same manner as does the value of friction factor with Reynolds number. At the lower flow rates, the value of K as determined from pages A-31 and A-32 should be multiplied by the ratio:

$$\frac{f \text{ (at calculated Reynolds number)}}{f \text{ (in range of complete turbulence, where } f \text{ is constant)}}$$

When K has been corrected for flow in the transition or laminar flow range, C_v can be obtained directly from page A-32 by employing the corrected K factor. However, if the C_v factor is furnished for the completely turbulent flow range, as defined by the friction factor charts on pages A-24 and A-25, it must be corrected by multiplying it by the ratio,

$$\sqrt{\frac{f \text{ (in range of complete turbulence, where } f \text{ is constant)}}{f \text{ (at calculated Reynolds number)}}}$$

since C_v varies inversely with the square root of the friction factor.

Resistance of Bends

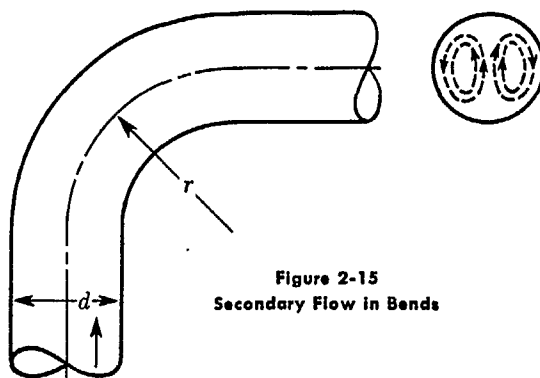


Figure 2-15
Secondary Flow in Bends

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 frictional resistance of the pipe walls and the action of centrifugal force combine to produce this rotation. Figure 2-15 illustrates this phenomenon.

Resistance of bend, 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:

$$h_t = h_p + h_c + h_L \quad \text{Equation 2-9}$$

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:

$$h_b = h_p + h_c \quad \text{Equation 2-10}$$

then:

$$h_t = h_b + h_L$$

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

$$h_b = K_b \frac{v^2}{2g} \quad \text{Equation 2-11}$$

where:

K_b = the bend coefficient

v = velocity through pipe, feet per second

g = 32.2 feet per second per second

The relationship between K_b and r/d (relative radius*) is not well defined, as can be observed by reference to Figure 2-16 (taken from the work of Beij²¹). The curves in this chart indicate that K_b has a minimum value when r/d is between 3 and 5.

The chart on page A-27 shows the resistance of 90 degree bends in terms of equivalent length of straight pipe. These curves . . . also based on the work of Beij . . . are believed to represent average conditions for the flow of fluids in 90 degree bends.

Tests have shown that the loss due to continuous bends greater than 90 degrees, such as in pipe coils, is less than the summation of the losses in the 90 degree bends contained in the coil, considered separately. This is reasonable, since the loss h_p in Equation 2-9 occurs only once in such a bend. Reasonably accurate results for pipe coils and expansion loops consisting of continuous bends can be obtained by the use of the chart on page A-27 . . . if the number of 90 degree bends contained in the coil minus one, multiplied by the resistance due to length plus one-half of the bend resistance, is added to the total resistance of a 90 degree bend.

For example, a pipe coil consisting of four complete turns . . . sixteen 90 degree bends . . . and having a relative radius of five pipe diameters, would have a total equivalent length, in pipe diameters, of:

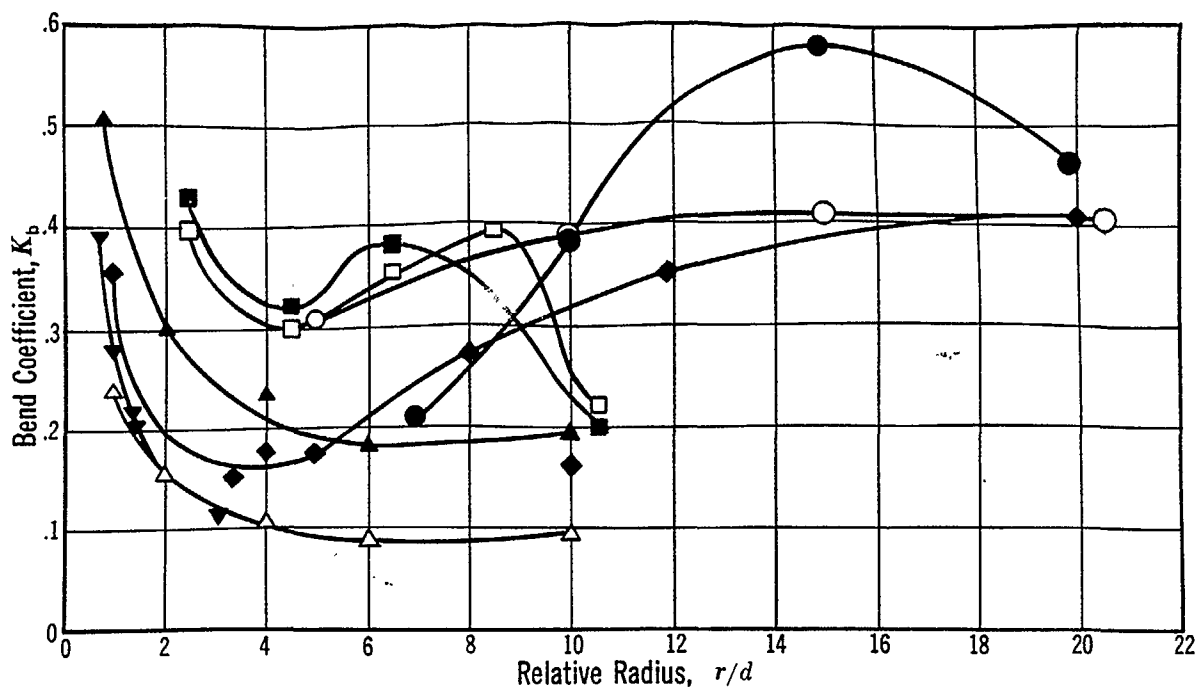
$$15(8 + 4) + 16 = 196$$

It will be noted that this assumes $h_p = h_c$ in Equation 2-9; this relationship has not been established by tests but is believed to represent the most accurate estimate that can be made until further experimental data are available.

Resistance of miter bends: The equivalent length of miter bends, based on the work of H. Kirchbach⁴, is also shown on page A-27.

*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.

Resistance of Bends — continued

Figure 2-16, Bend Coefficients Found by Various Investigators (Beij²¹)

From "Pressure Losses for Fluid Flow in 90° Pipe Bends" by K. H. Beij.
 Courtesy of Journal of Research of National Bureau of Standards.

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

Other Resistances to Flow

In addition to the resistance due to valves and fittings already discussed, losses due to sudden enlargement and sudden contraction are encountered whenever fittings such as reducing or increasing flanges, bushings, etc., are used. Also, when a fluid enters or leaves an open end pipe, entrance and exit losses occur. As in the case of valves and fittings, these losses can be expressed by the formula:

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

Unlike most other fittings, there is no length involved in losses due to these conditions; thus, relative roughness is not a factor in these resistances, and geometric

similarity does exist. The resistance due to sudden enlargement and sudden contraction, as well as entrance and exit losses expressed in terms of velocity head or K factor, are therefore independent of pipe size. Resistance coefficient K for such conditions are given on page A-26.

Equivalent lengths corresponding to these resistance coefficients for any size can be readily determined from the chart on page A-31. For example, the equivalent lengths of sharp edged entrances ($K = 0.5$) to 2 and 6-inch pipes can be read from the nomograph on page A-31 as 26 diameters of 2-inch pipe and 33 diameters of 6-inch pipe, respectively.

Flow Through Nozzles and Orifices

The discharge of fluids through nozzles and orifices has been subject to continued investigation and, as a result, well-established data are still being supplemented. A portion of the subject is covered on these facing pages but more complete references will be found in the Bibliography^{8, 9, 10}, or from the data supplied by meter manufacturers.

The rate of flow of any fluid through an orifice or nozzle, neglecting the velocity of approach, may be expressed by:

$$q = C_d A \sqrt{2g h_L} \quad \text{Equation 2-12}$$

Velocity of approach may have considerable effect on the quantity discharged through a nozzle or orifice. The factor correcting for velocity of approach,

$$\frac{1}{\sqrt{1 - \left(\frac{d_0}{d_1}\right)^4}}$$

may be incorporated in Equation 2-12 as follows:

$$q = \frac{C_d A}{\sqrt{1 - \left(\frac{d_0}{d_1}\right)^4}} \sqrt{2g h_L} \quad \text{Equation 2-13}$$

The quantity

$$\frac{C_d}{\sqrt{1 - \left(\frac{d_0}{d_1}\right)^4}}$$

is defined as the flow coefficient C . Values of C for nozzles and orifices are shown on page A-19. Use of the flow coefficient C eliminates the necessity for calculating the velocity of approach, and Equation 2-13 may now be written:

$$q = C A \sqrt{2g h_L} = C A \sqrt{\frac{2g(144)\Delta P}{\rho}} \quad \text{Equation 2-14}$$

Orifices and nozzles are normally used in piping systems as metering devices and are installed with flange taps or pipe taps in accordance with ASME specifications. The values of h_L and ΔP in Equation 2-14 are the measured differential static head or pressure across flange taps when values of C are taken from page A-19. The flow coefficient C is plotted for Reynolds numbers based on the internal diameter of the upstream pipe.

Flow of liquids: For nozzles and orifices discharging incompressible fluids to atmosphere, C values may be taken from page A-19 if h_L or ΔP in Equation 2-14 is taken as the upstream head or gauge pressure. For most conditions of flow of fluids hav-

ing a low viscosity, i.e., water, gasoline, etc., the Reynolds number need not be calculated since it will fall in the range of the values on page A-19, where the flow coefficient C is a constant.

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

$$q = YCA \sqrt{\frac{2g(144)\Delta P}{\rho}} \quad \text{Equation 2-15}$$

The expansion factor Y is a function of:

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

This factor^{9, 10} has been experimentally determined on the basis of air, which has a specific heat ratio of 1.4, and steam having specific heat ratios of approximately 1.3. The data is plotted on page A-20 and values of other specific heat ratios have been included to extend the use of the data. Values of k for some of the common vapors and gases are given on pages A-8 and A-9. 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 2-15 may be used for orifices discharging compressible fluids to atmosphere by using:

1. Flow coefficient C given on page A-19 in the Reynolds number range where C is a constant for the given diameter ratio.
2. Expansion factor Y per page A-20.
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 on the next page. When the absolute inlet pressure is greater than this amount, flow through nozzles should be calculated as outlined on the following page.

Flow Through Nozzles and Orifices — continued

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-21.

Flow through nozzles and venturi meters is limited by critical pressure ratio, and minimum values of Y to be used in Equation 2-15 for this condition, are indicated on page A-20 by the termination of the curves at $P'_2/P'_1 = r_c$.

Equation 2-15 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-20
- C page A-19
- ΔP $P'_1(1 - r_c)$; r_c per page A-21
- 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)¹ are not available, it is suggested that reasonably accurate approximations may be obtained by using Equations 2-14 and 2-15, 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 entrance would have characteristics similar to those for square edged orifices.

Discharge of Fluids Through Valves, Fittings, and Pipe

Liquid flow: To determine the flow of liquid through pipe, the Darcy formula is used. Equation 1-4 (page 1-6) has been converted to more convenient terms in Chapter 3 and has been rewritten as Equation 3-14. The form of Equation 3-14 which is most applicable to liquid flow is written in terms of flow rate in gallons per minute.

$$h_L = \frac{0.00259 KQ^2}{d^5}$$

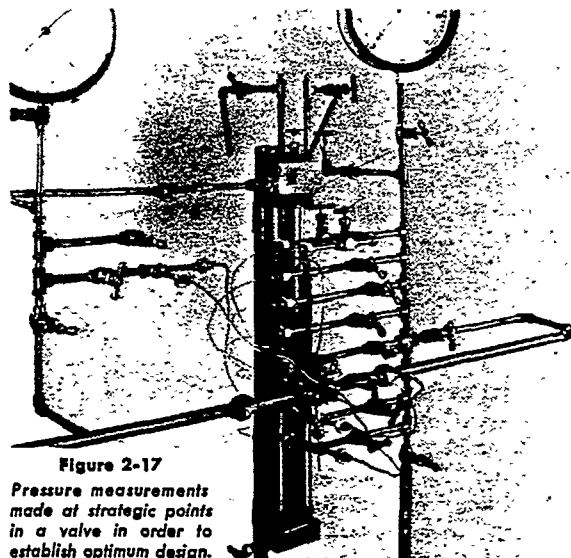


Figure 2-17
Pressure measurements
made at strategic points
in a valve in order to
establish optimum design.

Loss of head in terms of resistance coefficient K has been selected since entrance and exit losses are usually given in terms of velocity head loss, K (see page A-26). Solving for Q , the equation can be rewritten,

$$Q = \sqrt{\frac{h_L d^5}{0.00259 K}} = 19.65 d^2 \sqrt{\frac{h_L}{K}}$$

$$Q = 19.65 d^2 \sqrt{\frac{h_L}{f \frac{L}{D}}} \quad \text{Equation 2-16}$$

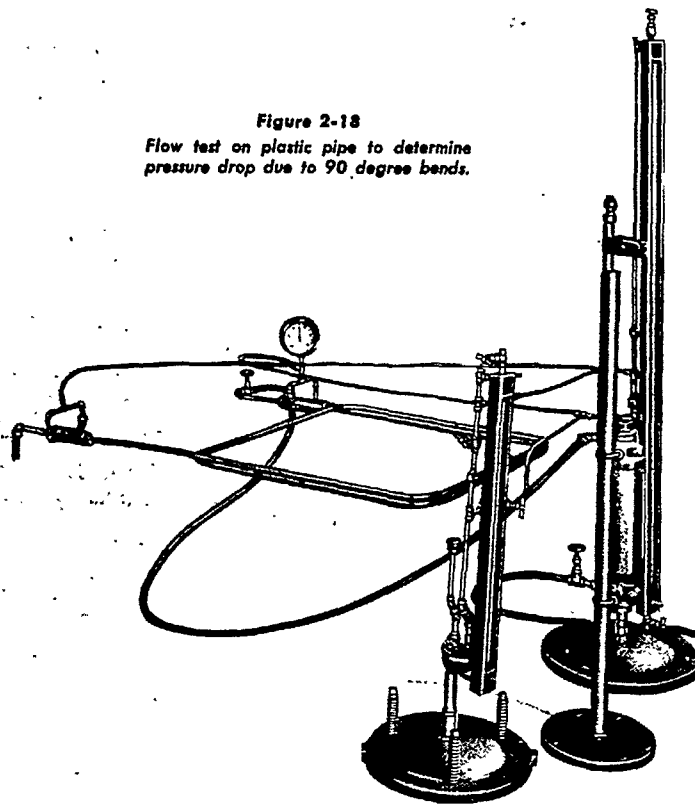
Equation 2-16 can be employed for valves, fittings, and pipe where K would be the sum of all the resistances in the piping system, including entrance and exit losses when they exist. Examples of problems of this type are shown on page 4-12.

Compressible flow: When a compressible fluid flows from a piping system into an area of larger cross section than that of the pipe, as in the case of discharge to atmosphere, a modified form of the Darcy formula, Equation 1-11 developed on page 1-9, is used.

$$w = 0.525 Y d^2 \sqrt{\frac{\Delta P}{K V_1}}$$

The determination of values of K , Y , and ΔP in this equation is described on page 1-9 and is illustrated in the examples on pages 4-13 and 4-14.

Figure 2-18
*Flow test on plastic pipe to determine
pressure drop due to 90 degree bends.*



Formulas and Nomographs For Flow Through Valves, Fittings, and Pipe

CHAPTER 3

Only basic formulas needed for the presentation of the theory of fluid flow through valves, fittings, and pipe were presented in the first two 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 in this country. 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.

Nomographs presented in this chapter are graphical solutions of the flow formulas applying to pipe. Valve and fitting flow problems may also be solved by means of these nomographs by determining their equivalent length in terms of feet of straight pipe.

Due to the wide variety of terms and the variation in the physical properties of liquids and gases, it was necessary to divide the nomographs into two parts: the first part (pages 3-6 to 3-15) pertains to liquid flow, and the second part (pages 3-16 to 3-25), pertains to compressible flow.

All nomographs for the solution of pressure drop problems are based upon Darcy's formula, since it is a general formula which is applicable to all fluids and can be applied to all types of pipe through the use of the Moody Friction Factor Diagram. Darcy's formula also provides a means of solving problems of flow through valves and fittings on the basis of equivalent length or resistance coefficient. Nomographs provide simple, rapid, practical, and reasonably accurate solutions to flow formulas and the decimal point is accurately located.

Accuracy of a nomograph is limited by the available page space, length of scales, number of units provided on each scale, and the angle at which the connecting line crosses the scale. Whenever the solution of a problem falls beyond the range of a nomograph, the slide rule or arithmetical solution of the formula must be employed.

Summary of Formulas

To eliminate needless duplication, formulas have been written in terms of either specific volume $\bar{V}$ or weight density ρ , but not in terms of both, since one is the reciprocal of the other.

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

These equations may be substituted in any of the formulas shown in this paper whenever necessary.

● Bernoulli's theorem:

Equation 3-1

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

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

● Mean velocity of flow in pipe: (Continuity Equation)

Equation 3-2

$$v = \frac{q}{A} = 183.3 \frac{q}{d^2} = 0.408 \frac{Q}{d^2}$$

$$v = 0.286 \frac{B}{d^2} = 183.3 \frac{w\bar{V}}{d^2} = 0.0509 \frac{W\bar{V}}{d^2}$$

$$v = 0.00144 \frac{q'_h T}{P' d^2} = 0.00389 \frac{q'_h S_g}{\rho d^2}$$

$$V = \frac{q_m}{A} = 2.40 \frac{W\bar{V}}{a} = 3.06 \frac{W\bar{V}}{d^2}$$

$$V = 0.0865 \frac{q'_h T}{P' d^2} = 0.233 \frac{q'_h S_g}{\rho d^2}$$

● Reynolds number of flow in pipe:

Equation 3-3

$$R_e = \frac{Dv\rho}{\mu_e} = \frac{Dv\rho}{32.2\mu'_e} = 123.9 \frac{dv\rho}{\mu}$$

$$R_e = 22700 \frac{qp}{d\mu} = 50.6 \frac{Qp}{d\mu} = 35.4 \frac{Bp}{d\mu}$$

$$R_e = 6.31 \frac{W}{d\mu} = 0.482 \frac{q'_h S_g}{d\mu}$$

$$R_e = \frac{Dv}{\nu'} = \frac{dv}{12\nu'} = 7740 \frac{dv}{\nu}$$

$$R_e = 1419000 \frac{q}{\nu d} = 3160 \frac{Q}{\nu d} = 394 \frac{W\bar{V}}{\nu d}$$

● Viscosity equivalents:

Equation 3-4

$$\nu = \frac{\mu}{\rho'} = \frac{\mu}{S}$$

● Head loss and pressure drop in straight pipe:

Pressure loss due to flow is the same in a sloping, vertical, or horizontal pipe. However, the difference in pressure due to the difference in head must be considered in pressure drop calculations; see page 1-5.

Darcy's formula:

Equation 3-5

$$h_L = f \frac{L}{D} \frac{v^2}{2g} = 0.1863 \frac{fLv^2}{d}$$

$$h_L = 6260 \frac{fLq^2}{d^5} = 0.0311 \frac{fLQ^2}{d^5}$$

$$h_L = 0.01524 \frac{fLB^2}{d^5} = 0.000483 \frac{fLW^2\bar{V}}{d^5}$$

$$\Delta P = 0.001294 \frac{fL\rho v^2}{d} = 0.00000359 \frac{fL\rho V^2}{d}$$

$$\Delta P = 43.5 \frac{fL\rho q^2}{d^5} = 0.000216 \frac{fL\rho Q^2}{d^5}$$

$$\Delta P = 0.0001058 \frac{fL\rho B^2}{d^5} = 0.00000336 \frac{fLW^2\bar{V}}{d^5}$$

$$\Delta P = 0.0000000726 \frac{fLT(q'_h)^2 S_g}{d^5 P'}$$

$$\Delta P = 0.0000001959 \frac{fL(q'_h)^2 S_g^2}{d^5 \rho}$$

For simplified compressible fluid formula, see page 3-22.

● Head loss and pressure drop with laminar flow in straight pipe:

For laminar flow conditions ($R_e < 2000$), the friction factor is a direct mathematical function of the Reynolds number only, and can be expressed by the formula: $f = 64/R_e$. Substituting this value of f in the Darcy formula, it can be rewritten:

$$h_L = 0.0962 \frac{\mu L v}{d^2 \rho} \quad \text{Equation 3-6}$$

$$h_L = 17.65 \frac{\mu L q}{d^4 \rho} = 0.0393 \frac{\mu L Q}{d^4 \rho}$$

$$h_L = 0.0275 \frac{\mu L B}{d^4 \rho} = 0.00490 \frac{\mu L W}{d^4 \rho^2}$$

$$\Delta P = 0.000668 \frac{\mu L v}{d^2} = 0.1225 \frac{\mu L q}{d^4}$$

$$\Delta P = 0.000273 \frac{\mu L Q}{d^4} = 0.000191 \frac{\mu L B}{d^4}$$

$$\Delta P = 0.0000340 \frac{\mu L W}{d^4 \rho}$$

Summary of Formulas — continued

● Limitations of Darcy formula

Non-compressible flow; liquids:

The Darcy formula 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.

Compressible flow; gases and vapors:

When pressure drop is less than 10% of P_1 , use ρ or $\bar{V}$ based on either inlet or outlet conditions.

When pressure drop is greater than 10% of P_1 but less than 40% of P_1 , use the average of ρ or $\bar{V}$ based on inlet and outlet conditions, or use Equation 3-20.

When pressure drop is greater than 40% of P_1 , use the rational or empirical formulas given on this page for compressible flow, or use Equation 3-20 (for theory, see page 1-9).

● Isothermal flow of gas in pipe lines

Equation 3-7

$$w = \sqrt{\frac{144g A^2}{\bar{V}_1 \left(f \frac{L}{D} + 2 \log_e \frac{P'_1}{P'_2} \right) \left(\frac{(P'_1)^2 - (P'_2)^2}{P'_1} \right)}}$$

$$w = 0.371 \sqrt{\frac{d^4}{\bar{V}_1 \left(f \frac{L}{D} + 2 \log_e \frac{P'_1}{P'_2} \right) \left(\frac{(P'_1)^2 - (P'_2)^2}{P'_1} \right)}}$$

● Simplified compressible flow for long pipe lines

Equation 3-7a

$$w = \sqrt{\left(\frac{144g A^2}{\bar{V}_1 f \frac{L}{D}} \right) \left(\frac{(P'_1)^2 - (P'_2)^2}{P'_1} \right)}$$

$$w = 0.1072 \sqrt{\left(\frac{d^5}{\bar{V}_1 f L} \right) \left(\frac{(P'_1)^2 - (P'_2)^2}{P'_1} \right)}$$

$$q'_h = 114.2 \sqrt{\left(\frac{(P'_1)^2 - (P'_2)^2}{f L_m T S_g} \right) d^5}$$

● Maximum (sonic) velocity of compressible fluids in pipe

The maximum possible velocity of a compressible fluid in a pipe is equivalent to the speed of sound in the fluid; this is expressed as:

$$v_s = \sqrt{k g R T} \quad \text{Equation 3-8}$$

$$v_s = \sqrt{k g 144 P' \bar{V}}$$

$$v_s = 68.1 \sqrt{k P' \bar{V}}$$

● Empirical formulas for the flow of water, steam, and gas

Although the rational method (using Darcy's formula) for solving flow problems has been recommended in this paper, some engineers prefer to use empirical formulas.

Hazen and Williams formula for flow of water:

Equation 3-9

$$Q = 0.442 d^{2.63} c \left(\frac{P_1 - P_2}{L} \right)^{0.54}$$

where:

 $c = 140$ for new steel pipe $c = 130$ for new cast iron pipe $c = 110$ for riveted pipe

Babcock formula for steam flow:

Equation 3-10

$$\Delta P = 0.000000363 \left(\frac{d + 3.6}{d^6} \right) W^2 L \bar{V}$$

$$\Delta P = 0.470 \left(\frac{d + 3.6}{d^6} \right) w^2 L \bar{V}$$

Spitzglass formula for low pressure gas: (pressure less than one pound gauge)

Equation 3-11

$$q'_h = 3550 \sqrt{\frac{\Delta h_w d^5}{S_g L \left(1 + \frac{3.6}{d} + 0.03 d \right)}}$$

Flowing temperature is 60 F.

Weymouth formula for high pressure gas:

Equation 3-12

$$q'_h = 28.0 d^{2.667} \sqrt{\left(\frac{(P'_1)^2 - (P'_2)^2}{S_g L_m} \right) \left(\frac{520}{T} \right)}$$

Panhandle formula³ for natural gas pipe lines 6 to 24-inch diameter and $R_s = (5 \times 10^4)$ to (14×10^4) :

Equation 3-13

$$q'_h = 36.8E d^{2.6182} \left(\frac{(P'_1)^2 - (P'_2)^2}{L_m} \right)^{0.5394}$$

where: gas temperature = 60 F

 $S_g = 0.6$ E = flow efficiency $E = 1.00$ (100%) 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

Summary of Formulas — continued

● 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:

$$h_L = f \frac{L}{D} \frac{v^2}{2g} \quad \text{Equation 3-5}$$

and head loss through a valve is:

$$h_L = K \frac{v^2}{2g} \quad \text{Equation 3-14}$$

$$\text{therefore: } K = f \frac{L}{D} \quad \text{Equation 3-15}$$

To eliminate needless duplication of formulas, the following are all given in terms of K . Whenever necessary, substitute $(f L/D)$ for (K) .

$$h_L = \frac{522 K q^2}{d^4} = 0.00259 \frac{K Q^2}{d^4} \quad \text{Equation 3-14}$$

$$h_L = 0.001270 \frac{K B^2}{d^4} = 0.0000403 \frac{K W^2 \bar{V}^2}{d^4}$$

$$\Delta P = 0.0001078 K \rho v^2 = 0.000000300 K \rho V^2$$

$$\Delta P = 3.62 \frac{K \rho q^2}{d^4} = 0.00001799 \frac{K \rho Q^2}{d^4}$$

$$\Delta P = 0.00000882 \frac{K \rho B^2}{d^4}$$

$$\Delta P = 0.000000280 \frac{K W^2 \bar{V}}{d^4}$$

$$\Delta P = 0.00000000605 \frac{K (q'_h)^2 T S_g}{d^4 P}$$

$$\Delta P = 0.00000001633 \frac{K (q'_h)^2 S_g^2}{d^4 \rho}$$

● Pressure drop and flow of liquids, with viscosity similar to water at 60 F, using flow coefficient

$$\Delta P = \left(\frac{Q}{C_v} \right)^2 \frac{\rho}{62.4} \quad \text{Equation 3-16}$$

$$Q = C_v \sqrt{\Delta P \frac{62.4}{\rho}} = 7.90 C_v \sqrt{\frac{\Delta P}{\rho}}$$

$$C_v = Q \sqrt{\frac{\rho}{\Delta P (62.4)}} = \frac{29.9 d^2}{\sqrt{f L/D}} = \frac{29.9 d^2}{\sqrt{K}}$$

$$K = \frac{891 d^4}{(C_v)^2}$$

$$\frac{L}{D} = \frac{K}{f} = \frac{891 d^4}{f (C_v)^2} \quad L = \frac{74.3 d^5}{f (C_v)^2}$$

● Head loss and pressure drop with laminar flow through valves; Darcy's formula

$$h_L = 0.00328 \left(\frac{L}{D} \right) \frac{\mu Q}{d^3 \rho} \quad \text{Equation 3-17}$$

$$h_L = 1.470 \left(\frac{L}{D} \right) \frac{\mu q}{d^3 \rho} = 0.00802 \left(\frac{L}{D} \right) \frac{\mu v}{d \rho}$$

$$h_L = 0.000408 \left(\frac{L}{D} \right) \frac{\mu W \bar{V}^2}{d^3}$$

$$\Delta P = 0.0000557 \left(\frac{L}{D} \right) \frac{\mu v}{d} = 0.01021 \left(\frac{L}{D} \right) \frac{\mu q}{d^3}$$

$$\Delta P = 0.0000228 \left(\frac{L}{D} \right) \frac{\mu Q}{d^3}$$

$$\Delta P = 0.00001593 \left(\frac{L}{D} \right) \frac{\mu B}{d^3}$$

$$\Delta P = 0.00000284 \left(\frac{L}{D} \right) \frac{\mu W \bar{V}}{d^3}$$

● Equivalent length correction for laminar flow with $R_e < 1000$

$$\left(\frac{L}{D} \right)_s = \left(\frac{L}{D} \right)_t \frac{R_e}{1000} \quad \text{Equation 3-18}$$

See pages 2-11 and A-30. Minimum $(L/D)_s$ = length of center line of actual flow path through valve or fitting. Subscript s refers to equivalent length with $R_e < 1000$. Subscript t refers to equivalent length with $R_e > 1000$.

● Discharge of fluid through valves, fittings, and pipe; Darcy's formula

$$\text{Liquid flow: } \quad \text{Equation 3-19}$$

$$q = 0.0438 d^2 \sqrt{\frac{h_L}{K}} = 0.525 d^2 \sqrt{\frac{\Delta P}{K \rho}}$$

$$Q = 19.65 d^2 \sqrt{\frac{h_L}{K}} = 236 d^2 \sqrt{\frac{\Delta P}{K \rho}}$$

$$w = 0.0438 d^2 \sqrt{\frac{h_L \rho^2}{K}} = 0.525 d^2 \sqrt{\frac{\Delta P \rho}{K}}$$

$$W = 157.6 d^2 \sqrt{\frac{h_L \rho^2}{K}} = 1891 d^2 \sqrt{\frac{\Delta P \rho}{K}}$$

Compressible flow:

$$q'_h = 40700 Y d^2 \sqrt{\frac{\Delta P P_1}{K T_1 S_g}} \quad \text{Equation 3-20}$$

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

$$q'_m = 678 Y d^2 \sqrt{\frac{\Delta P P_1}{K T_1 S_g}} = 412 \frac{Y d^2}{S_g} \sqrt{\frac{\Delta P \rho_1}{K}}$$

$$q' = 11.30 Y d^2 \sqrt{\frac{\Delta P P_1}{K T_1 S_g}} = 6.87 \frac{Y d^2}{S_g} \sqrt{\frac{\Delta P \rho_1}{K}}$$

$$w = 0.525 Y d^2 \sqrt{\frac{\Delta P}{K V_1}} \quad W = 1891 Y d^2 \sqrt{\frac{\Delta P}{K V_1}}$$

Values of Y are shown on page A-22. For K , Y , and ΔP determination, see examples on pages 4-13 and 4-14.

Summary of Formulas — concluded

● **Flow through nozzles and orifices**
(h_L and ΔP measured across flange taps)

Liquid: Equation 3-21

$$q = AC \sqrt{2g h_L}$$

$$q = 0.0438 d^2_0 C \sqrt{h_L} = 0.525 d^2_0 C \sqrt{\frac{\Delta P}{\rho}}$$

$$Q = 19.65 d^2_0 C \sqrt{h_L} = 236 d^2_0 C \sqrt{\frac{\Delta P}{\rho}}$$

$$w = 0.0438 d^2_0 C \sqrt{h_L \rho^2} = 0.525 d^2_0 C \sqrt{\Delta P \rho}$$

$$W = 157.6 d^2_0 C \sqrt{h_L \rho^2} = 1891 d^2_0 C \sqrt{\Delta P \rho}$$

Values of C are shown on page A-19.

Compressible fluids: Equation 3-22

$$q'_h = 40\,700 Y d^2_0 C \sqrt{\frac{\Delta P P_1}{T_1 S_g}}$$

$$q'_h = 24\,700 \frac{Y d^2_0 C}{S_g} \sqrt{\Delta P P_1}$$

$$q'_m = 678 Y d^2_0 C \sqrt{\frac{\Delta P P_1}{T_1 S_g}}$$

$$q'_m = 412 \frac{Y d^2_0 C}{S_g} \sqrt{\Delta P P_1}$$

$$q' = 11.30 Y d^2_0 C \sqrt{\frac{\Delta P P_1}{T_1 S_g}}$$

$$q' = 6.87 \frac{Y d^2_0 C}{S_g} \sqrt{\Delta P P_1}$$

$$w = 0.525 Y d^2_0 C \sqrt{\frac{\Delta P}{V_1}}$$

$$W = 1891 Y d^2_0 C \sqrt{\frac{\Delta P}{V_1}}$$

Values of C are shown on page A-19.
Values of Y are shown on page A-20.

● **Equivalents of head loss and pressure drop**

Equation 3-23

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

● **Changes in resistance coefficient K and equivalent length L/D required to compensate for different pipe I. D.**

$$K_a = K_b \left(\frac{d_a}{d_b} \right)^4 \quad \text{Equation 3-24}$$

$$\left(\frac{L}{D} \right)_a = \left(\frac{L}{D} \right)_b \left(\frac{d_a}{d_b} \right)^4 \quad (\text{see page A-30})$$

Subscript a refers to pipe in which valve will be installed.
Subscript b refers to pipe for which the resistance coefficient K or the equivalent length L/D was established.

● **Specific gravity of liquids**

Any liquid: Equation 3-25

$$s = \frac{\rho}{\rho_{\text{water at 60 F}}} \quad \left(\begin{array}{l} \text{any liquid at 60 F,} \\ \text{unless otherwise specified} \end{array} \right)$$

Oils: Equation 3-26

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

Liquids lighter than water: Equation 3-27

$$S (60 \text{ F}/60 \text{ F}) = \frac{140}{130 + \text{Deg Baumé}}$$

Liquids heavier than water: Equation 3-28

$$S (60 \text{ F}/60 \text{ F}) = \frac{145}{145 - \text{Deg Baumé}}$$

● **Specific gravity of gases**

Equation 3-29

$$S_g = \frac{R (\text{air})}{R (\text{gas})} = \frac{53.3}{R (\text{gas})}$$

$$S_g = \frac{M (\text{gas})}{M (\text{air})} = \frac{M (\text{gas})}{29}$$

● **General gas laws for perfect gases**

$$p'V_a = w_a RT \quad \text{Equation 3-30}$$

$$\rho = \frac{w_a}{V_a} = \frac{p'}{RT} = \frac{144 p'}{RT} \quad \text{Equation 3-31}$$

$$R = \frac{1544}{M} = \frac{144 p'}{\rho T} \quad \text{Equation 3-32}$$

Equation 3-33

$$p'V_a = n_a MRT = n_a 1544 T = \frac{w_a}{M} 1544 T$$

Equation 3-34

$$\rho = \frac{w_a}{V_a} = \frac{p' M}{1544 T} = \frac{P' M}{10.72 T} = \frac{2.70 P' S_g}{T}$$

where:

$$n_a = w_a/M = \text{number of mols of a gas}$$

● **Hydraulic radius***

Equation 3-35

$$\text{Hydraulic radius} = \frac{\text{cross sectional area of flowing fluid}}{\text{wetted perimeter}}$$

Equivalent diameter:

$$D = 4 \text{ times hydraulic radius (feet)}$$

$$d = 4 \text{ times hydraulic radius (inches)}$$

*See page 1-4 for limitations.

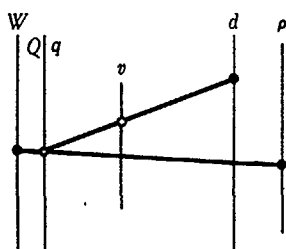
Velocity of Liquids in Pipe

The mean velocity of any flowing liquid can be calculated from the following formula, or, from the nomograph on the opposite page. The nomograph is a graphical solution of the formula.

$$v = 183.3 \frac{q}{d^2} = 0.408 \frac{Q}{d^2} = 0.0509 \frac{W}{d^2 \rho}$$

(For values of d^2 , see pages B-16 to B-18)

The pressure drop per 100 feet and the velocity in Schedule 40 pipe, for water at 60 F, have been calculated for commonly used flow rates for pipe sizes of $\frac{1}{8}$ to 24-inch; these values are tabulated on page B-14.



Example 1

Given: No. 3 Fuel Oil at 60 F flows through a 2-inch Schedule 40 pipe at the rate of 45,000 pounds per hour.

Find: The rate of flow in gallons per minute and the mean velocity in the pipe.

Solution:

1. $\rho = 56.02$ page A-7

	Connect	Read
2.	$W = 45\ 000$ $\rho = 56.02$	$Q = 100$
3.	$Q = 100$ 2" Sched 40	$v = 10$

Example 2

Given: Maximum flow rate of a liquid will be 300 gallons per minute with maximum velocity limited to 12 feet per second through Schedule 40 pipe.

Find: The smallest suitable pipe size and the velocity through the pipe.

Solution:

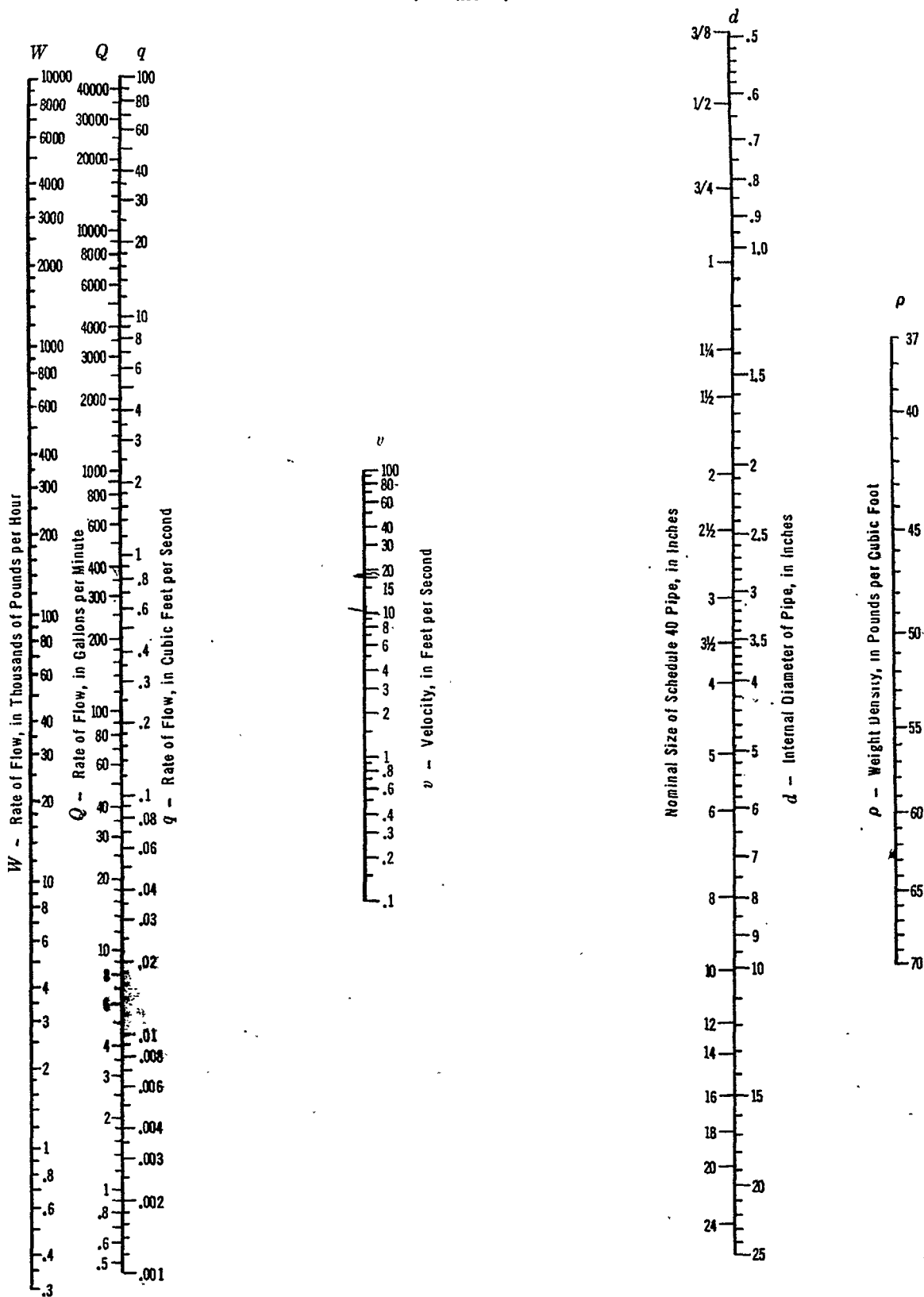
	Connect	Read
1.	$Q = 300$ $v = 12$	$d = 3.2$
2.	3½" Schedule 40 pipe suitable	
3.	$Q = 300$ 3½" Sched 40	$v = 10$

Reasonable Velocities For the Flow of Water through Pipe

Service Condition	Reasonable Velocity
Boiler Feed.....	8 to 15 feet per second
Pump Suction and Drain Lines.....	4 to 7 feet per second
General Service.....	4 to 10 feet per second
City.....	to 7 feet per second

Velocity of Liquids in Pipe

(continued)



Reynolds Number for Liquid Flow

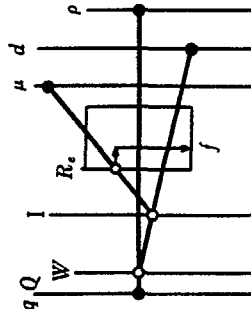
Friction Factor for Clean Steel and Wrought Iron Pipe

Reynolds number may be calculated from the formula below, or, from the nomograph on the opposite page. The nomograph is a graphical solution of the formula.

$$R_e = 22\,700 \frac{q\rho}{d\mu} = 50.6 \frac{Q\rho}{d\mu} = 6.31 \frac{W}{d\mu}$$

(For values of d , see pages B-16 to B-18)

The friction factor for clean steel and wrought iron pipe can be obtained from the chart in the center of the nomograph. Friction factors for other types of pipe can be determined by using the calculated Reynolds number and referring to pages A-23 and A-24.



Example 1

Given: Water at 200 F flows through 4-inch Schedule 40 steel pipe at a rate of 415 gallons per minute.

Find: The flow rate in pounds per hour, the Reynolds number, and the friction factor.

Solution:

1. $\rho = 60.13$ page A-6
2. $\mu = 0.30$ page A-3

Connect		Read
$Q = 415$	$\rho = 60.13$	$W = 200\,000$
$W = 200\,000$	4" Sched 40	Index
Index	$\mu = 0.30$	$R_e = 1\,000\,000$
$R_e = 1\,000\,000$	horizontally to, $d = 4.03$	$f = 0.017$

Example 2

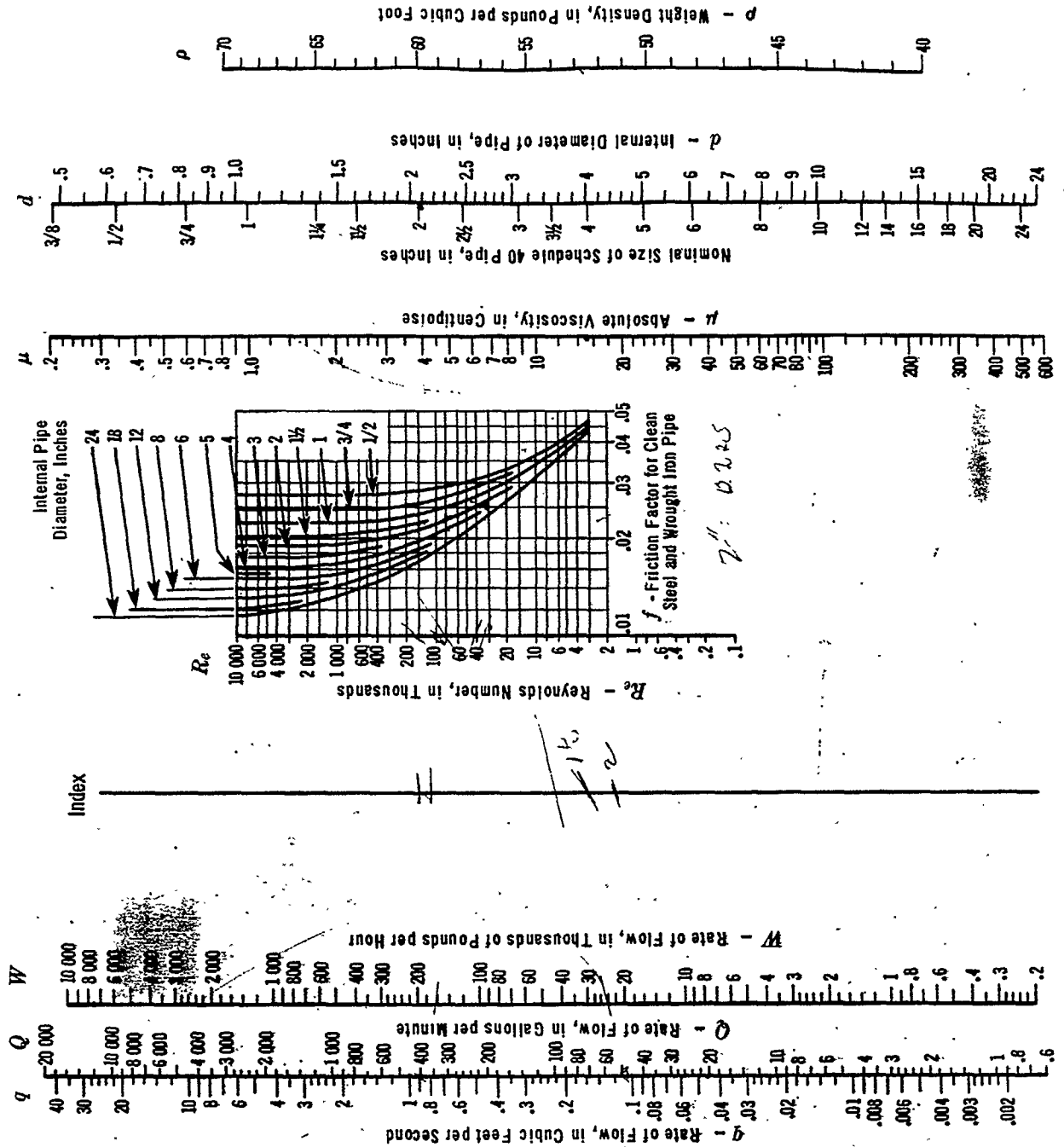
Given: Fuel Oil No. 3 at 60 F flows through 2-inch Schedule 40 steel pipe at a rate of 100 gallons per minute.

Find: The flow rate in pounds per hour, the Reynolds number, and the friction factor.

Solution:

1. $\rho = 56.02$ page A-7
2. $\mu = 9.4$ page A-3

Connect		Read
$Q = 100$	$\rho = 56.02$	$W = 45\,000$
$W = 45\,000$	2" Sched 40	Index
Index	$\mu = 9.4$	$R_e = 14\,600$
$R_e = 14\,600$	horizontally to, $d = 2.07$	$f = 0.03$



Pressure Drop in Liquid Lines for Turbulent Flow

Example 1

Given: Water at 200 F flows through 4-inch Schedule 40 new steel pipe at a rate of 200,000 pounds per hour.

Find: The pressure drop per 100 feet of pipe.

Solution:

1. $\rho = 60.13$ page A-6
2. $\mu = 0.30$ page A-3
3. $f = 0.017$ Example 1, page 3-8
4. $Q = 415$ Example 1, page 3-8

Connect		Read
$f = 0.017$	$\rho = 60.13$	Index 1
Index 1	$Q = 415$	Index 2
Index 2	4" Sched 40	$\Delta P_{100} = 3.6$

- 5.
- 6.
- 7.

Example 2

Given: No. 3 Fuel Oil at 60 F flows through a 2-inch Schedule 40 pipe at a velocity of 10 feet per second.

Find: The pressure drop per 100 feet of pipe.

Solution:

1. $\rho = 56.02$ page A-7
2. $Q = 100$ Example 1, Step 3; page 3-8
3. $f = 0.030$ Example 2, page 3-8

Connect		Read
$\rho = 56.02$	$f = 0.030$	Index 1
Index 1	$Q = 100$	Index 2
Index 2	2" Sched 40	$\Delta P_{100} = 10$

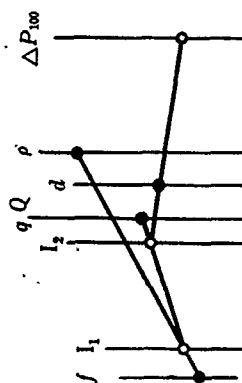
- 4.
- 5.
- 6.

The pressure drop of flowing liquids can be calculated from the Darcy formula that follows, or, from the nomograph on the opposite page. The nomograph is a graphical solution of the formula.

$$\Delta P_{100} = 0.1294 \frac{f \rho W^2}{d^5} = 4350 \frac{f \rho Q^2}{d^5}$$

$$\Delta P_{100} = 0.0216 \frac{f \rho Q^2}{d^5} = 0.000336 \frac{f W^2}{d^5 \rho}$$

(For values of d and d^5 , see pages B-16 to B-18)



When flow rate is given in pounds per hour (W), use the following equation to convert to gallons per minute (Q), or use the nomograph on the preceding page.

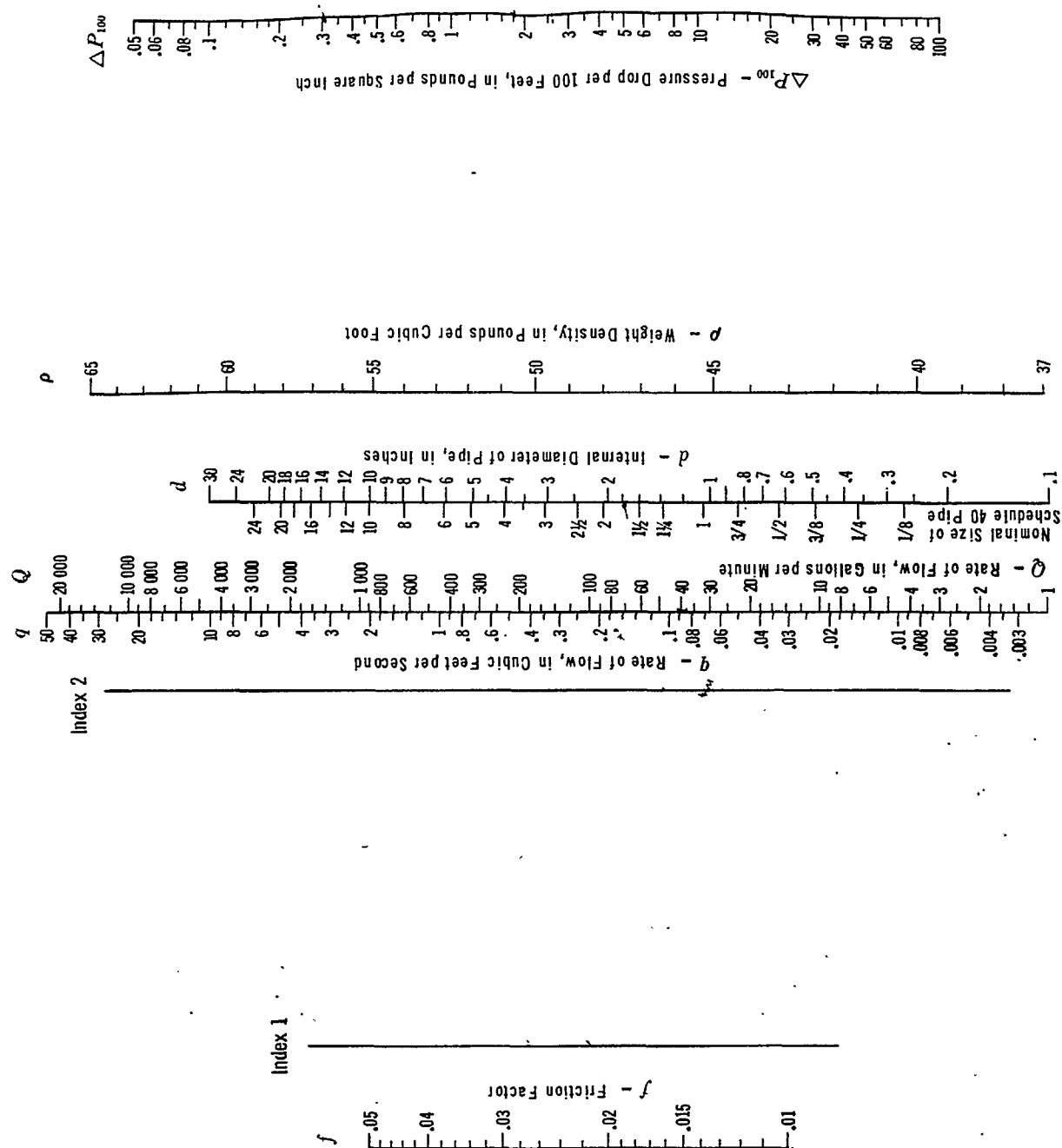
$$Q = \frac{W}{8.02 \rho}$$

For Reynolds number less than 2000, flow is considered laminar and the nomograph on page 3-13 should be used.

The pressure drop per 100 feet and the velocity in Schedule 40 pipe, for water at 60 F, have been calculated for commonly used flow rates for pipe sizes of $\frac{1}{8}$ to 24-inch; these values are tabulated on page B-14.

Pressure Drop in Liquid Lines for Turbulent Flow

(continued)



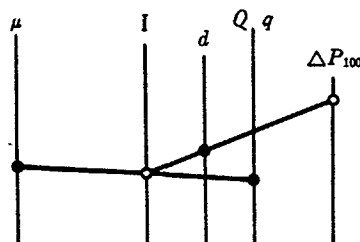
Pressure Drop in Liquid Lines for Laminar Flow

Pressure drop can be calculated from the formula below, or, from the nomograph on the opposite page, only when the flow is laminar. The nomograph is a graphical solution of the formula.

Flow is considered to be laminar at Reynolds number of 2000 or less; therefore, before using the formula or nomograph, determine the Reynolds number from the formula on page 3-2 or the nomograph on page 3-9.

$$\Delta P_{100} = 0.0668 \frac{\mu v}{d^2} = 12.25 \frac{\mu q}{d^4} = 0.0273 \frac{\mu Q}{d^4}$$

(For values of d^2 and d^4 , see pages B-16 to B-18)



Example 1

Given: SAE 30 Lube Oil at 60 F flows through a 6-inch Schedule 40 steel pipe at a rate of 500 gallons per minute.

Find: The pressure drop per 100 feet of pipe.

Solution:

1. $\rho = 56.02$ page A-7
2. $\mu = 450$ page A-3
3. $R_e = 550$ page 3-9
4. Since $R_e < 2000$, the flow is laminar and the nomograph on the opposite page may be used.

	Connect		Read
5.	$\mu = 450$	$Q = 500$	Index
6.	Index	6" Sched 40	$\Delta P_{100} = 4.5$

Example 2

Given: SAE 10 Lube Oil at 60 F flows in a 3-inch Schedule 40 pipe at a velocity of 5 feet per second.

Find: The flow rate in gallons per minute and the pressure drop per 100 feet of pipe.

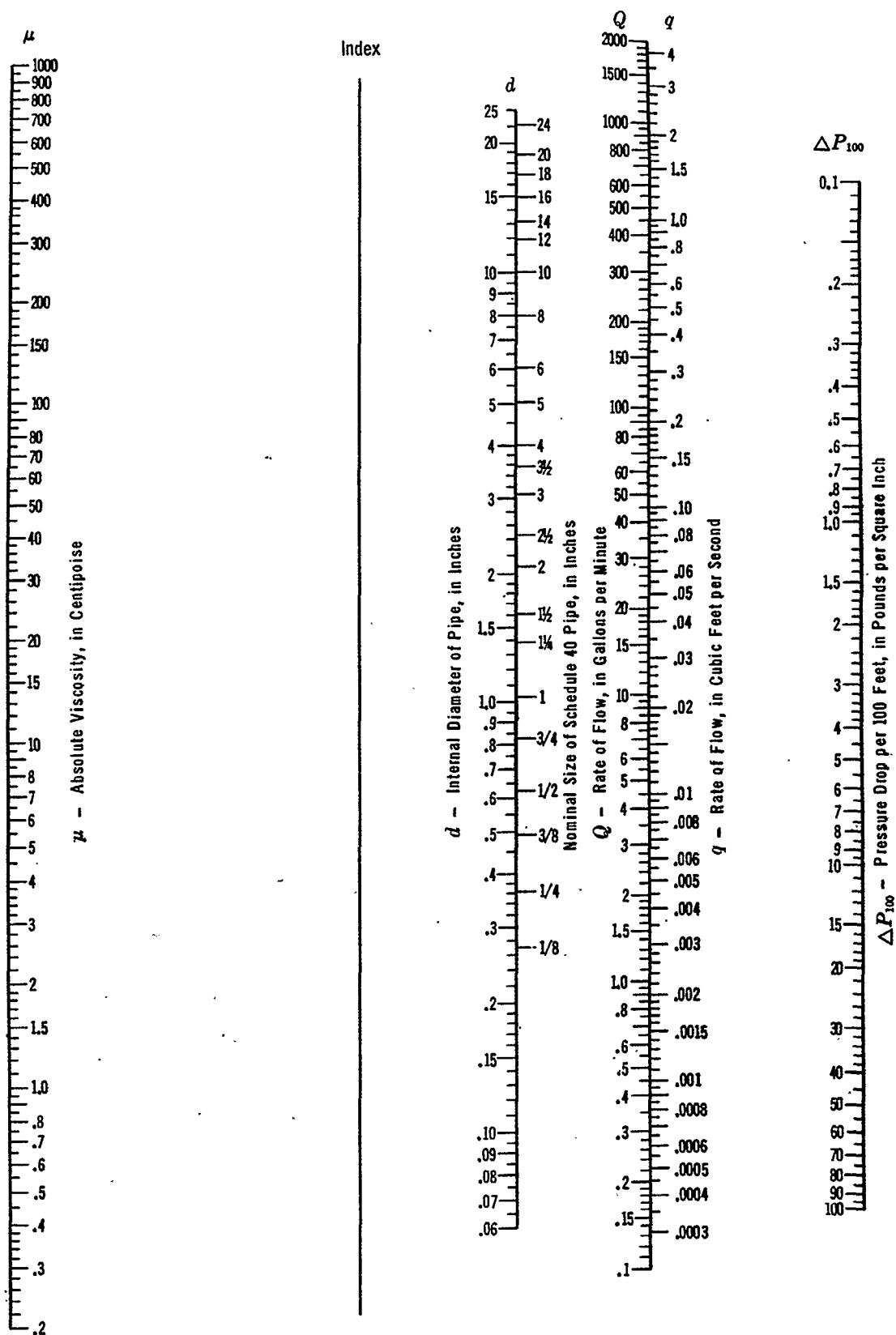
Solution:

1. $\rho = 54.64$ page A-7
2. $Q = 115$ page 3-7
3. $\mu = 95$ page A-3
4. $R_e = 1100$ page 3-9
5. Since $R_e < 2000$, the flow is laminar and the nomograph on the opposite page may be used.

	Connect		Read
6.	$\mu = 95$	$Q = 115$	Index
7.	Index	3" Sched 40	$\Delta P_{100} = 3.4$

Pressure Drop in Liquid Lines for Laminar Flow

(continued)



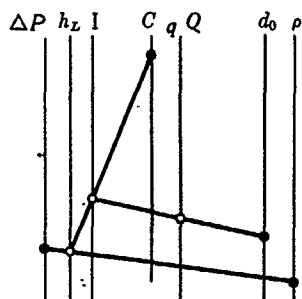
Flow of Liquids Through Nozzles and Orifices

The flow of liquids through nozzles and orifices can be determined from the following formula, or, from the nomograph on the opposite page. The nomograph is a graphical solution of the formula.

$$q = 0.0438 d_o^2 C \sqrt{h_L} = 0.525 d_o^2 C \sqrt{\frac{\Delta P}{\rho}}$$

$$Q = 19.65 d_o^2 C \sqrt{h_L} = 236 d_o^2 C \sqrt{\frac{\Delta P}{\rho}}$$

Head loss or pressure drop is measured across the flange taps.



Example 1

Given: A differential pressure of 2.5 psi is measured across the flange taps of a 2.000-inch I.D. nozzle assembled in a 3-inch Schedule 80 steel pipe carrying water at 60 F.

Find: The flow rate in gallons per minute.

Solution:

1. $d_1 = 2.900$ 3" Sched 80 pipe; page B-17
2. $d_o/d_1 = (2.000 \div 2.900) = 0.69$
3. $C = 1.07$.. turbulent flow assumed; page A-19
4. $\rho = 62.34$ page A-6

Connect		Read
$\Delta P = 2.5$	$\rho = 62.34$	$h_L = 5.8$
$h_L = 5.8$	$C = 1.07$	Index
Index	$d_o = 2.000$	$Q = 200$

5. Calculate R_e based on I.D. of pipe (2.900").
6. $\mu = 1.1$ page A-3
7. $R_e = 200\ 000$ page 3-9
8. $C = 1.07$.. correct for $R_e = 200\ 000$; page A-19
9. When the C factor assumed in Step 3 is not in agreement with page A-19, for the Reynolds number based on the calculated flow, the factor must be adjusted until reasonable agreement is reached by repeating Steps 3 to 11 inclusive.

Example 2

Given: The flow of water, at 60 F through a 6-inch Schedule 40 pipe, is to be restricted to 225 gpm by means of a square edged orifice, across which there will be a differential head of 4 feet of water.

Find: The size of the orifice opening.

Solution:

1. $\rho = 62.34$ page A-6
2. $\mu = 1.1$ page A-3
3. $R_e = 105\ 000 = (1.05 \times 10^5)$.. page 3-9
4. Assume a ratio of d_o/d_1 , say 0.50
5. $d_1 = 6.065$ page B-16
6. $d_o = 0.50 d_1 = (0.50 \times 6.065) = 3.033$
7. $C = 0.624$ page A-19

Connect		Read
$h_L = 4.0$	$C = 0.624$	Index
Index	$Q = 225$	$d_o = 3"$

8. An orifice diameter of 3 inches will be satisfactory, since this is reasonably close to the assumed value used in Step 6.
9. If the value of d_o determined from the nomograph is smaller than the assumed value used in Step 6, repeat Steps 6 to 10 inclusive, using reduced assumed values for d_o until it is in reasonable agreement with the value determined in Step 9.

Example 3

Given: A differential pressure of 0.5 psi is measured across flange taps of a 1.000-inch I.D. square edged orifice assembled in 1¼-inch Schedule 80 steel pipe carrying SAE 30 lubricating oil at 60 F.

Find: The flow rate in cubic feet per second.

Solution:

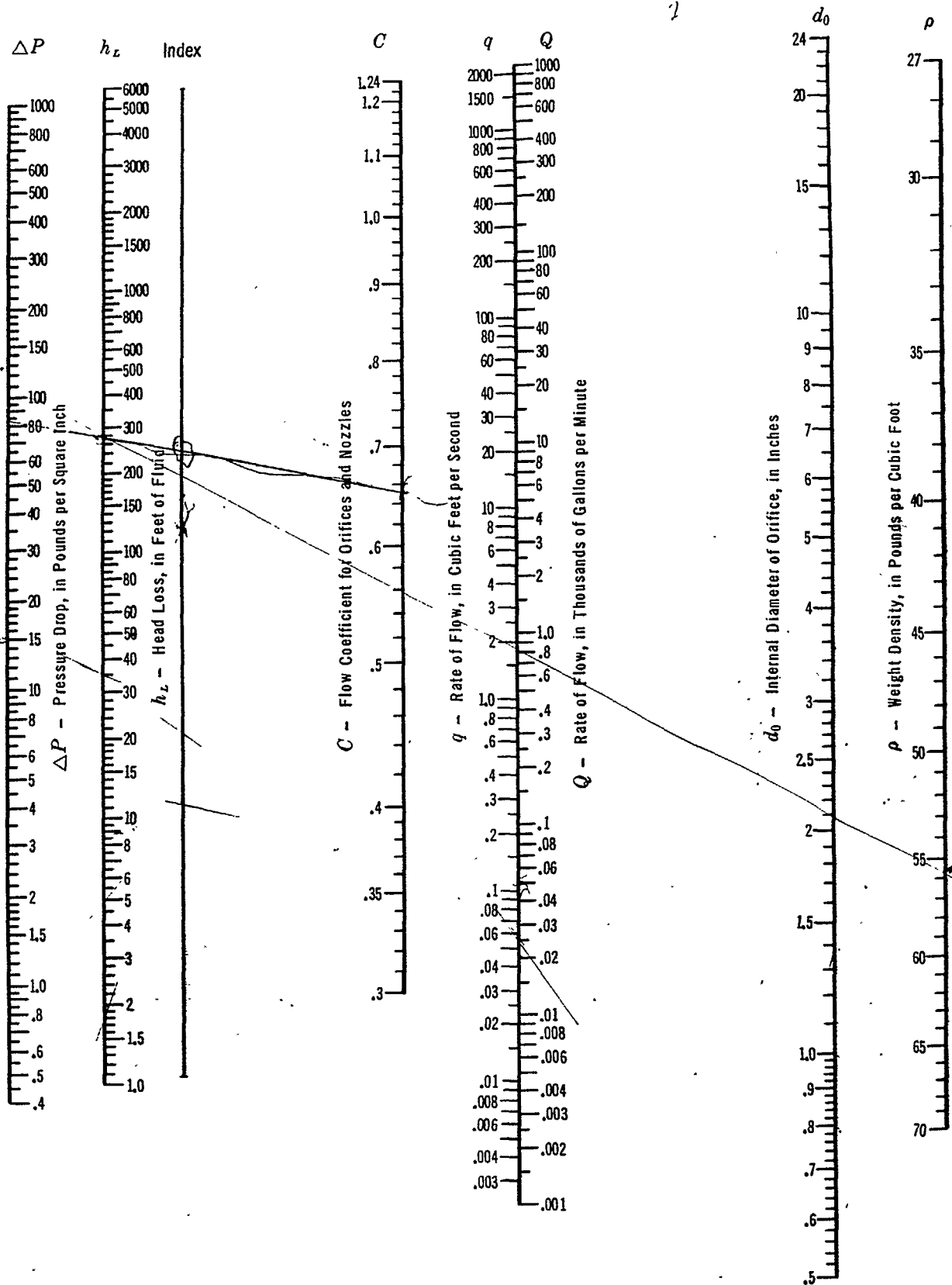
1. $\rho = 56.02$ page A-7
2. $d_1 = 1.278$ 1¼" Sched 80 pipe; page B-16
3. $d_o/d_1 = (1.000 \div 1.278) = 0.783$
4. $\mu = 450$ suspect flow is laminar since viscosity is high; page A-3
5. $C = 1.0$ assumed; page A-19

Connect		Read
$\Delta P = 0.5$	$\rho = 56.02$	$h_L = 1.3$
$h_L = 1.3$	$C = 1.0$	Index
Index	$d_o = 1.0$	$q = 0.05$

6. Calculate R_e based on I.D. of pipe (1.278").
7. $R_e = 110$ page 3-9
8. $C = 1.0$.. correct for $R_e = 110$; page A-19
9. When the C factor assumed in Step 5 is not in agreement with page A-19, for the Reynolds number based on the calculated flow, it must be adjusted until reasonable agreement is reached by repeating Steps 5 to 11 inclusive.

Flow of Liquids Through Nozzles and Orifices

(continued)



Velocity of Compressible Fluids in Pipe

The mean velocity of compressible fluids in pipe can be computed by means of the following formula, or, by using the nomograph on the opposite page. The nomograph is a graphical solution of the formula.

$$V = \frac{3.06 W \bar{V}}{d^2 \rho} = \frac{3.06 W}{d^2 \rho}$$

(For values of d^2 , see pages B-16, to B-18)

Example 1

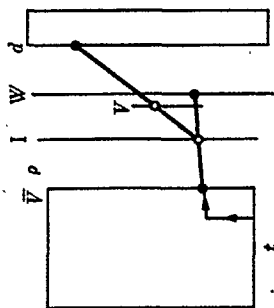
Given: Steam at 600 pounds per square inch gauge and 850 F is to flow through a Schedule 80 pipe at a rate of 30,000 pounds per hour with the velocity limited to 8,000 feet per minute.

Find: The suitable pipe size and the velocity through the pipe.

Solution:

Connect		Read
850 F	vertically to	600 psig
600 psig	horizontally to	$\bar{V} = 1.21$
$\bar{V} = 1.21$	$W = 30\ 000$	Index
Index	$V = 8\ 000$	$d = 3.7$
4" Schedule 80 pipe is suitable.		
Index	4" Sched 80 pipe	$V = 7\ 600$

- 1.
- 2.
- 3.
- 4.
- 5.
- 6.



Example 2

Given: Air at 400 pounds per square inch gauge and 60 F flows through a 1½-inch Schedule 40 pipe at the rate of 144,000 cubic feet per hour at standard conditions (14.7 psia and 60 F).

Find: The flow rate in pounds per hour and the velocity in feet per minute.

Solution:

1. $W = 11\ 000$, using $S_g = 1.0$ page B-2
2. $\rho = 2.16$ page A-10

Connect		Read
$\rho = 2.16$	$W = 11\ 000$	Index
Index	1½" Sched 40	$V = 6\ 000$

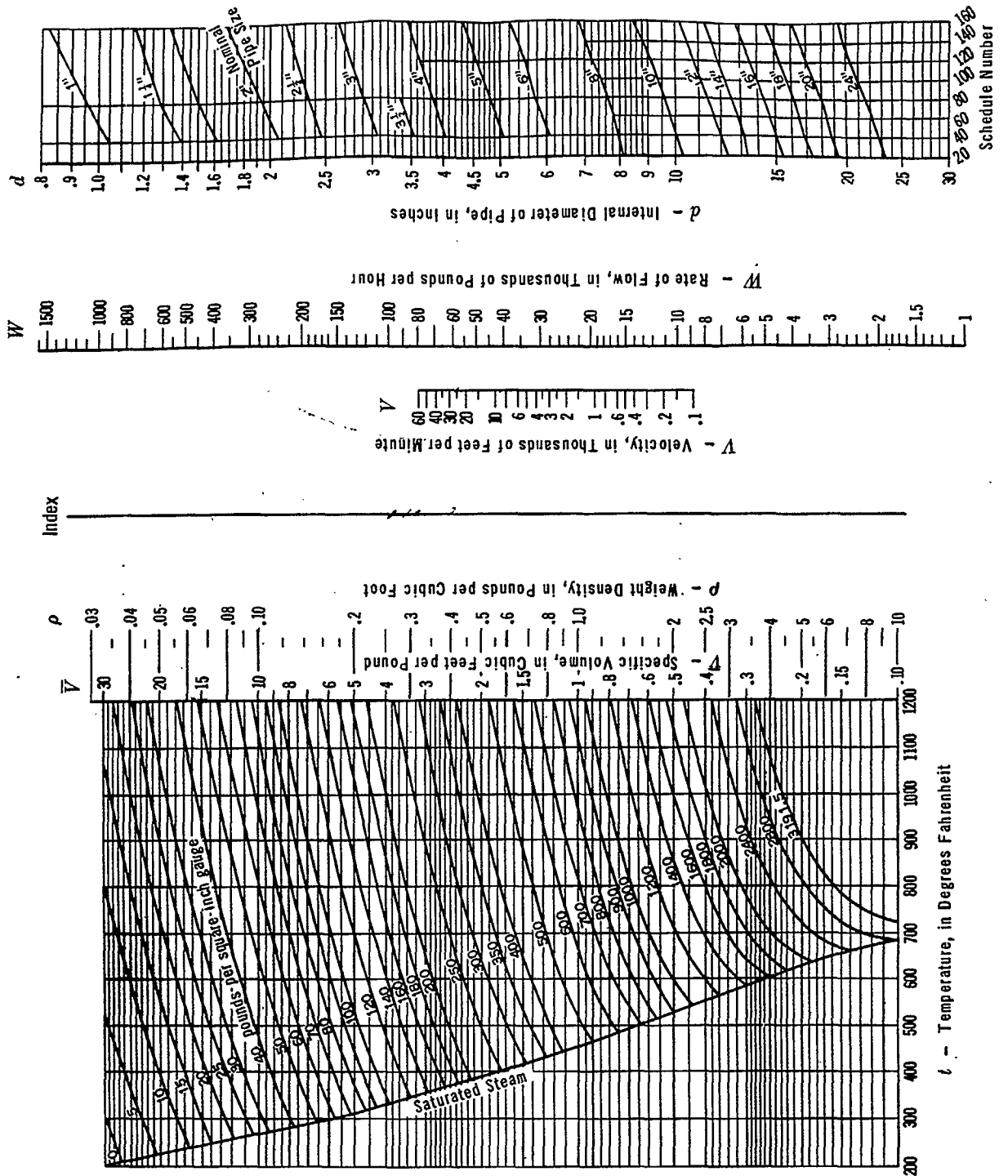
- 3.
- 4.

Reasonable Velocities for Flow of Steam Through Pipe

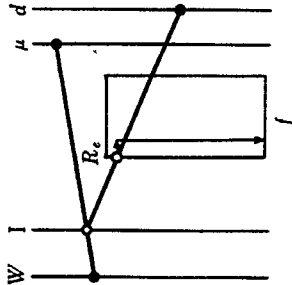
Condition of Steam	Pressure (P) Psig	Service	Reasonable Velocity (V) Feet per Minute	
			Feet per Minute	
Saturated	0 to 25	Heating (short lines)	4 000 to	6 000
	25 and up	Power house equipment, process piping, etc.	6 000 to	10 000
Superheated	200 and up	Boiler and turbine leads, etc.	7 000 to	20 000

Velocity of Compressible Fluids in Pipe

(continued)



Reynolds Number for Compressible Flow Friction Factor for Clean Steel and Wrought Iron Pipe

**Example 2**

Given: Steam at 600 psig and 850 F flows through a 4-inch Schedule 80 steel pipe at a rate of 30,000 pounds per hour.

Find: The Reynolds number and the friction factor.

Solution:

1. $d = 3.826$ page B-17
2. $\mu = 0.029$ page A-2

Connect		Read
$W = 30\,000$	$\mu = 0.029$	Index
Index	$d = 3.826$	$R_e = 1\,700\,000$
$R_e = 1\,700\,000$	horizontally to: $d = 3.826$	$f = 0.017$

- 3.
- 4.
- 5.

The Reynolds number may be determined from the formula below or from the nomograph on the opposite page. The nomograph is a graphical solution of the formula.

The friction factor for clean steel and wrought iron pipe can be obtained from the chart in the center of the nomograph. Friction factors for other types of pipe can be determined by using the calculated Reynolds number and referring to pages A-23 and A-24.

$$R_e = 6.31 \frac{W}{d\mu} = 0.482 \frac{q' S_g}{d\mu}$$

(For values of d , see pages B-16 to B-18)

Example 1

Given: Natural gas at 250 psig and 60 F, with a specific gravity of 0.75, flows through an 8-inch Schedule 40 clean steel pipe at a rate of 1,200,000 cubic feet per hour at standard conditions (scfh).

Find: The flow rate in pounds per hour, the Reynolds number, and the friction factor.

Solution:

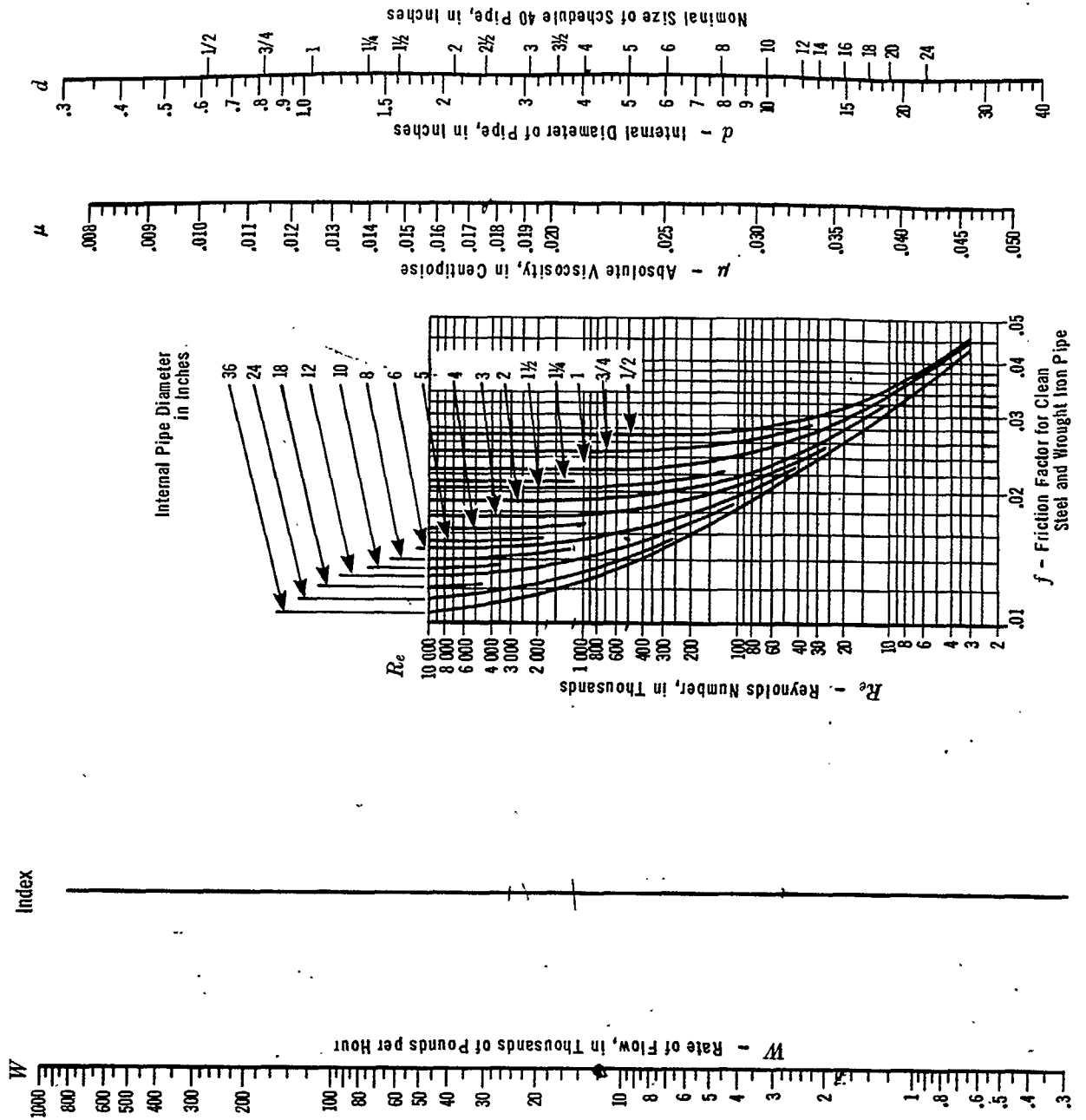
1. $W = 69\,000$, using $S_g = 0.75$ page B-2
2. $\mu = 0.011$ page A-5

Connect		Read
$W = 69\,000$	$\mu = 0.011$	Index
Index	8" Sched 40 pipe	$R_e = 5\,000\,000$
$R_e = 5\,000\,000$	horizontally to: 8" I.D.	$f = 0.014$

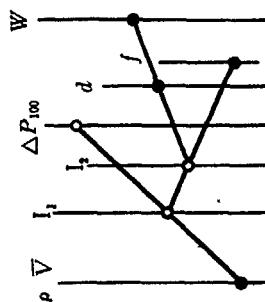
- 3.
- 4.
- 5.

Note: Flowing pressure of gases has a negligible effect upon viscosity, Reynolds number, and friction factor.

Reynolds Number for Compressible Flow Friction Factor for Clean Steel and Wrought Iron Pipe (continued)



Pressure Drop in Compressible Flow Lines



The pressure drop of flowing compressible fluids can be calculated from the Darcy formula below, or, from the nomograph on the opposite page. The nomograph is a graphical solution of the formula.

$$\Delta P_{100} = 0.000336 \frac{f W^2 \bar{V}}{d^5} = 0.000336 \frac{f W^2}{d^5 \rho}$$

$$\Delta P_{100} = 0.0000001959 \frac{f (q'_h)^2 S_g^2}{d^5 \rho}$$

(For values of d^5 , see pages B-16 to B-18)

When the flow rate is given in cubic feet per hour at standard conditions (q'_h), use the following equation or the nomograph on page B-2 to convert to pounds per hour (W).

$$W = 0.0764 q'_h S_g$$

Air: For pressure drop, in pounds per square inch per 100 feet of Schedule 40 pipe, for air at 100 psig and 60 F, see page B-15.

Example 1

Given: Steam at 600 psig and 850 F flows through a 4-inch Schedule 80 steel pipe at a rate of 30,000 pounds per hour.

Find: The pressure drop per 100 feet of pipe.

Solution:

1. $d = 3.826$ page B-17
2. $\mu = 0.029$ page A-2
3. $f = 0.017$ page 3-19
4. $\bar{V} = 1.21$ page 3-17 or A-17

Connect		Read
$W = 30\,000$	$d = 3.826$	Index 2
Index 2	$f = 0.017$	Index 1
Index 1	$\bar{V} = 1.21$	$\Delta P_{100} = 7.5$

- 5.
- 6.
- 7.

Example 2

Given: Natural gas at 250 psig and 60 F flows through an 8-inch Schedule 40 pipe at a rate of 1,200,000 standard cubic feet per hour; its specific gravity is 0.75.

Find: The flow rate in pounds per hour and the pressure drop per 100 feet of pipe.

Solution:

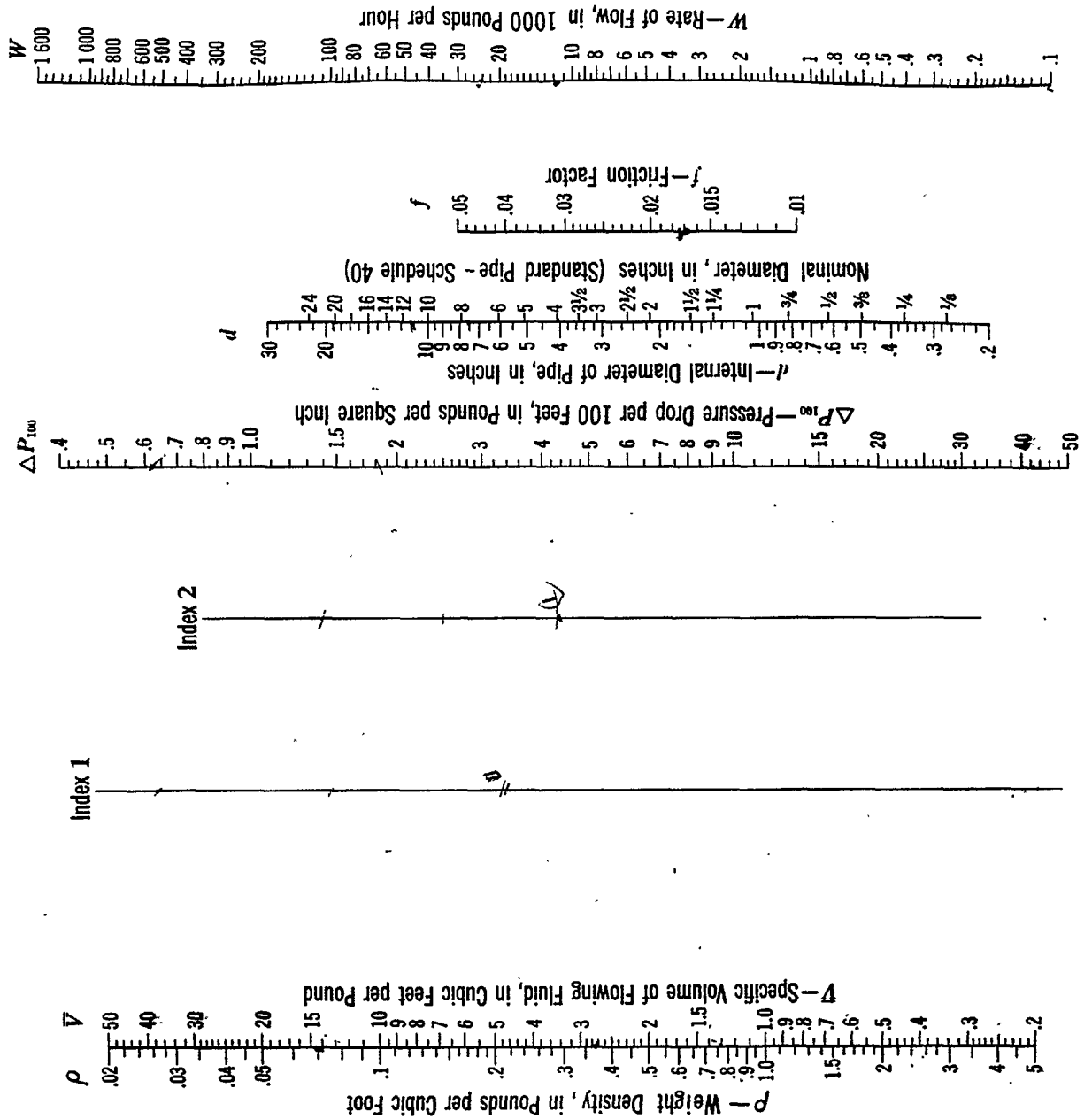
1. $W = 69\,000$ using $S_g = 0.75$; page B-2
2. $\mu = 0.011$ page A-5
3. $f = 0.014$ page 3-19
4. $\rho = 1.03$ page A-10

Connect		Read
$W = 69\,000$	8" Sched 40 pipe	Index 2
Index 2	$f = 0.014$	Index 1
Index 1	$\rho = 1.03$	$\Delta P_{100} = 0.68$

- 5.
- 6.
- 7.

Pressure Drop in Compressible Flow Lines

(continued)



Simplified Flow Formula for Compressible Fluids

Pressure Drop, Rate of Flow, and Pipe Size

The simplified flow formula for compressible fluids is accurate for fully turbulent flow; in addition, its use provides a good approximation in calculations involving compressible fluid flow through wrought iron or commercial steel pipe for most normal flow conditions.

If velocities are low, friction factors assumed in the simplified formula may be too low; in such cases, the formula and nomograph shown on pages 3-20 and 3-21 may be used to provide greater accuracy.

The Darcy formula can be written in the following form:

$$\Delta P_{100} = W^2 \left(\frac{0.000336f}{d^5} \right) \bar{V} = (W^2 10^{-9}) \left(\frac{336000f}{d^5} \right) \bar{V}$$

$$C_1 = W^2 10^{-9} \quad C_2 = \frac{336000f}{d^5}$$

The simplified flow formula can then be written:

$$\Delta P_{100} = C_1 C_2 \bar{V} = \frac{C_1 C_2}{\rho}$$

$$C_1 = \frac{\Delta P_{100}}{C_2 \bar{V}} = \frac{\Delta P_{100} \rho}{C_2} \quad C_2 = \frac{\Delta P_{100}}{C_1 \bar{V}} = \frac{\Delta P_{100} \rho}{C_1}$$

C_1 = discharge factor from chart at right. *1.45*
 C_2 = size factor, from table on next page. *0.146*

The limitations of the Darcy formula for compressible flow, as outlined on page 3-3, apply also to the simplified flow formula.

Example 1

Given: Steam at 345 psig and 500 F flows through 8-inch Schedule 40 pipe at a rate of 240,000 pounds per hour.

Find: The pressure drop per 100 feet of pipe.

Solution: $C_1 = 57$
 $C_2 = 0.146$
 $\bar{V} = 1.45$ page 3-17 or A-15
 $\Delta P_{100} = 57 \times 0.146 \times 1.45 = 12$

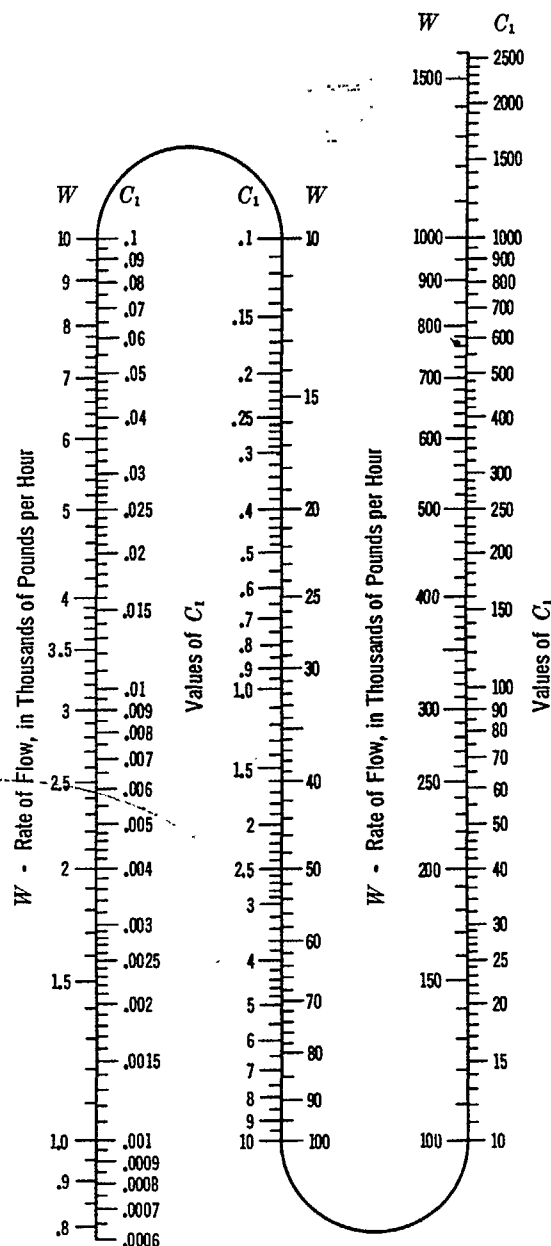
Example 2

Given: Pressure drop is 5 psi with 100 psig air at 90 F flowing through 100 feet of 4-inch Schedule 40 pipe.

Find: The flow rate in standard cubic feet per minute.

Solution: $\Delta P_{100} = 5.0$
 $C_2 = 5.17$
 $\rho = 0.564$ page A-10
 $C_1 = (5.0 \times 0.564) \div 5.17 = 0.545$
 $W = 23000$
 $q'_m = W \div (4.58 S_g)$ page B-2
 $q'_m = 23000 \div (4.58 \times 1.0) = 5000$ scfm

Values of C_1



For C_2 values and an example on "determining pipe size", see the opposite page.

Simplified Flow Formula for Compressible Fluids

Pressure Drop, Rate of Flow, and Pipe Size — continued

Values of C_2								
Nominal Pipe Size Inches	Schedule Number	Value of C_2	Nominal Pipe Size Inches	Schedule Number	Value of C_2	Nominal Pipe Size Inches	Schedule Number	Value of C_2
$\frac{1}{8}$	40 s	7 920 000.	5	40 s	1.59	16	10	0.004 63
	80 x	26 200 000.		80 x	2.04		20	0.004 21
$\frac{1}{4}$	40 s	1 590 000.	6	120	2.69	18	30 s	0.005 04
	80 x	4 290 000.		160	3.59		40 x	0.005 49
$\frac{3}{8}$	40 s	319 000.	8	... xx	4.93	20	60	0.006 12
	80 x	718 000.		40 s	0.610		80	0.007 00
$\frac{1}{2}$	40 s	93 500.	10	80 x	0.798	24	100	0.008 04
	80 x	186 100.		120	1.015		120	0.009 26
$\frac{3}{4}$	160	4 300 000.	12	160	1.376	28	140	0.010 99
	... xx	11 180 000.		... xx	1.861		160	0.012 44
1	40 s	21 200.	14	20	0.133	32	10	0.002 47
	80 x	36 900.		30	0.135		20	0.002 56
$1\frac{1}{4}$	160	100 100.	16	40 s	0.146	36	... s	0.002 66
	... xx	627 000.		60	0.163		30	0.002 76
$1\frac{1}{2}$	40 s	5 950.	18	80 x	0.185	40	... x	0.002 87
	80 x	9 640.		100	0.211		40	0.002 98
2	160	22 500.	20	120	0.252	48	60	0.003 35
	... xx	114 100.		140	0.289		80	0.003 76
$2\frac{1}{2}$	40 s	1 408.	24	... xx	0.317	56	100	0.004 38
	80 x	2 110.		160	0.333		120	0.005 04
3	160	3 490.	28	20	0.039 7	64	140	0.005 73
	... xx	13 640.		30	0.042 1		160	0.006 69
$3\frac{1}{2}$	40 s	627.	32	40 s	0.044 7	72	10	0.001 41
	80 x	904.		60 x	0.051 4		20 s	0.001 50
4	160	1 656.	36	80	0.056 9	80	30 x	0.001 61
	... xx	4 630.		100	0.066 1		40	0.001 69
$4\frac{1}{2}$	40 s	169.	40	120	0.075 3	96	60	0.001 91
	80 x	236.		140	0.090 5		80	0.002 17
5	160	488.	48	160	0.105 2	112	100	0.002 51
	... xx	899.		20	0.015 7		120	0.002 87
$5\frac{1}{2}$	40 s	66.7	56	30	0.016 8	128	140	0.003 35
	80 x	91.8		... s	0.017 5		160	0.003 85
6	160	146.3	64	40	0.018 0	144	10	0.000 534
	... xx	380.0		... x	0.019 5		20 s	0.000 565
$6\frac{1}{2}$	40 s	21.4	72	60	0.020 6	160	30 x	0.000 597
	80 x	28.7		80	0.023 1	176	40	0.000 614
7	160	48.3	80	100	0.026 7	192	60	0.000 651
	... xx	96.6		120	0.031 0		80	0.000 835
$7\frac{1}{2}$	40 s	10.0	96	140	0.035 0	200	100	0.000 972
	80 x	37.7		160	0.042 3	224	120	0.001 119
8	40 s	5.17	112	20	0.009 49	240	140	0.001 274
	80 x	6.75		30 s	0.009 96	256	160	0.001 478
$8\frac{1}{2}$	160	8.94	128	40	0.010 46	272	80	0.000 835
	... xx	18.59		... x	0.011 55	288	100	0.000 972
9	40 s	5.17	144	60	0.012 44	304	120	0.001 119
	80 x	6.75		80	0.014 16	320	140	0.001 274
$9\frac{1}{2}$	160	8.94	160	100	0.016 57	336	160	0.001 478
	... xx	18.59		120	0.018 98	352	80	0.000 835
10	40 s	5.17	180	140	0.021 8	368	100	0.000 972
	80 x	6.75		160	0.025 2	384	120	0.001 119

Note
The letters s, x, and xx in the columns of Schedule Numbers indicate Standard, Extra Strong, and Double Extra Strong pipe respectively.

Example 3

Given: An 85 psig saturated steam line with 20,000 pounds per hour flow is permitted a maximum pressure drop of 10 psi per 100 feet of pipe.

Find: The smallest size of Schedule 40 pipe suitable.

Solution: $\Delta P_{100} = 10$ $\bar{V} = 4.5$ page 3-17 or A-13
 $C_1 = 0.4$ $C_2 = 10 \div (0.4 \times 4.5) = 5.56$

Reference to the table of C_2 values above shows that the 4-inch size is the smallest Schedule 40 pipe having a C_2 value less than 5.56.

The actual pressure drop per 100 feet of 4-inch Schedule 40 pipe is:

$$\Delta P_{100} = 0.4 \times 5.17 \times 4.5 = 9.3$$

Flow of Compressible Fluids Through Nozzles and Orifices

The flow of compressible fluids through nozzles and orifices can be determined from the following formula, or, by using the nomograph on the next page. The nomograph is a graphical solution of the formula.

$$w = 0.525 Y d_0^2 C \sqrt{\Delta P \rho_1} = 0.525 Y d_0^2 C \sqrt{\frac{\Delta P}{V_1}}$$

$$W = 1891 Y d_0^2 C \sqrt{\Delta P \rho_1} = 1891 Y d_0^2 C \sqrt{\frac{\Delta P}{V_1}}$$

(Pressure drop is measured across the flange taps)

Example 1

Given: A differential pressure of 11.5 psi is measured across the flange taps of a 1.000-inch I.D. nozzle assembled in a 2-inch Schedule 40 steel pipe, in which, dry carbon dioxide (CO₂) gas is flowing at 100 psig pressure and 200 F.

Find: The flow rate in cubic feet per hour at standard conditions (scfh).

Solution:

1. $R = 35.1$
2. $S_g = 1.516$
3. $k = 1.28$

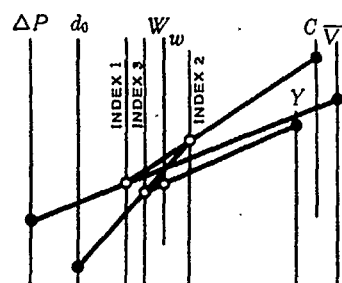
Steps 3 through 7 are used to determine the Y factor.

4. $P'_1 = P + 14.7 = 100 + 14.7 = 114.7$
5. $\Delta P/P'_1 = 11.5 \div 114.7 = 0.1003$
6. $d_1 = 2.067$ 2" Sched 40 pipe; page B-16
7. $d_0/d_1 = 1.00 \div 2.067 = 0.484$
8. $Y = 0.93$ page A-20
9. $C = 1.003$.. turbulent flow assumed; page A-19
10. $T = 460 + t = 460 + 200 = 660$
11. $\rho_1 = 0.71$ page A-10

	Connect		Read
12.	$\Delta P = 11.5$	$\rho_1 = 0.71$	Index 1
13.	Index 1	$C = 1.003$	Index 2
14.	Index 2	$d_0 = 1.000$	Index 3
15.	Index 3	$Y = 0.93$	$W = 5000$

16. $q'_h = 44\,000$ scfh page B-2
17. $\mu = 0.018$ page A-5
18. $R_e = 860\,000$ or 8.6×10^5 page 3-21
19. $C = 1.003$ is correct for
 $R_e = 8.6 \times 10^5$... page A-19

20. When the C factor assumed in Step 9 is not in agreement with page A-19, for the Reynolds number based on the calculated flow, it must be adjusted until reasonable agreement is reached by repeating Steps 9 through 19.



Example 2

Given: A differential pressure of 3 psi is measured across the flange taps of a 0.750-inch I.D. square edged orifice assembled in 1-inch Schedule 40 wrought iron pipe, in which, dry ammonia (NH₃) gas is flowing at 40 psig pressure and 50 F.

Find: The flow rate in pounds per second and in cubic feet per minute at standard conditions (scfm).

Solution:

1. $R = 90.8$
2. $S_g = 0.587$
3. $k = 1.29$

Steps 3 through 7 are used to determine the Y factor.

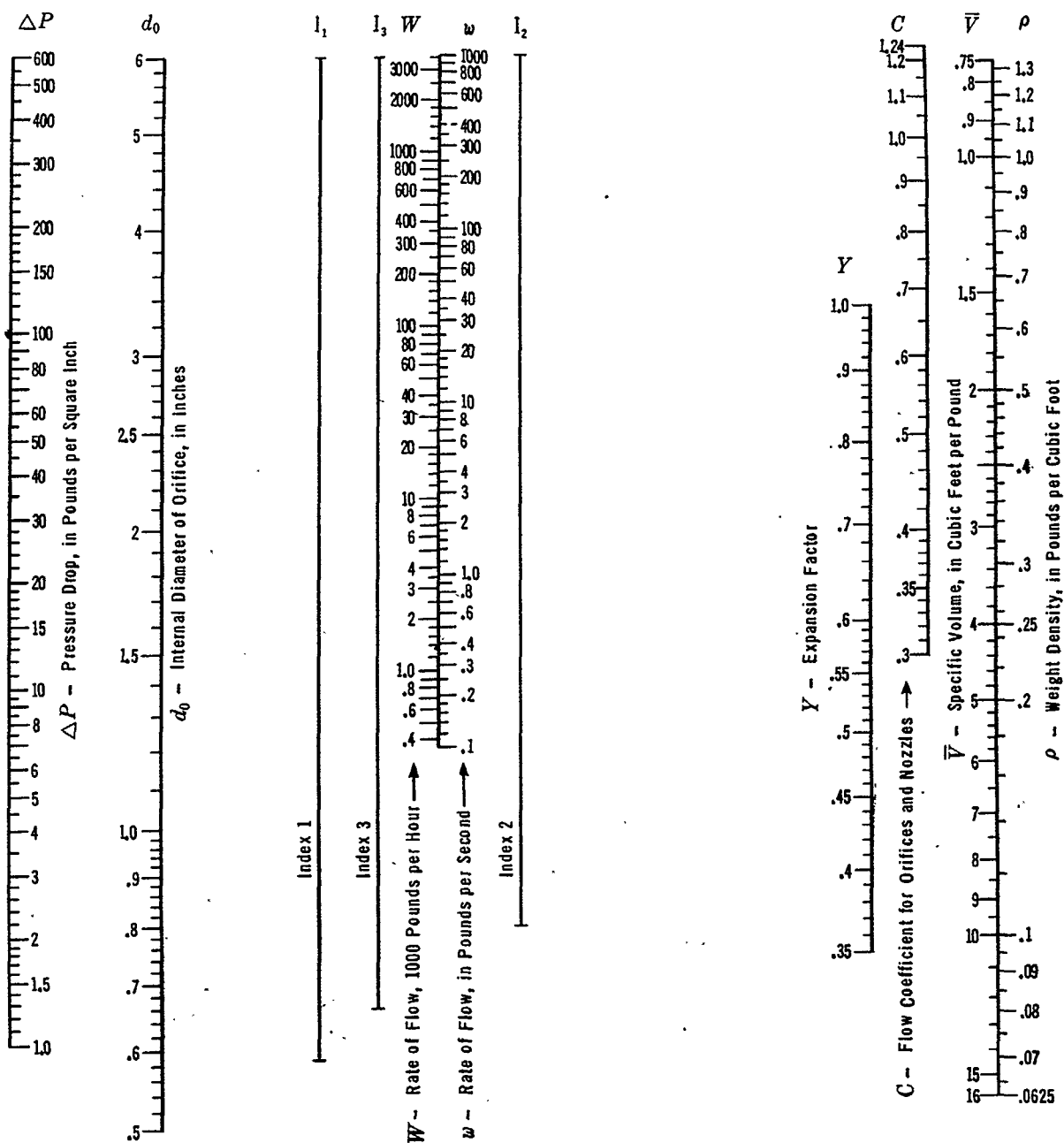
4. $P'_1 = P + 14.7 = 40 + 14.7 = 54.7$
5. $\Delta P/P'_1 = 3.0 \div 54.7 = 0.0549$
6. $d_1 = 1.049$ 1" Sched 40 pipe; page B-16
7. $d_0/d_1 = 0.750 \div 1.049 = 0.716$
8. $Y = 0.98$ page A-20
9. $C = 0.702$.. turbulent flow assumed; page A-19
10. $T = 460 + t = 460 + 50 = 510$
11. $\rho_1 = 0.17$ page A-10

	Connect		Read
12.	$\Delta P = 3.0$	$\rho_1 = 0.17$	Index 1
13.	Index 1	$C = 0.702$	Index 2
14.	Index 2	$d_0 = 0.75$	Index 3
15.	Index 3	$Y = 0.98$	$w = 0.145$
16.	Index 3	$Y = 0.98$	$W = 520$

17. $q'_m = \frac{W}{4.58 S_g} = \frac{520}{4.58 \times 0.587} = 195$.. page B-2
18. $\mu = 0.010$ page A-5
19. $R_e = 310\,000$ or 3.10×10^5 page 3-21
20. $C = 0.702$ is correct for
 $R_e = 3.10 \times 10^5$... page A-19
21. When the C factor assumed in Step 9 is not in agreement with page A-19, for the Reynolds number based on the calculated flow, it must be adjusted until reasonable agreement is reached by repeating Steps 9 through 20.

Flow of Compressible Fluids Through Nozzles and Orifices

(continued)



Examples of Flow Problems

CHAPTER 4

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 simple flow problems were presented in Chapter 3 to illustrate the use of the nomographs. Other 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 and 2; these formulas were rewritten in more commonly used terms for Chapter 3. Use of nomographs, when applicable, are indicated in the solution of these problems.

The controversial subject regarding the selection of a formula most applicable to the flow of gas through long pipe lines 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 or Wrought Iron

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 or wrought iron, such as concrete, wood stave, riveted steel, etc. For relative roughness of these and other piping materials, see page A-23.

Example 4-1 . . . Smooth Pipe (Plastic)

Given: Water at 80 F is flowing through 70 feet of 2-inch standard wall plastic pipe (smooth wall) at a rate of 50 gallons per minute.

Find: The Reynolds number and friction factor.

Solution:

$$1. \quad R_e = \frac{50.6 \, Q \rho}{d \mu} \quad \dots \text{page 3-2}$$

2. $\rho = 62.19 \quad \dots \text{page A-6}$
3. $d = 2.067 \quad \dots \text{page B-18}$
4. $\mu = 0.85 \quad \dots \text{page A-3}$
5. $R_e = \frac{50.6 \times 50 \times 62.19}{2.067 \times 0.85}$
 $R_e = 89 \, 600 \text{ or } 8.96 \times 10^4$
6. $f = 0.0182 \text{ for smooth pipe} \quad \dots \text{page A-24}$

Determination of Valve Resistance In L , L/D , K , and Flow Coefficient C_v

Example 4-2 . . . L , L/D , and K from C_v for Conventional Type Valves

Given: A 6-inch 125-pound Y-pattern globe valve has a flow coefficient, C_v , of 600.

Find: The resistance coefficient K and equivalent lengths L/D and L for fully turbulent flow.

Solution:

1. K , L/D , and L should be given in terms of 6-inch Schedule 40 pipe; see page A-30.

2. $K = \frac{891 d^4}{C_v^2}$ page 3-4 or A-32

3. $d = 6.065$ $d^4 = 1352.8$ page B-16
 $D = 0.5054$ $d^5 = 8206$

4. $K = \frac{891 \times 1352.8}{600^2} = 3.35$ (based on 6" Sched 40 pipe)

5. $\frac{L}{D} = \frac{K}{f}$ page 3-4

6. $f = 0.015$ (for 6.065" I.D. pipe in fully turbulent flow range; page A-25)

7. $\frac{L}{D} = \frac{K}{f} = \frac{3.35}{0.015} = 223$

8. $L = \left(\frac{L}{D}\right)D = 223 \times 0.5054 = 113$

Alternate solution:

9. $L = \frac{74.3 d^5}{f C_v^2}$ page 3-4

$$L = \frac{74.3 \times 8206}{0.015 \times 600^2} = 113$$

Example 4-3 . . . L , L/D , K , and C_v for Conventional Type Valves

Given: A 4-inch 600-pound conventional angle valve with no obstruction in flat seat.

Find: The resistance coefficient K , flow coefficient C_v , and equivalent lengths L/D and L for fully turbulent flow.

Solution:

1. K , L/D , and L should be given in terms of 4-inch Schedule 80 pipe; see page A-30.

2. $\frac{L}{D} = 145$ page A-30

3. $d = 3.826$ page B-16

4. $L = 46$ page A-31

5. $K = 2.4$ based on 4" Sched 80 pipe; page A-31

6. $C_v = 280$ page A-32

Example 4-4 . . . Venturi Type Valves

Given: A 6 × 4-inch 600-pound steel gate valve.

Find: The valve resistance coefficient K , and the equivalent lengths L/D and L , for fully turbulent flow of Reynolds numbers indicated on the Moody Friction Factor diagram; see page A-25.

Solution:

1. K , L/D , and L should be given in terms of 6-inch Schedule 80 pipe; see page A-30.

2. For venturi port gate valves:

$$\left(\frac{L}{D}\right)_a = \left(\frac{L}{D}\right)_b \left(\frac{d_a}{d_b}\right)^4$$

..... page 3-5

$$\left(\frac{L}{D}\right)_a = \left(\frac{L}{D}\right)_b \left(\frac{d_a}{d_b}\right)^4$$

3. For 6-inch Schedule 80 pipe:

$$d = 5.761$$

$$D = 0.4801$$
 page B-17

$$d^4 = 1101.6$$

4. For 4-inch Schedule 80 pipe:

$$d = 3.826$$

$$d^4 = 214.33$$
 page B-17

5. $\frac{L}{D} = 13$ (for 4" constant diameter port gate valve; page A-30)

6. $\left(\frac{L}{D}\right)_a = 13 \left(\frac{1101.6}{214.33}\right)$

$$\left(\frac{L}{D}\right)_a = 66.8$$
 for 6" Sched 80 pipe

7. $K = f \frac{L}{D}$ page 3-4

8. $f = 0.0151$ (for fully turbulent flow, 6" Sched 80 pipe; page A-25)

9. $K = 0.0151 \times 66.8$

$$K = 1.01 \text{ or say } 1.0$$
 based on 6" Sched 80 pipe

10. $L = \left(\frac{L}{D}\right)D = 66.8 \times 0.4801$

$$L = 32.1$$
 for 6" Sched 80 pipe

Check Valves — Determination of Size

Example 4-5... Lift Checks

Given: A globe type lift check valve with a wing-guided disc is required in a 3-inch Schedule 40 horizontal pipe carrying 70 F water at the rate of 100 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 under normal flow conditions; see page 2-7 for discussion.

Solution:

1. Solve Darcy Equation for d^4 :

$$\Delta P = \frac{0.0001799 f(L/D) \rho Q^2}{d^4} \quad \text{page 3-4}$$

$$d^4 = \frac{0.0001799 f(L/D) \rho Q^2}{\Delta P}$$
2. Calculate Reynolds number to determine the friction factor based on flow in 3-inch pipe.

$$R_e = \frac{50.6 Q \rho}{d \mu} \quad \text{page 3-2 or 3-8}$$
3. $\Delta P = 2.0$ page A-30
4. $L/D = 450$ page A-30
5. $\rho = 62.27$ page A-6
6. $d = 3.068$ page B-16
7. $\mu = 0.95$ page A-3
8. $R_e = \frac{50.6 \times 100 \times 62.27}{3.068 \times 0.95}$
 $R_e = 108,000 \text{ or } 1.08 \times 10^5$
9. $f = 0.021$ page A-25
10. $d^4 = \frac{0.0001799 \times 0.021 \times 450 \times 62.27 \times 100^2}{2}$
 $d^4 = 52.9$
11. Referring to page B-16, note d^4 is 37.161 for the 2½-inch size and 88.605 for the 3-inch size. It is evident that for this flow condition, the pressure drop through a wide-open valve is more than 2.0 psi for the 2½-inch size and less than 2.0 psi for the 3-inch size. Therefore, the 3-inch valve would not be fully lifted and a 2½-inch size should be used.
12. $\left(\frac{L}{D}\right)_a = \left(\frac{L}{D}\right)_b \left(\frac{d_a}{d_b}\right)^4 = \left(\frac{L}{D}\right)_b \left(\frac{d_a}{d_b}\right)^4$ page 3-5
13. $\left(\frac{L}{D}\right)_b = 450$ page A-30
14. $(d_a)^4 = 88.605$ page B-16
15. $(d_b)^4 = 37.161$ page B-16
16. $\left(\frac{L}{D}\right)_a = \frac{450 \times 88.605}{37.161} = 1073$
17. $\Delta P = \frac{0.0001799 \times 0.021 \times 1073 \times 62.27 \times 100^2}{88.605}$
 $\Delta P = 2.85$

Example 4-6... In-Line Ball Checks

Given: A 600-pound steel in-line ball check valve (page A-29) is required in a 2-inch pipe line carrying 48 degree API crude oil at 150 F at a rate of 37 gallons per minute.

Find: The proper size check valve and the pressure drop with the valve installed in both vertical and horizontal positions. The valve should be sized so that the disc is fully lifted under normal flow conditions; see page 2-7 for discussion.

Solution: Procedure is same as for Example 4-5.

1. $\Delta P = 2.5$ vertical position; page A-30
 $\Delta P = 0.25$ horizontal position; page A-30
2. $L/D = 150$ page A-30
3. S values for 48 degree API crude oil:
 $S = 0.788$ at 60 F; 0.75 at 150 F page A-7
4. $\rho = 62.4 \times 0.75 = 46.8$ page A-7
5. Sched 80 pipe used for calculations page A-30
6. $d = 1.939$ page B-17
7. $\mu = 1.1$ at 150 F page A-3
8. $R_e = \frac{50.6 \times 37 \times 46.8}{1.939 \times 1.1} = 41,100 \text{ or } 4.11 \times 10^4$
9. $f = 0.025$ 2" Sch. 30 pipe; page A-25
10. For valve in vertical position:
 $d^4 = \frac{0.0001799 \times 0.025 \times 150 \times 46.8 \times 37^2}{2.5}$
 $d^4 = 1.73$
11. For valve in horizontal position:
 $d^4 = \frac{0.0001799 \times 0.025 \times 150 \times 46.8 \times 37^2}{0.25}$
 $d^4 = 17.3$
12. $d^4 = 0.8387$ for 1" Sched 80 pipe; page B-16
 $d^4 = 2.6667$ for 1¼" Sched 80 pipe; page B-16
 $d^4 = 14.136$ for 2" Sched 80 pipe; page B-17
 $d^4 = 29.117$ for 2½" Sched 80 pipe; page B-17
13. As explained in the solution to Example 4-5, it is obvious a 1-inch valve in a vertical position and a 2-inch valve in a horizontal position are the largest sizes that can be used to assure "full-open" valves at specified flow conditions.
14. Pressure drop through 2-inch valve in a 2-inch pipe line:
 $\Delta P = \frac{0.0001799 \times 0.025 \times 150 \times 46.8 \times 37^2}{14.136}$
 $\Delta P = 0.306$, or say 0.31
15. Pressure drop through 1-inch valve in a 2-inch pipe line:
 $\left(\frac{L}{D}\right)_a = \frac{150 \times 14.136}{0.8387} = 2528$ page 3-5
 $\Delta P = \frac{0.0001799 \times 0.025 \times 2528 \times 46.8 \times 37^2}{14.136}$
 $\Delta P = 5.15$, or say 5.2

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 4-7

Given: Bunker C fuel oil at 90 F is flowing through a wide open 5-inch, 150-pound steel gate valve at a rate of 400 gallons per minute; the face to face dimension of the valve is 10 inches.

Find: The pressure drop through the valve.

Solution:

1. $\mu = 3000$ page A-3
2. $R_e = \frac{50.6 Q \rho}{d \mu}$ page 3-2
3. $S = 1.014$ at 60 Fpage A-7
 $S = 1.000$ at 90 F ..estimated; page A-7
4. $\rho = 62.4 \times 1.000 = 62.4$ page A-7
5. Calculations should be based on 5-inch Schedule 40 pipe; refer to page A-30.
6. $d = 5.047$ page B-16
 $D = 0.4206$ page B-16
 $d^3 = 128.56$ page B-16

7. $R_e = \frac{50.6 \times 400 \times 62.4}{5.047 \times 3000} = 83.4$
8. $R_e < 2000$ indicates the flow is laminar; therefore, use the following formula:

$$\Delta P = 0.0000228 \left(\frac{L}{D} \right) \frac{\mu Q}{d^3} \quad \text{.....page 3-4}$$

9. $\left(\frac{L}{D} \right)_v = \left(\frac{L}{D} \right)_p \frac{R_e}{1000}$ page 3-4
10. $\left(\frac{L}{D} \right)_v = 13$ page A-30

$$11. \left(\frac{L}{D} \right)_v = 13 \left(\frac{83.4}{1000} \right) = 1.08$$

Determine L , the equivalent length of the valve in feet:

$$12. L = \left(\frac{L}{D} \right)_v D = 1.08 \times 0.4206 = 0.454 \text{ feet}$$

13. The actual face to face of the valve ($L = 10 \div 12$) equals 0.833 feet. When the actual length of the valve flow path is greater than the equivalent length, the actual length should be used; refer to page 3-4.

$$14. \Delta P = 0.0000228 \left(\frac{0.833}{0.4206} \right) \left(\frac{3000 \times 400}{128.56} \right)$$

$$\Delta P = 0.421$$

Example 4-8

Given: S.A.E. 70 Lube Oil at 100 F is flowing at the rate of 600 barrels per hour through 200 feet of 8-inch Schedule 40 pipe, in which an 8-inch conventional globe valve is installed.

Find: The pressure loss through the pipe and valve.

Solution:

1. $\mu = 470$ page A-3
2. $R_e = 35.4 \frac{B \rho}{d \mu}$ page 3-2
3. $S = 0.916$ at 60 Fpage A-7
 $S = 0.90$ at 100 Fpage A-7
4. $\rho = 62.4 \times 0.9 = 56.2$ page A-7
5. $d = 7.981$ page B-16
 $D = 0.6651$ page B-16
 $d^4 = 4057.7$ page B-16

6. $R_e = \frac{35.4 \times 600 \times 56.2}{7.981 \times 470} = 318$
7. $R_e < 2000$ indicates the flow is laminar; therefore, use the following formula:

$$\Delta P = 0.000191 \frac{\mu L B}{d^4} \quad \text{.....page 3-2}$$

8. $\frac{L}{D} = 340$..for conventional globe; page A-30

$$9. \left(\frac{L}{D} \right)_v = \left(\frac{L}{D} \right)_p \frac{R_e}{1000} \quad \text{.....page 2-11 or 3-4}$$

$$\left(\frac{L}{D} \right)_v = \frac{340 \times 318}{1000} = 108$$

Then, 108 diameters of 8-inch Schedule 40 pipe is the equivalent length of globe valve at this laminar flow.

$$10. L = \left(\frac{L}{D} \right)_v D = 108 \times 0.6651 = 71.8$$

11. The total equivalent length of the valve plus the pipe equals $71.8 + 200 = 271.8$, or say 272 feet.

$$12. \Delta P = \frac{0.000191 \times 470 \times 272 \times 600}{4057.7}$$

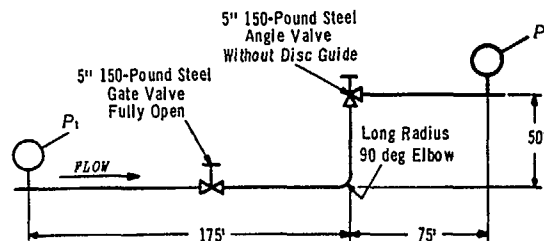
$$\Delta P = 3.6$$

Laminar Flow in Valves, Fittings, and Pipe — continued

In flow problems where viscosity is high, calculate the Reynolds Number to determine whether the flow is laminar or turbulent.

Example 4-9

Given: S.A.E. 70 Lube Oil at 100 F is flowing through 5-inch Schedule 40 pipe at a rate of 600 gallons per minute, as shown in the following sketch.



Plan view of a horizontal piping system

Find: The velocity in feet per second and pressure drop between Bourdon pressure gauges P_1 and P_2 .

Solution:

1. $v = \frac{0.408 Q}{d^2}$ page 3-2
2. $d = 5.047$ page B-16
 $D = 0.4206$ page B-16
 $d^2 = 25.47$ page B-16
 $d^4 = 648.72$ page B-16
3. $v = \frac{0.408 \times 600}{25.47} = 9.6$
4. $\mu = 480$ page A-3
5. Calculate Reynolds number to determine if flow is laminar since the viscosity is high.
6. $R_e = \frac{50.6 Q \rho}{d \mu}$ page 3-2
7. $S = 0.916$ at 60 F page A-7
 $S = 0.905$ at 100 F page A-7
8. $\rho = 62.4 \times 0.905 = 56.5$ page A-7
9. $R_e = \frac{50.6 \times 600 \times 56.5}{5.047 \times 480} = 708$
10. $R_e < 2000$ indicates the flow is laminar.
11. $\Delta P = \frac{0.000 273 \mu L Q}{d^4}$ page 3-2
12. $\frac{L}{D} = 13$ {for gate valve with turbulent flow; page A-30}
13. $\frac{L}{D} = 20$ {for 90° long radius elbow; turbulent flow; page A-30}
14. $\frac{L}{D} = 145$ {angle valve without disc guide, with turbulent flow; page A-30}
15. Sum of L/D for gate valve, long radius elbow; and angle valve, based on turbulent flow:
 $\frac{L}{D} = 13 + 20 + 145 = 178$
16. For laminar flow:
 $\left(\frac{L}{D}\right)_L = \left(\frac{L}{D}\right)_T \frac{R_e}{1000}$ page 3-4
 $\left(\frac{L}{D}\right)_L = 178 \left(\frac{708}{1000}\right) = 126$ {sum for elbow, gate, and angle}
17. $L = \left(\frac{L}{D}\right)_L D = 126 \times 0.4206$
 $L = 53.0$ feet of 5" Sched 40 pipe
18. Length of straight pipe:
 $175 + 75 + 50 = 300$ feet
19. Total equivalent length of pipe plus gate valve, long radius elbow, and angle valve:
 $300 + 53.0 = 353.0$
20. $\Delta P = \frac{0.000 273 \times 480 \times 353 \times 600}{648.72} = 42.8$

Pressure Drop and Velocity in Piping Systems

Example 4-10...Valves, Fittings, and Pipe—Steam

Given: Superheated steam, at 600 psig and 850 F, is flowing through 400 feet of horizontal 6-inch Schedule 80 pipe at the rate of 90,000 pounds per hour.

The pipe contains three long radius 90 degree elbows, one 6 x 4-inch venturi type gate valve which is fully open, and one Y-pattern globe valve having its stem 45 degrees from the run and its disc fully lifted.

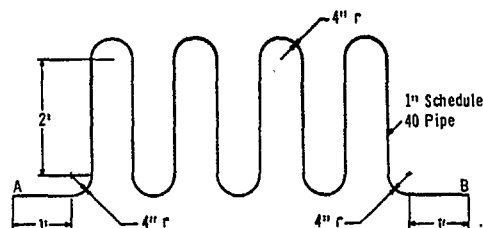
Find: The pressure drop through the piping system.

Solution:

1. $\Delta P = 0.00000336 \frac{fLW^2\bar{V}}{d^5}$ page 3-2 or 3-20
 $R_e = 6.31 \frac{W}{d\mu}$ page 3-2 or 3-18
 $L = \left(\frac{L}{D}\right) D$
2. $d = 5.761$ page B-17
 $D = 0.4801$ page B-17
 $d^5 = 6346$ page B-17
3. $\mu = 0.029$ page A-2
4. $R_e = \frac{6.31 \times 90000}{5.761 \times 0.029}$
 $R_e = 3400000$ or 3.4×10^6
5. $f = 0.015$ page A-25
6. $\frac{L}{D} = 20 \times 3 = 60$ {for 3 long radius elbows; page A-30
 $\frac{L}{D} = 145$ for Y-pattern globe; page A-30
 $\frac{L}{D} = 66.8$ {for 6 x 4" venturi type gate; Example 4-4, page 4-2
7. Sum of L/D for the three elbows and two valves:
 $\frac{L}{D} = 60 + 145 + 66.8 = 271.8$
8. Sum of L for the three elbows and two valves:
 $L = \left(\frac{L}{D}\right) D = 271.8 \times 0.4801 = 130.5$
9. Total equivalent length of pipe plus elbows and valves:
 $L = 400 + 130.5 = 530.5$, or say 531
10. $\bar{V} = 1.215$ at 614.7 psia and 850 F page A-17 or 3-17
11. $\Delta P = \frac{0.00000336 \times 0.015 \times 531 \times 90000^2 \times 1.215}{6346}$
 $\Delta P = 41.5$

Example 4-11...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.



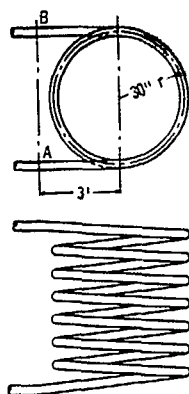
Find: The pressure drop from Point A to B.

Solution:

1. $\Delta P = 0.000216 \frac{fL\rho Q^2}{d^5}$ page 3-2
 $R_e = \frac{50.6 Q \rho}{d\mu}$ page 3-2 or 3-8
 $L = \left(\frac{L}{D}\right) D$
2. $\rho = 60.57$ page A-6
3. $d = 1.049$ $d^5 = 1.270$ page B-16
 $D = 0.0874$ page B-16
4. $\mu = 0.34$ page A-3
5. $R_e = \frac{50.6 \times 15 \times 60.57}{1.049 \times 0.34}$
 $R_e = 128900$ or 1.289×10^5
6. $f = 0.024$ page A-25
7. Length, straight pipe (see sketch) = 18 feet
8. $r/d = 4$
9. Total resistance for one 90 degree bend:
 $L/D = 13.5$ page A-27
10. The total resistance for one 180 degree bend is the resistance for the 90 degree bend ($L/D = 13.5$), plus the resistance due to length ($L/D = 7.5$), plus one-half the resistance due to bend ($L/D = 6 \div 2 = 3.0$), or a total of 24.0.
11. The total bend loss, L/D , is:
Two 90 degree bends. $\dots 2 \times 13.5 = 27.0$
Seven 180 degree bends. $\dots 7 \times 24.0 = 168.0$
 L/D for total bends = 195.0
12. Total bend equivalent length in straight pipe:
 $L = \left(\frac{L}{D}\right) D = 195 \times 0.0874 = 17.04$
13. Total equivalent length of pipe is:
 $L = 18 + 17.04 = 35.04$, or say 35.0
14. $\Delta P = \frac{0.000216 \times 0.024 \times 35.0 \times 60.57 \times 15^2}{1.27}$
 $\Delta P = 1.95$

Pressure Drop and Velocity in Piping Systems — continued

Example 4-12...Cylindrical Pipe Coil—Steam



Given: Saturated steam at 150 psig is flowing through a cylindrical pipe coil, as shown in the sketch at the left, at a rate of 2000 pounds per hour. The coil is made of 2-inch Schedule 40 steel pipe.

Find: The pressure drop from Point A to B.

Solution:

$$1. \quad \Delta P = 0.00000336 \frac{f L W^2 \bar{V}}{d^5} \quad \dots \text{page 3-2 or 3-20}$$

$$R_s = \frac{6.31 W}{d \mu} \quad \dots \text{page 3-2 or 3-18}$$

$$L = \left(\frac{L}{D} \right) D$$

$$2. \quad d = 2.067 \quad d^5 = 37.72 \quad \dots \text{page B-16}$$

$$D = 0.1722 \quad \dots \text{page B-16}$$

$$3. \quad \mu = 0.018 \quad \dots \text{page A-2}$$

$$4. \quad R_s = \frac{6.31 \times 2000}{2.067 \times 0.018} = 339000 \text{ or } 3.39 \times 10^5$$

$$5. \quad f = 0.020 \quad \dots \text{page A-25}$$

$$6. \quad \text{Length of inlet and outlet pipe with slight bend (see sketch) equals 6.0 feet.}$$

$$7. \quad r/d = 15$$

$$8. \quad \text{Equivalent length of coiled pipe (page 2-12):}$$

$$\frac{L}{D} = 41 \quad \dots \left\{ \text{for one 90 degree bend; page A-27} \right.$$

$$\frac{L}{D} = 23.5 \quad \dots \left\{ \text{for resistance due to length of one 90 degree bend; page A-27} \right.$$

$$\frac{L}{D} = \frac{17.5}{2} = 8.75 \quad \left\{ \text{resistance due to bend of one 90 degree bend; page A-27} \right.$$

There are 6.5 coils or twenty-six 90 degree bends in the coil. Then, the equivalent length of the coil is:

$$\frac{L}{D} = (26-1)(23.5 + 8.75) + 41 = 847$$

$$L = \left(\frac{L}{D} \right) D = 847 \times 0.1722 = 146$$

$$9. \quad \text{The total equivalent length of the coil is: } 6.0 + 146 = 152 \text{ feet}$$

$$10. \quad \bar{V} = 2.757 \quad \dots \text{page A-14 or 3-17}$$

$$11. \quad \Delta P = \frac{0.00000336 \times 0.020 \times 152 \times 2000^2 \times 2.757}{37.72}$$

$$\Delta P = 3.0$$

Example 4-13...Flow Given in Metric Units—Oil

Given: Fuel oil, with a specific gravity of 0.815 and a kinematic viscosity of 0.027 cm²/sec, is flowing through a 50 millimeter I.D. steel pipe of 30 meters length at a rate of 7.0 liters per second.

Find: The pressure drop in pounds per square inch and in kilograms per square centimeter.

Solution:

$$1. \quad \Delta P = 0.000216 \frac{f L \rho Q^2}{d^5} \quad \dots \text{page 3-2 or 3-10}$$

$$R_s = 3160 \frac{Q}{\nu d} \quad \dots \text{page 3-2}$$

$$2. \quad \text{Convert units given to those used in this paper; refer to page B-10.}$$

$$1 \text{ mm} = 0.00328 \text{ ft}$$

$$1 \text{ m} = 3.28 \text{ ft}$$

$$1 \text{ mm} = 0.03937 \text{ in.}$$

$$3. \quad L = 30.0 \text{ m} = 30.0 \times 3.28 = 98.4 \text{ ft}$$

$$4. \quad d = 50 \times 0.03937 = 1.97 \text{ in.}$$

$$5. \quad \rho = 62.4 S = 62.4 \times 0.815 \quad \dots \text{page A-7}$$

$$\rho = 50.9 \text{ lb/ft}^3$$

$$6. \quad 7.0 \text{ liters} = 7.0 \times 0.2642 = 1.849 \text{ gal} \quad \dots \text{page B-11}$$

$$7. \quad Q = \left(\frac{1.849 \text{ gal}}{\text{sec}} \right) \left(\frac{60 \text{ sec}}{\text{min}} \right) = 110.9 \text{ gpm}$$

$$8. \quad \nu = 0.027 \times 100 = 2.7 \text{ centistokes.} \quad \text{pg. B-3}$$

$$9. \quad R_s = \frac{3160 \times 110.9}{2.7 \times 1.97} = 65800 \text{ or } 6.58 \times 10^4$$

$$10. \quad f = 0.0230 \quad \dots \text{page A-25}$$

$$11. \quad d^5 = 1.97^5 = 29.6$$

$$12. \quad \Delta P = \frac{0.000216 \times 0.0230 \times 98.4 \times 50.9 \times 110.9^2}{29.6}$$

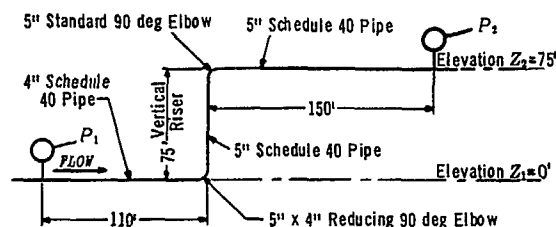
$$\Delta P = 10.32$$

$$13. \quad \text{Pressure drop} = 10.32 \times 0.0703 = 0.725 \text{ kg/cm}^2$$

Pressure Drop and Velocity in Piping Systems — continued

Example 4-14...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-inch pipe sizes and the pressure differential between gauges P_1 and P_2 .

Solution:

1. Use Bernoulli's theorem (see page 3-2):

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

Since, $P_1 = P_2$

$$P_1 - P_2 = \frac{\rho}{144} \left((Z_2 - Z_1) + \frac{v_2^2 - v_1^2}{2g} + h_L \right)$$

$$h_L = \frac{144 \Delta P}{\rho} \quad \dots \text{page 3-5}$$

ΔP = pressure drop due to flow

$$L = \left(\frac{L}{D} \right) D$$

2. $\rho = 62.34$ page A-6

3. $Z_2 - Z_1 = 75 - 0 = 75$

4. $v_1 = 10.08$ in 4" pipe; page B-14

5. $v_2 = 6.42$ in 5" pipe; page B-14

6. $\frac{v_2^2 - v_1^2}{2g} = \frac{6.42^2 - 10.08^2}{2 \times 32.2}$
 $= -0.938$ or say -0.9 feet

7. Pressure drop through 100 feet of 4-inch Schedule 40 pipe is:

$$\Delta P = 3.68 \quad \dots \text{page B-14}$$

8. Then the pressure drop through 110 feet of 4-inch Schedule 40 pipe is:

$$\Delta P = \frac{110}{100} \times 3.68 = 4.05 \quad \dots \text{page B-14}$$

9. $\frac{L}{D} = 30$ (for 90 degree standard elbow; page A-30)

10. $D = 0.3355$..for 4" Sched 40 pipe; page B-16

- $D = 0.4206$..for 5" Sched 40 pipe; page B-16

11. $L = \left(\frac{L}{D} \right) D = 30 \times 0.3355 = 10.07$
 (equivalent length in feet of 4" Schedule 40 pipe ... for 4" elbow)

$$L = \left(\frac{L}{D} \right) D = 30 \times 0.4206 = 12.62$$

(equivalent length in feet of 5" Schedule 40 pipe ... for 5" elbow)

12. For a 4-inch standard 90 degree elbow,
 $\Delta P = (10.07 + 100) 3.68 = 0.371$..page B-14

13. For 100 feet of 5-inch Schedule 40 pipe,
 $\Delta P = 1.19$ page B-14

14. For a 5-inch standard 90 degree elbow,
 $\Delta P = (12.62 + 100) 1.19 = 0.150$..page B-14

15. For a 5 x 4-inch 90 degree reducing elbow,
 $\Delta P = \frac{0.371 + 0.150}{2} = 0.261$, or say 0.26

16. For the 5-inch Schedule 40 pipe,
 $\Delta P = \left(\frac{150 + 75}{100} \right) 1.19 = 2.68$..page B-14

17. The pressure drop in the pipe between the gauges, due to fluid flow, is:

$$\Delta P = 4.05 \quad \dots 110 \text{ feet 4" Sched 40 pipe}$$

$$\Delta P = 0.26 \quad \dots 5 \times 4" 90^\circ \text{ reducing elbow}$$

$$\Delta P = 0.15 \quad \dots 5" \text{ standard } 90^\circ \text{ elbow}$$

$$\Delta P = 2.68 \quad \dots 225 \text{ feet 5" Sched 40 pipe}$$

$$\Delta P = 7.14 \text{ (total)}$$

18. $h_L = \frac{144 \Delta P}{\rho} = \frac{144 \times 7.14}{62.34} = 16.5$

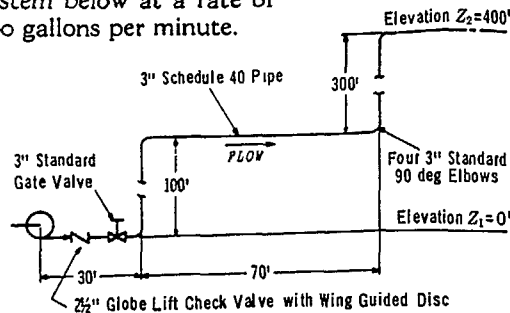
19. $P_1 - P_2 = \left(\frac{62.34}{144} \right) (75.0 - 0.9 + 16.5)$

$$P_1 - P_2 = 39.2$$

Pressure Drop and Velocity in Piping Systems — continued

Example 4-15... 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 at flowing conditions and the brake horsepower required for a pump having an efficiency of 70 per cent.

Solution: 1. Use Bernoulli's theorem (see page 3-2):

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

2. Since $\rho_1 = \rho_2$ and $v_1 = v_2$, the equation can be rewritten to establish the pump head, H :

$$\frac{144}{\rho} (P_1 - P_2) = (Z_2 - Z_1) + h_L$$

3. $h_L = 0.1863 \frac{f L v^2}{d}$ page 3-2

$R_e = 123.9 \frac{d v \rho}{\mu}$ page 3-2 or 3-8

$v = \frac{0.408 Q}{d^2}$ page 3-2 or 3-6

$K = f \frac{L}{D}$, then $\frac{L}{D} = \frac{K}{f}$ page 3-4

$$L = (L/D) D$$

brake horsepower = $\frac{Q H \rho}{247,000 e_p}$.. page B-9

4. $d = 3.068$ $d^2 = 9.413$ page B-16
 $D = 0.2557$ page B-16

5. $v = \frac{0.408 \times 100}{9.413} = 4.33$

6. $\rho = 62.27$ page A-6

7. $\mu = 0.95$ page A-3

8. $R_e = \frac{123.9 \times 3.068 \times 4.33 \times 62.27}{0.95}$

$$R_e = 107,900 \text{ or } 1.079 \times 10^5$$

9. $f = 0.021$ page A-25 or 3-8

10. $\frac{L}{D} = 30 \times 4 = 120$ {for four std. 90° elbows; page A-30}

11. $\frac{L}{D} = 1073$ {for one 2 1/2" globe lift check with wing-guided disc; Prob. 4-5, Step 16}

12. $K = 1.0$ for exit loss; page A-26

13. $\frac{L}{D} = \frac{K}{f} = \frac{1.000}{0.021} = 47.6$ for exit loss

14. For elbows, valve, and exit loss,

$$\frac{L}{D} = 120 + 1073 + 47.6 = 1240.6 \text{ or say } 1241$$

$$L = \left(\frac{L}{D}\right) D = 1241 \times 0.2557 = 317$$

15. The total equivalent length of the pipe is:

$$L = 30 + 100 + 70 + 300 + 317 = 817$$

16. The head loss due to the flow through elbows, valve, and pipe, and the exit loss is:

$$h_L = \frac{0.1863 \times 0.021 \times 817 \times 4.33^2}{3.068} = 19.53$$

17. The total pump head is:

$$400 - 0 + 19.53 = 419.53 \text{ or say } 420 \text{ ft}$$

18. Brake horsepower = $\frac{100 \times 420 \times 62.27}{247,000 \times 0.70} = 15.1$

Example 4-16... Air Lines

Given: Air at 65 psig and 110 F is flowing through 75 feet of 1-inch 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 upstream and downstream gauges.

Solution: 1. Referring to the table on page B-15, read pressure drop of 2.21 psi for 100 psi, 60 F air at a flow rate of 100 scfm through 75 feet of 1-inch Schedule 40 pipe.

2. Correction for length, pressure, and temperature (page B-15):

$$\Delta P = 2.21 \left(\frac{75}{100}\right) \left(\frac{100 + 14.7}{65 + 14.7}\right) \left(\frac{460 + 110}{520}\right)$$

$$\Delta P = 2.61$$

3. To find the velocity, the rate of flow in cubic feet per minute at flowing conditions must be determined from page B-15.

$$q_m = q'_m \left(\frac{14.7}{14.7 + P}\right) \left(\frac{460 + t}{520}\right)$$

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. $V = \frac{q_m}{A}$ page 3-2

5. $A = 0.006$ page B-16

6. $V = \frac{20.2}{0.006} = 3367$ at upstream gauge

$V = \frac{20.9}{0.006} = 3483$ at downstream gauge

Note: Example 4-16 may also be solved by use of the pressure drop formula and nomograph shown on pages 3-2 and 3-21 respectively or the velocity formula and nomograph shown on pages 3-2 and 3-17 respectively.

Pipe Line Flow Problems

Example 4-17...Sizing of Pump for Oil Pipe Lines

Given: Crude oil 30 degree API at 15.6 C with a viscosity of 75 Universal Saybolt seconds is flowing through a 12-inch Schedule 30 steel pipe at a rate of 1900 barrels per hour. The pipe line is 50 miles long with discharge at an elevation of 2000 feet above the pump inlet. Assume the pump has an efficiency of 67 per cent.

Find: The brake horsepower of the pump.

Solution:

1. $\Delta P = 0.0001058 \frac{f L \rho B^2}{d^5}$ {Equation 3-5 on page 3-2
or, after converting B to Q ,
use nomograph on page 3-11}

$t = 1.8 t_c + 32$ page B-10
 $R_e = 35.4 \frac{B \rho}{d \mu}$ page 3-2 or 3-8
 $h_L = \frac{144 \Delta P}{\rho}$ page 3-5
 $\text{brake horsepower} = \frac{QH_p}{247\,000 e_p}$ page B-9
2. $t = (1.8 \times 15.6) + 32 = 60 \text{ F}$
3. $\rho = 54.64$ page B-7
 $S = 0.8762$ page B-7
4. $d = 12.09$ page B-16
 $d^5 = 258\,304$ page B-16
5. 75 USS = 12.5 centipoise page B-5
6. $R_e = \frac{35.4 \times 1900 \times 54.64}{12.09 \times 12.5} = 24\,300$
7. $f = 0.025$ page A-25
8. $\Delta P = \frac{0.0001058 \times 0.025 \times 50 \times 5280 \times 54.64 \times 1900^2}{258\,304}$
 $\Delta P = 533$
9. $h_L = \frac{144 \times 533}{54.64} = 1405$
10. The total discharge head at the pump is:
 $H = 1405 + 2000 = 3405$
11. $Q = \left(\frac{1900 \text{ bbl}}{\text{hr}} \right) \left(\frac{42 \text{ gal}}{\text{bbl}} \right) \left(\frac{1 \text{ hr}}{60 \text{ min}} \right) = 1330$
12. Then, the brake horsepower is
 $\frac{1330 \times 3405 \times 54.64}{247\,000 \times 0.67} = 1496 \text{ or say } 1500$

Pipe Line Flow Problems — continued

Example 4-18... Gas

Given: A natural gas pipe line, made of 14-inch 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 Compressible Flow formula, the Weymouth formula, and the Panhandle formula.

Simplified Compressible Flow Formula
(see page 3-3)

1. $q'_h = 114.2 \sqrt{\left[\frac{(P'_1)^2 - (P'_2)^2}{f L_m T S_g} \right]} d^5$
2. $d = 13.376$ page B-16
 $d^5 = 428\ 185$ page B-16
3. $f = 0.0128$ turbulent flow assumed; page A-25
4. $T = 460 + t = 460 + 40 = 500$
5. Approximate atomic weights:
Carbon..... C = 12.0
Hydrogen.... H = 1.0
6. Approximate molecular weights:
Methane (CH_4)
 $M = (1 \times 12.0) + (4 \times 1.0) = 16$
Ethane (C_2H_6)
 $M = (2 \times 12.0) + (6 \times 1.0) = 30$
Propane (C_3H_8)
 $M = (3 \times 12.0) + (8 \times 1.0) = 44$
Natural Gas
 $M = (16 \times 0.75) + (30 \times 0.21) + (44 \times 0.04)$
 $M = 20.06$, or say 20.1
7. $S_g = \frac{M(\text{gas})}{M(\text{air})} = \frac{20.1}{29} = 0.693$... page 3-5
8. $q'_h = 114.2 \sqrt{\frac{(1300^2 - 300^2) 428\ 185}{0.0128 \times 100 \times 500 \times 0.693}}$
 $q'_h = 4\ 490\ 000$

9. $q'_d = \left(\frac{4\ 490\ 000\ \text{ft}^3}{1\ 000\ 000\ \text{hr}} \right) \left(\frac{24\ \text{hr}}{\text{day}} \right) = 107.8$
10. $R_e = \frac{0.482 q'_h S_g}{d \mu}$ page 3-2
11. $\mu = 0.011$ estimated; page A-5
12. $R_e = \frac{0.482 \times 4\ 490\ 000 \times 0.693}{13.376 \times 0.011}$
 $R_e = 10\ 190\ 000$ or 1.019×10^7
13. $f = 0.0128$ page A-25
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 Formula
(see page 3-3)

15. $q'_h = 28.0 d^{2.667} \sqrt{\left[\frac{(P'_1)^2 - (P'_2)^2}{S_g L_m} \right]} \left(\frac{520}{T} \right)$
16. $d^{2.667} = 1009$ (use a slide rule or logarithms, page B-12)
17. $q'_h = 28.0 \times 1009 \sqrt{\frac{(1300^2 - 300^2)}{0.693 \times 100}} \left(\frac{520}{500} \right)$
 $q'_h = 4\ 380\ 000$
18. $q'_d = \left(\frac{4\ 380\ 000\ \text{ft}^3}{1\ 000\ 000\ \text{hr}} \right) \left(\frac{24\ \text{hr}}{\text{day}} \right) = 105.1$

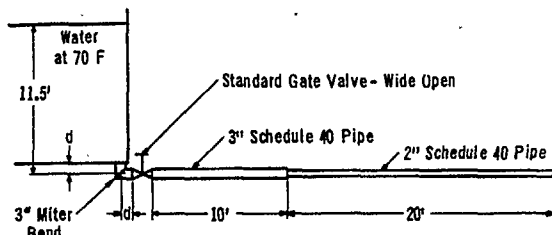
Panhandle Formula
(see page 3-3)

19. $q'_h = 36.8 E d^{2.6182} \left[\frac{(P'_1)^2 - (P'_2)^2}{L_m} \right]^{0.5394}$
20. Assume average operation conditions; then efficiency is 92 per cent:
 $E = 0.92$
21. $d^{2.6182} = 889$ (use a slide rule or logarithms, page B-12)
22. $q'_h = 36.8 \times 0.92 \times 889 \left(\frac{1300^2 - 300^2}{100} \right)^{0.5394}$
 $q'_h = 5\ 570\ 000$
23. $q'_d = \left(\frac{5\ 570\ 000\ \text{ft}^3}{1\ 000\ 000\ \text{hr}} \right) \left(\frac{24\ \text{hr}}{\text{day}} \right) = 133.7$

Discharge of Fluids from Piping Systems

Example 4-19...Water

Given: Water is flowing from a reservoir through the piping system sketched below. The reservoir has a constant head of 11.5 feet.



Find: The flow rate in gallons per minute.

Solution:

$$1. \quad Q = 19.65 d^2 \sqrt{\frac{h_L}{K}} \quad \text{page 3-4}$$

$$K_a = K_b \left(\frac{d_a}{d_b} \right)^4 = K_b \frac{(d_a)^4}{(d_b)^4} \quad \text{page 3-5}$$

$$2. \quad d^2 = 9.413 \quad \text{3" pipe; page B-16}$$

$$d^4 = 88.605 \quad \text{3" pipe; page B-16}$$

$$3. \quad d^4 = 18.25 \quad \text{2" pipe; page B-16}$$

4. Tabulation of resistances, assuming fully turbulent flow, follows. Subscript "a" refers to the 3-inch size and subscript "b" refers to the 2-inch size.

	K_a	$\left(\frac{L}{D} \right)_a$	K_b	$\left(\frac{L}{D} \right)_b$	Reference
Entrance loss	0.5	...	...	...	page A-26
Miter bend	1.0	58	...	...	page A-27 page A-31
Gate valve	0.23	13	...	...	page A-30 page A-31
10 feet of 3-inch pipe	0.7	39.1	...	...	10 ÷ 0.2557 page A-31
Sudden contraction	...	...	0.23	...	page A-26
Exit loss	...	...	1.0	...	page A-26
20 feet of 2-inch pipe	...	...	2.2	...	20 ÷ 0.1722 page A-31
Totals	2.43		3.43		

$$5. \quad K_a = 3.43 \left(\frac{88.605}{18.25} \right) = 16.65$$

$$\text{Total } K_a = 16.65 + 2.43 = 19.08$$

$$6. \quad Q = 19.65 (9.413) \sqrt{\frac{11.5}{19.08}} = 144$$

This solution assumes flow in the fully turbulent zone per page A-25, from which, $f = 0.0175$ for 3-inch pipe and 0.019 for 2-inch pipe.

7. Check assumed friction factors used in determining resistance of 2 and 3-inch pipe.

$$8. \quad \rho = 62.27 \quad \text{page A-6}$$

$$9. \quad \mu = 0.95 \quad \text{page A-3}$$

$$10. \quad R_e = 150\,000 \text{ or } 1.5 \times 10^5 \text{ for 3" pipe; page 3-9}$$

$$R_e = 230\,000 \text{ or } 2.3 \times 10^5 \text{ for 2" pipe; page 3-9}$$

$$f = 0.020 \quad \text{for 3" pipe; page A-25}$$

$$f = 0.0206 \quad \text{for 2" pipe; page A-25}$$

11. Since assumed friction factors (Step 6) are not in agreement with those based on approximate flow (Step 10), total K values in Step 4 must be corrected by multiplying the values by the ratio of f based on approximate flow to f assumed.

$$K_a = 2.43 \left(\frac{0.020}{0.0175} \right) = 2.78$$

$$K_b = 3.43 \left(\frac{0.0206}{0.019} \right) = 3.72$$

12. Repeat Steps 5 and 6, using the corrected values of K .

$$K_a = 3.72 \left(\frac{88.605}{18.25} \right) = 18.08$$

$$\text{Total } K_a = 2.78 + 18.08 = 20.86$$

$$Q = 19.65 (9.413) \sqrt{\frac{11.5}{20.86}} = 137$$

13. Further determination of Reynolds number and friction factor based on the corrected flow rate obtained in Step 12 is not necessary, since the effect on friction factor would be negligible.

Discharge of Fluids from Piping Systems — continued

Example 4-20...Steam at Sonic Velocity

Given: A header with 170 psia saturated steam is feeding a pulp stock digester through 30 feet of 2-inch 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 formula and the sonic velocity and continuity equations.

Solutions—for theory, see page 1-9:

Modified Darcy Formula

1. $W = 1891 Y d^2 \sqrt{\frac{\Delta P}{K V_1}}$ page 3-4
- $K = f \frac{L}{D}$ page 3-4
2. $k = 1.3$ page A-9
3. $D = 0.1722 \quad d^2 = 4.272$ page B-16
4. $\frac{L}{D} = \frac{30}{0.1722} = 174.2$ for pipe
- $\frac{L}{D} = 340$ for 2" globe valve; page A-30
- $\frac{L}{D} = 30$ for 2" elbow; page A-30
5. $f = 0.019$ {for fully turbulent flow; page A-25}
6. Sum of K for pipe, valve, and elbow:
 $K = 0.019(174.2 + 340 + 30) = 10.34$
7. $K = 0.5$ for entrance; page A-26
 $K = 1.0$ for exit; page A-26
8. Then, for the total system,
 $K = 10.34 + 0.5 + 1.0 = 11.84$
9. $\frac{\Delta P}{P'_1} = \frac{170 - 14.7}{170} = \frac{155.3}{170} = 0.914$
10. Using the chart on page A-22 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-22). Since $\Delta P/P'_1$ is less than indicated in Step 9, sonic velocity occurs at the end of the pipe, and ΔP in the equation of Step 1 is:
 $\Delta P = 0.785 \times 170 = 133.5$
11. $\bar{V}_1 = 2.675$ page A-14
12. $Y = 0.710$ {interpolated from table; page A-22}
13. $W = 1891 \times 0.71 \times 4.272 \sqrt{\frac{133.5}{11.84 \times 2.675}}$
 $W = 11780$

Sonic Velocity and Continuity Equations

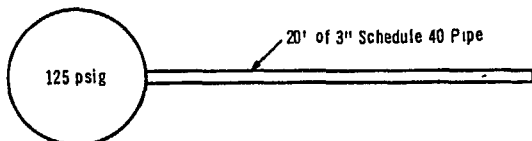
14. $v_s = \sqrt{kg \ 144 \ P' \bar{V}}$ *ft/sec* page 3-3
 $W = \frac{v \ d^2}{0.0509 \ \bar{V}}$ Equation 3-2; page 3-2
15. $P' = P'_1 - \Delta P$
 $P' = 170 - 133.5 = 36.5$
 ΔP determined in Step 10.
16. $h_g = 1196$... 170 psia saturated steam; page A-14
17. At 36.5 psia, the temperature of steam with total heat of 1196 Btu/lb equals 316 F, and
 $\bar{V} = 12.4$ pages A-13 and A-16
18. $v_s = \sqrt{1.3 \times 32.2 \times 144 \times 36.5 \times 12.4}$
 $v_s = 1652$
 $W = \frac{1652 \times 4.272}{0.0509 \times 12.4} = 11180$

$\frac{WV}{Y_2} = .0509$

Discharge of Fluids from Piping Systems — continued

Example 4-21...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-inch 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:

$$1. \quad q'_h = 40\,700 \, Y d^2 \sqrt{\frac{\Delta P P'_1}{K T_1 S_g}} \quad \dots \text{page 3-4}$$

$$K = f \frac{L}{D} \quad \dots \text{page 3-4}$$

$$2. \quad P'_1 = 125 + 14.7 = 139.7$$

$$3. \quad f = 0.0175 \quad \dots \text{page A-25}$$

Note: The Reynolds number need not be calculated since gas discharged to atmosphere through a short pipe will have a high Re , and flow will always be in a fully turbulent range, in which the friction factor is constant.

$$4. \quad D = 0.2557 \quad d^2 = 9.413 \quad \dots \text{page B-16}$$

$$5. \quad K = f \frac{L}{D} = \frac{0.0175 \times 20}{0.2557} = 1.369 \quad \dots \text{for pipe}$$

$$K = 0.5 \quad \dots \text{for entrance; page A-26}$$

$$K = 1.0 \quad \dots \text{for exit; page A-26}$$

$$K = 1.369 + 0.5 + 1.0 = 2.87 \quad \dots \text{total}$$

$$6. \quad \frac{\Delta P}{P'_1} = \frac{139.7 - 14.7}{139.7} = \frac{125.0}{139.7} = 0.895$$

7. Using the chart on page A-22 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-22). 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:

$$\Delta P = 0.657 P'_1 = 0.657 \times 139.7 = 91.8$$

$$8. \quad T_1 = 140 + 460 = 600$$

$$9. \quad Y = 0.637 \quad \dots \left\{ \begin{array}{l} \text{interpolated from} \\ \text{table; page A-22} \end{array} \right.$$

10. q'_h is equal to:

$$40\,700 \times 0.637 \times 9.413 \sqrt{\frac{91.8 \times 139.7}{2.87 \times 600 \times 0.42}}$$

$$q'_h = 1\,028\,000$$

Example 4-22...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 feet from the outlet of a 1/2-inch Schedule 80 pipe discharging to atmosphere.

Find: The flow rate in standard cubic feet per minute (scfm).

Solution:

$$1. \quad q'_m = 678 \, Y d^2 \sqrt{\frac{\Delta P P'_1}{K T_1 S_g}} \quad \dots \text{page 3-4}$$

$$K = f \frac{L}{D} \quad \dots \text{page 3-4}$$

$$2. \quad P'_1 = 19.3 + 14.7 = 34.0$$

$$3. \quad \Delta P = 19.3$$

$$4. \quad D = 0.0455 \quad d^2 = 0.2981 \quad \dots \text{page B-16}$$

$$5. \quad f = 0.0275 \quad \dots \text{fully turbulent flow; page A-25}$$

$$6. \quad K = f \frac{L}{D} = \frac{0.0275 \times 10}{0.0455} = 6.04 \quad \dots \text{for pipe}$$

$$K = 1.0 \quad \dots \text{for exit; page A-26}$$

$$K = 6.04 + 1 = 7.04 \quad \dots \text{total}$$

$$7. \quad \frac{\Delta P}{P'_1} = \frac{19.3}{34.0} = 0.568$$

$$8. \quad Y = 0.76 \quad \dots \text{page A-22}$$

$$9. \quad T_1 = 460 + t_1 = 460 + 100 = 560$$

$$10. \quad q'_m = 678 \times 0.76 \times 0.2981 \sqrt{\frac{19.3 \times 34.0}{7.04 \times 560 \times 1.0}}$$

$$q'_m = 62.7$$

Flow Through Orifice Meters

Example 4-23...Liquid Service

Given: A square edged orifice of 2.0-inch diameter is installed in a 4-inch Schedule 40 pipe having a mercury manometer connected between the flange taps.

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 inches.

Solution — (a)

$$1. \quad Q = 236 d^2_0 C \sqrt{\frac{\Delta P}{\rho}} \quad \dots \text{page 3-5 or 3-15}$$

$$R_e = \frac{50.6 Q \rho}{d \mu} \quad \dots \text{page 3-2 or 3-8}$$

2. To determine differential pressure across the flange taps,

$$\Delta P = \frac{\Delta h_m \rho}{12 \times 144} \quad \dots \text{page 3-5}$$

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 = 62.34 page A-6

S_{Hg} = specific gravity of mercury = 13.57 page A-7

S_w = specific gravity of water = 1.00 page A-6

4. And ρ of H_g under H_2O = 62.34 (13.57 - 1.00)
= 784 lb/ft³

$$5. \quad \Delta P = \frac{\Delta h_m (784)}{12 \times 144} = 0.454 \Delta h_m$$

6. $d_1 = 4.026$ page B-16

$$7. \quad \frac{d_0}{d_1} = \frac{2.00}{4.026} = 0.497$$

8. $C = 0.625$ page A-19

$$9. \quad Q = 236 \times (2.0)^2 \times 0.625 \sqrt{\frac{0.454 \Delta h_m}{62.34}}$$

$$Q = 50.4 \sqrt{\Delta h_m} \quad \dots \text{calibration constant}$$

Solution — (b):

$$10. \quad Q = 50.4 \sqrt{\Delta h_m} = 50.4 \sqrt{4.4} = 106$$

11. $\mu = 1.1$ page A-3

$$12. \quad R_e = \frac{50.6 \times 106 \times 62.34}{4.026 \times 1.1}$$

$$R_e = 75,500 \text{ or } 7.55 \times 10^4$$

13. $C = 0.625$ is correct for $R_e = 7.55 \times 10^4$, per page A-19; therefore, the flow rate through the pipe is 106 gallons per minute.

14. When the C factor on page A-19 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 4-24...Laminar Flow

In flow problems where the viscosity is high, calculate the Reynolds number to determine the type of flow.

Given: SAE 10 Lube Oil at 90 F is flowing through a 3-inch Schedule 40 pipe and produces 0.4 psi pressure differential between the flange taps of a 2.15-inch I. D. square edged orifice.

Find: The flow rate in gallons per minute.

Solution:

$$1. \quad Q = 236 d^2_0 C \sqrt{\frac{\Delta P}{\rho}} \quad \dots \text{page 3-5 or 3-15}$$

$$R_e = \frac{50.6 Q \rho}{d \mu} \quad \dots \text{page 3-2 or 3-8}$$

2. $\mu = 38$ suspect laminar flow; page A-3

3. $d_1 = 3.068$ page B-16

$$4. \quad \frac{d_0}{d_1} = \frac{2.15}{3.068} = 0.70$$

5. $C = 0.8$ {page A-19; assumed value based on laminar flow

6. $S = 0.876$ at 60 F page A-7

$S = 0.87$ at 90 F page A-7

7. $\rho = 62.4 \times 0.87 = 54.3$ page A-7

$$8. \quad Q = 236 \times 2.15^2 \times 0.8 \sqrt{\frac{0.4}{54.3}} = 75$$

$$9. \quad R_e = \frac{50.6 \times 75 \times 54.3}{3.068 \times 38} = 1768$$

10. $C = 0.92$ for $R_e = 1768$ page A-19

Since the assumed C value of 0.8 is not correct, it must be adjusted by repeating Steps 5, 8, 9, and 10.

11. $C = 0.91$ assumed; page A-19

$$12. \quad Q = 236 \times 2.15^2 \times 0.91 \sqrt{\frac{0.4}{54.3}} = 85.3$$

$$13. \quad R_e = \frac{50.6 \times 85.3 \times 54.3}{3.068 \times 38} = 2010$$

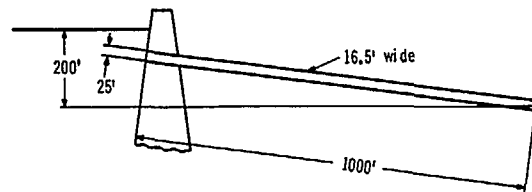
14. $C = 0.91$ for $R_e = 2010$ page A-19

Since $C = 0.91$ is correct for the flow, the flow through the meter is 85.3 gallons per minute.

Application of Hydraulic Radius to Flow Problems

Example 4-25...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:

$$1. \quad q = 0.0438 d^2 \sqrt{\frac{h_L}{K_e + K_a}} = \sqrt{\frac{h_L}{K_e + f \frac{L}{D}}} \quad \text{.....page 3-4}$$

where;

K_e = resistance of entrance and exit

K_a = resistance of aqueduct

To determine the flow rate, calculate the equivalent diameters for actual flow area.

$$a = \frac{\pi d^2}{4}, \text{ therefore, } d^2 = \frac{4a}{\pi}$$

$$A = \frac{\pi D^2}{4}, \text{ therefore, } D = \sqrt{\frac{4A}{\pi}}$$

To determine the friction factor from the Moody diagram, an equivalent diameter four times the hydraulic radius is used; refer to page 3-5.

$$\text{hydraulic radius} = \frac{\text{cross sectional flow area}}{\text{wetted perimeter}}$$

$$R_e = 22\,700 \frac{q\rho}{d\mu} \quad \text{.....page 3-2}$$

$$2. \quad d^2 = \frac{4a}{\pi} = \frac{4 \times 25 \times 16.5 \times 144}{\pi} = 75\,600$$

$$3. \quad D = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4 \times 25 \times 16.5}{\pi}} = 22.9$$

$$4. \quad \text{Assuming a sharp edged entrance,} \\ K = 0.5 \quad \text{.....page A-26}$$

$$\text{Assuming a sharp edged exit to atmosphere,} \\ K = 1.0 \quad \text{.....page A-26}$$

Then, resistance of entrance and exit,

$$K_e = 1.0 + 0.5 = 1.5$$

$$5. \quad \text{Hydraulic radius} = \frac{16.5 \times 25}{2(16.5 + 25)} = 4.97 \text{ ft}$$

$$6. \quad \text{Equivalent diameter, } d = 4 \times 4.97 \times 12 \\ d = 239 \text{ in.}$$

$$7. \quad \text{Relative roughness, } \epsilon/D = 0.0005 \quad \text{..page A-23}$$

$$8. \quad f = 0.017 \quad \text{.....fully turbulent flow assumed; page A-23}$$

$$9. \quad q = 0.0438 \times 75\,600 \sqrt{\frac{200}{1.5 + \frac{0.017 \times 1000}{22.9}}}$$

$$q = 31\,300$$

$$10. \quad \text{Calculate } R_e \\ \text{and check, } f = 0.017 \text{ for } 31\,300 \text{ cfs flow.}$$

$$11. \quad \rho = 62.34 \quad \text{.....page A-6}$$

$$12. \quad \mu = 1.1 \quad \text{.....page A-3}$$

$$13. \quad R_e = \frac{22\,700 \times 31\,300 \times 62.34}{239 \times 1.1}$$

$$R_e = 168\,400\,000 \text{ or } 1.684 \times 10^8$$

$$14. \quad f = 0.017 \quad \text{.....for calculated } R_e; \text{ page A-24}$$

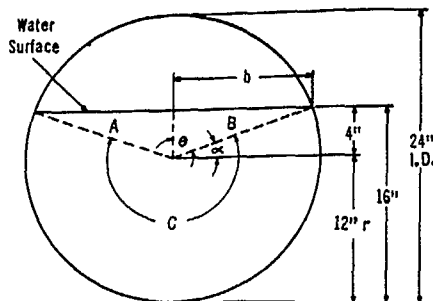
$$15. \quad \text{Since the friction factor assumed in Step 8 and that determined in Step 14 are in agreement, the discharge flow will be } 31\,300 \text{ cfs.}$$

$$16. \quad \text{If the assumed friction factor and the friction factor based on the calculated Reynolds number were not in reasonable agreement, the former should be adjusted and calculations repeated until reasonable agreement is reached.}$$

Application of Hydraulic Radius to Flow Problems — continued

Example 4-26...Pipe Partially Filled With Flowing Water

Given: A cast iron water pipe is two-thirds full of flowing water (60 F). The pipe has an inside diameter of 24 inches and a slope of $\frac{3}{4}$ -inch per foot. Note the sketch that follows.



Find: The flow rate in gallons per minute.

Solution:

$$1. \quad Q = 19.65 d^2 \sqrt{\frac{h_L D}{f L}} \quad \dots \text{page 3-4}$$

To determine the flow rate in a partially full pipe, calculate the equivalent pipe diameters for the actual flow area.

$$a = \frac{\pi d^2}{4}, \text{ therefore, } d^2 = \frac{4a}{\pi}$$

$$A = \frac{\pi D^2}{4}, \text{ therefore, } D = \sqrt{\frac{4A}{\pi}}$$

To determine the friction factor from the Moody diagram, an equivalent diameter four times the hydraulic radius is used; refer to page 3-5.

$$\text{hydraulic radius} = \frac{\text{cross sectional flow area}}{\text{wetted perimeter}}$$

$$R_e = 50.6 \frac{Q \rho}{d \mu} \quad \dots \text{page 3-2 or 3-8}$$

2. Depth of flowing water equals:

$$\frac{2}{3} (24) = 16 \text{ in.}$$

$$3. \quad \cos \theta = \frac{4}{r} = \frac{4}{12} = 0.333$$

$$\theta = 70^\circ 32'$$

$$\alpha = 90^\circ - 70^\circ 32' = 19^\circ 28' = 19.47^\circ$$

$$4. \quad \text{Area C} = \frac{\pi d^2}{4} \left[\frac{180 + (2 \times 19.47)}{360} \right]$$

$$\text{Area C} = \frac{\pi 24^2}{4} \left(\frac{218.94}{360} \right) = 275 \text{ in}^2$$

$$5. \quad b = \sqrt{r^2 - 4^2} = \sqrt{12^2 - 16} = 11.31 \text{ in.}$$

$$6. \quad \text{Area A} = \text{Area B} = \frac{1}{2} (4 b) = \frac{1}{2} (4 \times 11.31) \\ \text{Area A or B} = 22.6 \text{ in}^2$$

$$7. \quad \text{The cross sectional area of flow equals:} \\ A + B + C = 22.6 + 22.6 + 275 \\ = 320.2 \text{ in}^2$$

$$8. \quad d^2 = \frac{4a}{\pi} = \frac{4 \times 320.2}{\pi} = 408$$

$$D = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4 \times 320.2}{\pi \times 144}} = 1.683$$

$$9. \quad h_L = \Delta h = \frac{0.75}{12} = 0.0625 \text{ ft per ft}$$

10. The wetted perimeter equals:

$$\pi d \left(\frac{218.94}{360} \right)$$

$$\text{or, } \pi 24 \left(\frac{218.94}{360} \right) = 45.9 \text{ in.}$$

$$11. \quad \text{Hydraulic radius} = \frac{320.2}{45.9} = 6.98 \text{ in.}$$

$$12. \quad \text{Equivalent diameter } d = 4 \times 6.98 = 27.92 \text{ in.}$$

$$13. \quad \text{Relative roughness } \frac{\epsilon}{D} = 0.00036 \dots \text{page A-23}$$

$$14. \quad f = 0.0155 \quad \dots \left\{ \begin{array}{l} \text{assuming fully turbu-} \\ \text{lent flow; page A-23} \end{array} \right.$$

$$15. \quad Q = 19.65 \times 408 \sqrt{\frac{0.0625 \times 1.683}{0.0155 \times 1}} \\ Q = 20900$$

16. Calculate the Reynolds number to check the friction factor assumed in Step 14.

$$17. \quad \rho = 62.34 \quad \dots \text{page A-6}$$

$$18. \quad \mu = 1.1 \quad \dots \text{page A-3}$$

$$19. \quad R_e = \frac{50.6 \times 20900 \times 62.34}{27.92 \times 1.1}$$

$$R_e = 2150000 \text{ or } 2.15 \times 10^6$$

$$20. \quad f = 0.0155 \quad \dots \text{page A-24}$$

21. Since the friction factor assumed in Step 14 and that determined in Step 20 are in agreement, the flow rate will be 20 900 gpm.

22. 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 the calculations repeated until reasonable agreement is reached.

Determination of Boiler Capacity

Example 4-27

Given: A steam boiler operating at 300 psia saturated steam has a maximum capacity of 100,000 pounds per hour.

Find: The boiler capacity in both kilo Btu per hour and in boiler horsepower.

Solutions:

Kilo Btu per Hour:

1. Boiler capacity = $\frac{W(h_g - h_f)}{1000}$ page B-8
2. h_g = total heat of steam page A-15
 $h_g = 1202.8$ Btu/lb
3. h_f = heat of liquid page A-15
 $h_f = 393.8$ Btu/lb
4. Boiler capacity = $\frac{100\,000 (1202.8 - 393.8)}{1000}$
 $= 80\,900$ kilo Btu/hr

Boiler Horsepower:

5. Boiler horsepower = $\frac{W (h_g - h_f)}{970.3 \times 34.5}$
6. For values of h_g and h_f , see Steps 2 and 3.
7. Boiler horsepower = $\frac{100\,000 (1202.8 - 393.8)}{970.3 \times 34.5}$
 $= 2420$

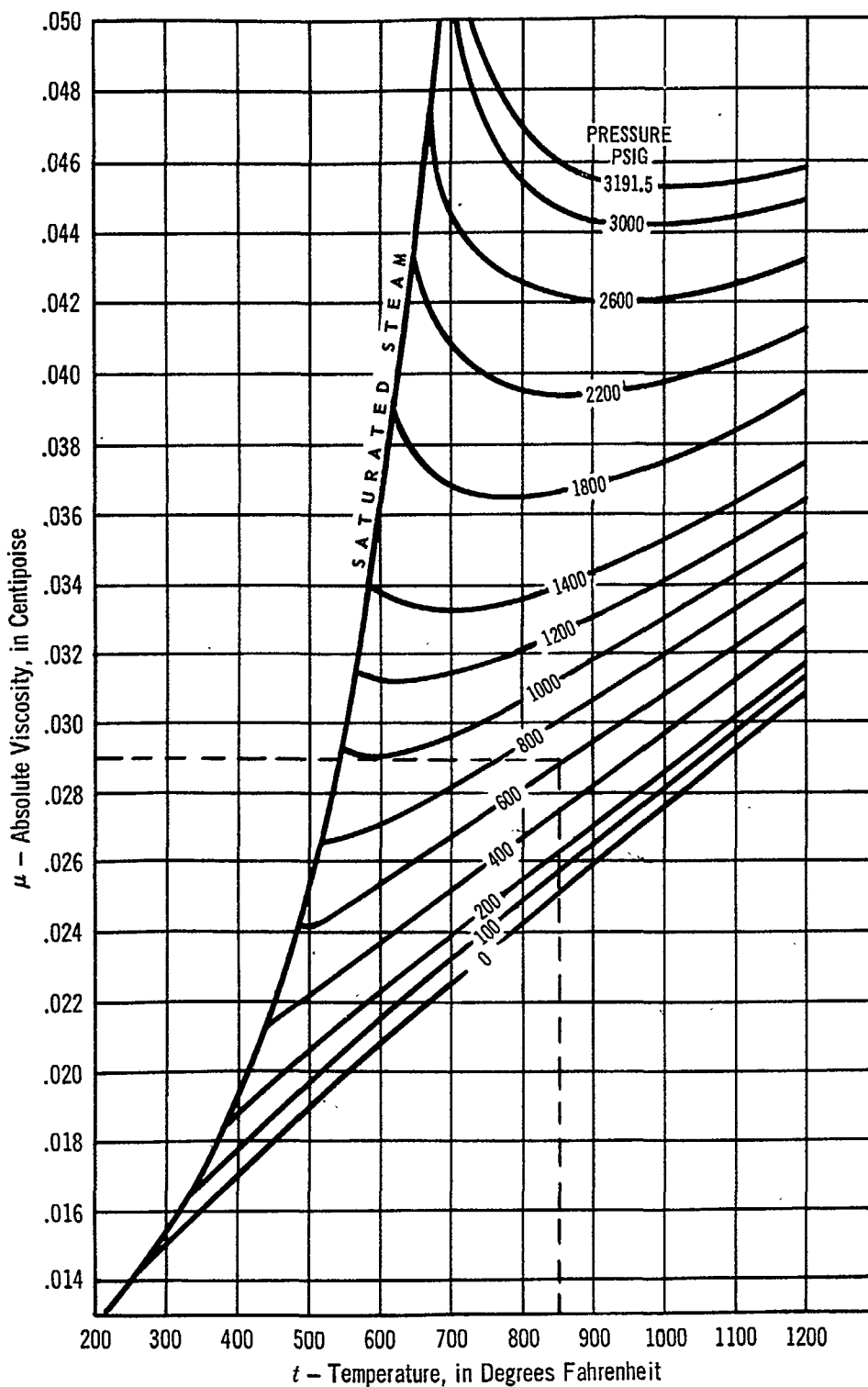
Physical Properties of Fluids and Flow Characteristics of Valves, Fittings, and Pipe

APPENDIX A

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.

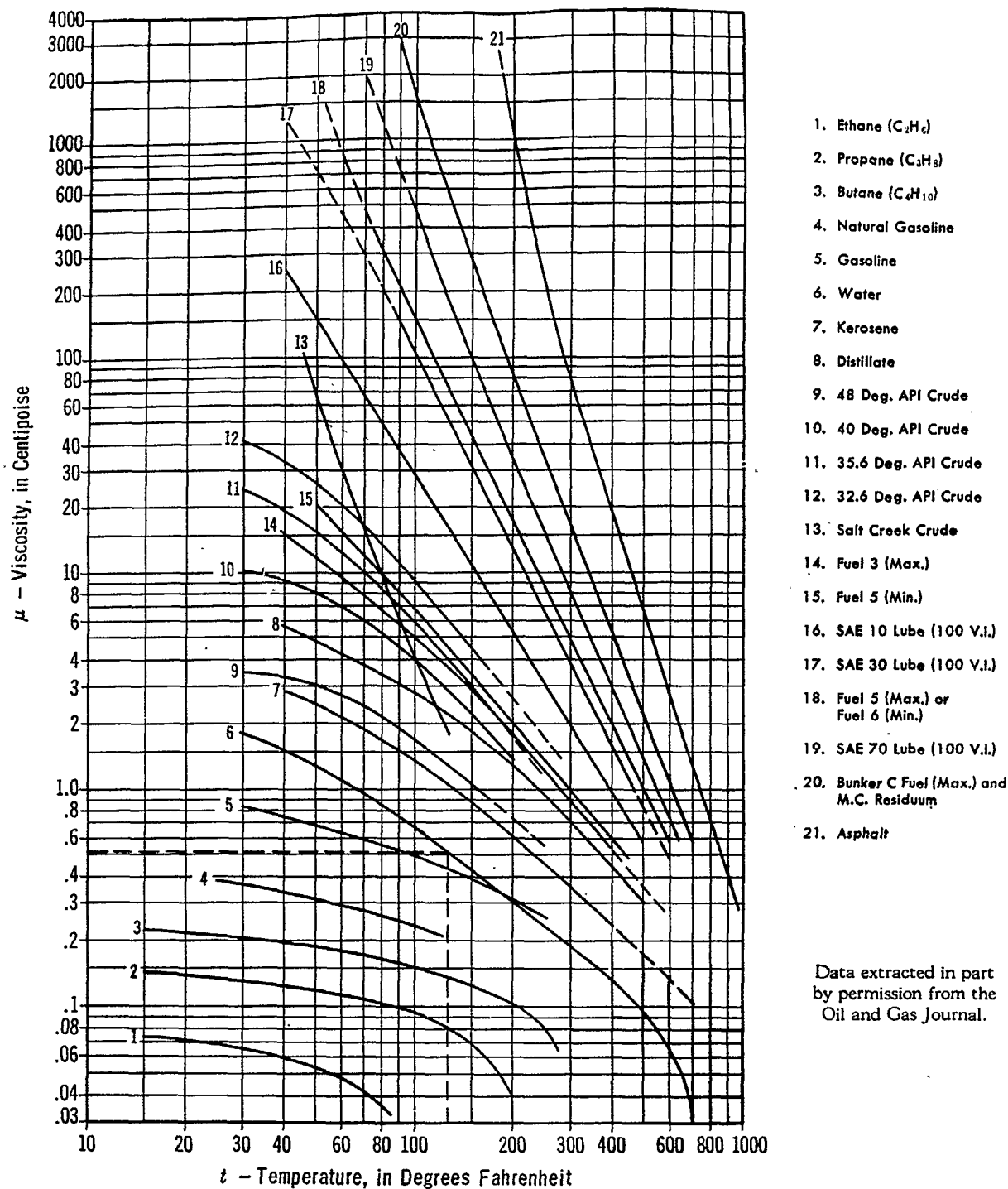
The Y net expansion factors for discharge of compressible fluids from piping systems, which are presented here for the first time, provide means for a greatly simplified solution of a heretofore complex problem.

Viscosity of Steam¹⁴

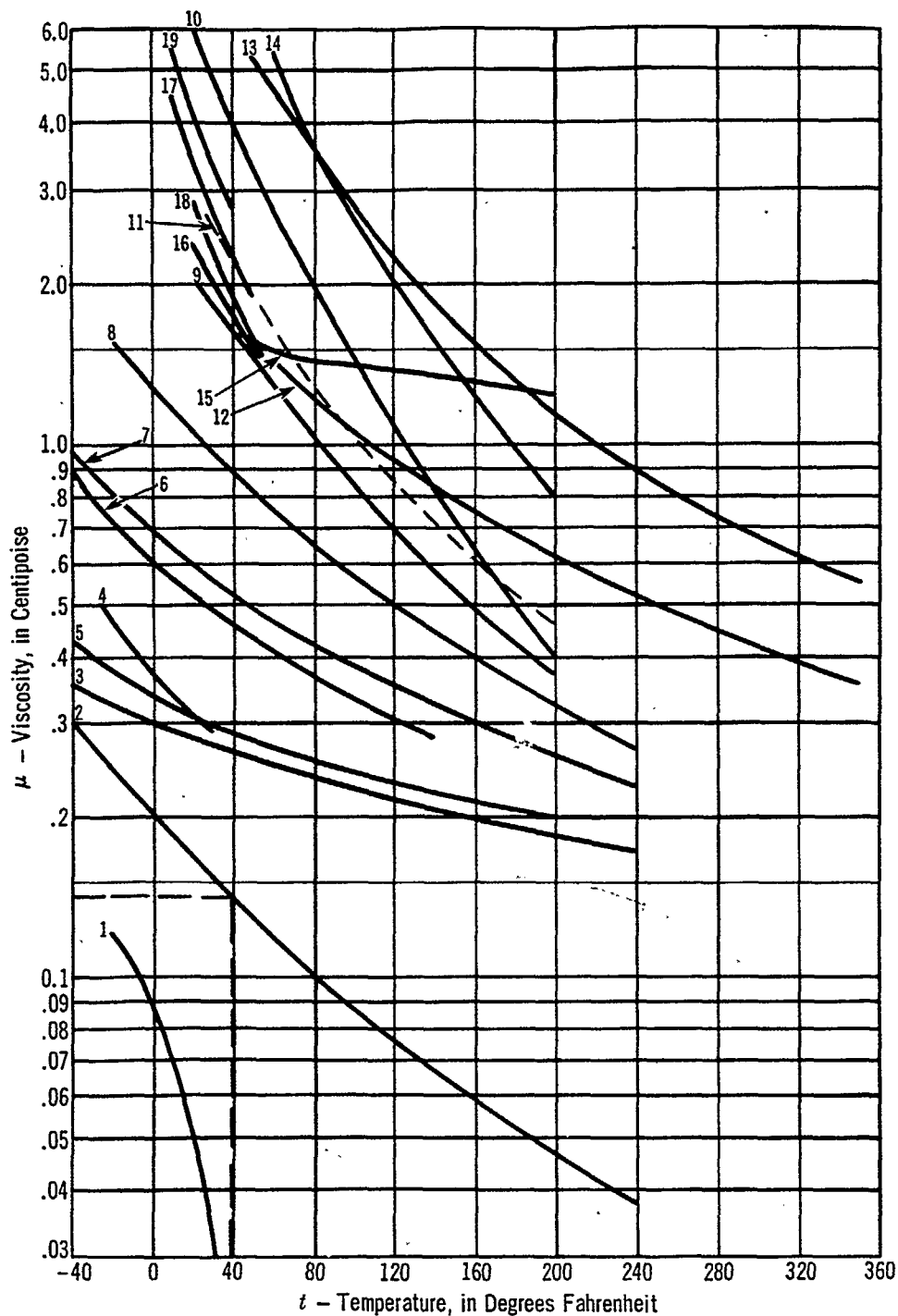
Example: Viscosity of 600 psig,
850 F steam is 0.029 centipoise.

Adapted from:
Philip J. Potter — *Steam Power Plants*,
Copyright 1949, The Ronald Press Company.

Viscosity of Water and Liquid Petroleum Products^{8, 12, 23}



Example: The viscosity of water at 125 F is 0.52 centipoise (Curve No. 6).

Viscosity of Various Liquids^{5, 8, 11}

1. Carbon Dioxide... CO_2
2. Ammonia..... NH_3
3. Methyl Chloride... CH_3Cl
4. Sulphur Dioxide... SO_2
5. Freon 12.....F-12
6. Freon 114.....F-114
7. Freon 11.....F-11
8. Freon 113.....F-113

9. Ethyl Alcohol
10. Isopropyl Alcohol
11. 20% Sulphuric Acid.....20% H_2SO_4
12. Dowtherm E
13. Dowtherm A
14. 20% Sodium Hydroxide...20% NaOH
15. Mercury

16. 10% Sodium Chloride Brine...10% NaCl
17. 20% Sodium Chloride Brine...20% NaCl
18. 10% Calcium Chloride Brine...10% CaCl_2
19. 20% Calcium Chloride Brine...20% CaCl_2

Example: The viscosity of ammonia at 40 F is 0.14 centipoise.

Viscosity of Gases and Vapors

The curves for hydrocarbon vapors and natural gases in the chart at the upper right are taken from Maxwell¹⁵; the curves for all other gases in the chart are based upon Sutherland's formula, as follows:

$$\mu = \mu_0 \left(\frac{0.555 T_0 + C}{0.555 T + C} \right) \left(\frac{T}{T_0} \right)^{3/2}$$

where:

μ = viscosity, in centipoise at temperature T .

μ_0 = viscosity, in centipoise at temperature T_0 .

T = absolute temperature, in degrees Rankine ($460 + \text{deg. F}$) for which viscosity is desired.

T_0 = absolute temperature, in degrees Rankine, for which viscosity is known.

C = Sutherland's constant.

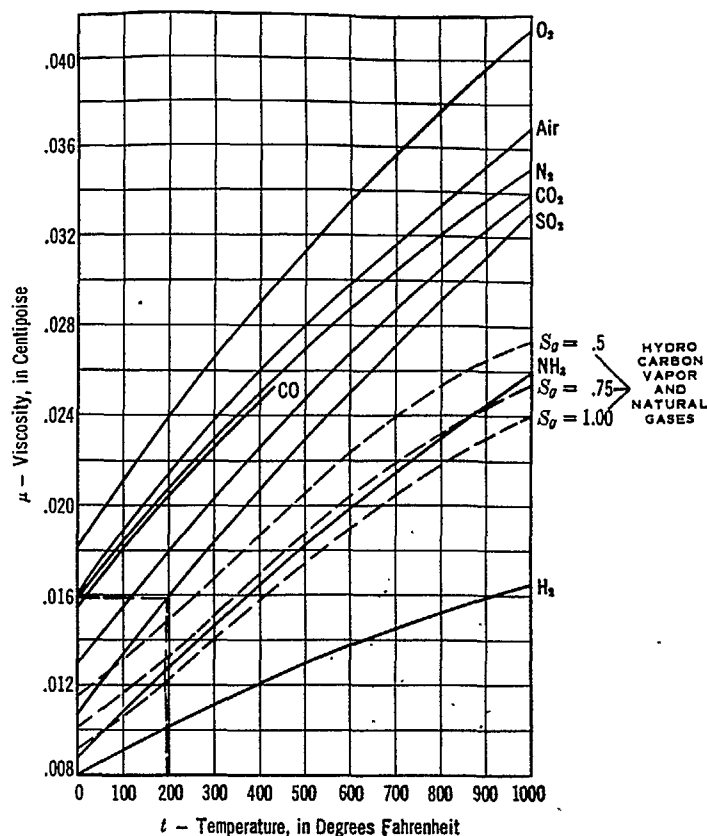
Note: The variation of viscosity with pressure is small for most gases. For gases given on this page, the correction of viscosity for pressure is less than 10 per cent for pressures up to 500 pounds per square inch.

Fluid	Approximate Values of "C"
O ₂	127
Air	120
N ₂	111
CO ₂	240
CO	118
SO ₂	416
NH ₃	370
H ₂	72

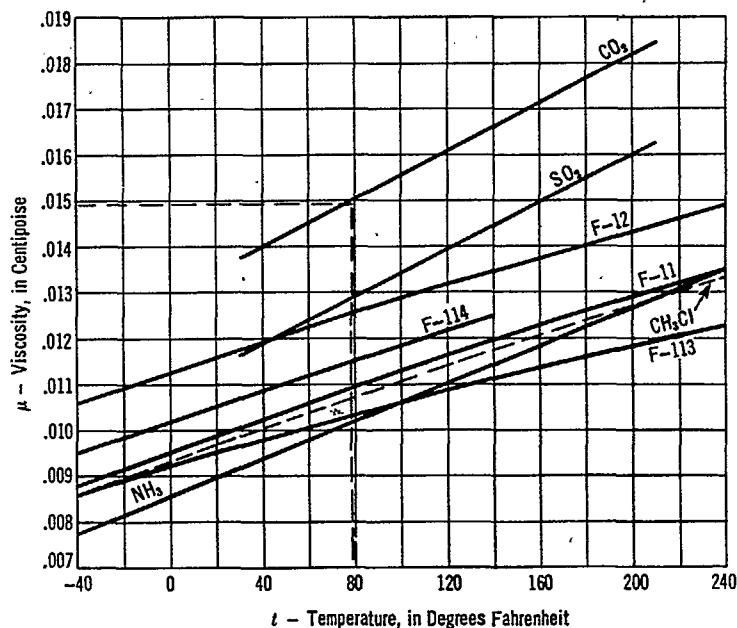
Upper chart example: The viscosity of sulphur dioxide gas (SO₂) at 200 F is 0.016 centipoise.

Lower chart example: The viscosity of carbon dioxide gas (CO₂) at about 80 F is 0.015 centipoise.

Viscosity of Various Gases



Viscosity of Refrigerant Vapors¹¹ (saturated and superheated vapors)



Physical Properties of Water

Temperature of Water	Saturation Pressure	Specific Volume	Weight Density	Weight
<i>t</i>	<i>P'</i>	$\bar{V}$	ρ	
Degrees Fahrenheit	Pounds per Square Inch Absolute	Cubic Feet Per Pound	Pounds per Cubic Foot	Pounds Per Gallon
32	.08854	.01602	62.42	8.345
40	.12170	.01602	62.42	8.345
50	.17811	.01603	62.38	8.340
60	.2563	.01604	62.34	8.334
70	.3631	.01606	62.27	8.325
80	.5069	.01608	62.19	8.314
90	.6982	.01610	62.11	8.303
100	.9492	.01613	62.00	8.289
110	1.2748	.01617	61.84	8.267
120	1.6924	.01620	61.73	8.253
130	2.2225	.01625	61.54	8.227
140	2.8886	.01629	61.39	8.207
150	3.718	.01634	61.20	8.182
160	4.741	.01639	61.01	8.156
170	5.992	.01645	60.79	8.127
180	7.510	.01651	60.57	8.098
190	9.339	.01657	60.35	8.068
200	11.526	.01663	60.13	8.039
210	14.123	.01670	59.88	8.005
212	14.696	.01672	59.81	7.996
220	17.186	.01677	59.63	7.972
240	24.969	.01692	59.10	7.901
260	35.429	.01709	58.51	7.822
280	49.203	.01726	57.94	7.746
300	67.013	.01745	57.31	7.662
350	134.63	.01799	55.59	7.432
400	247.31	.01864	53.65	7.172
450	422.6	.0194	51.55	6.892
500	680.8	.0204	49.02	6.553
550	1045.2	.0218	45.87	6.132
600	1542.9	.0236	42.37	5.664
700	3093.7	.0369	27.10	3.623

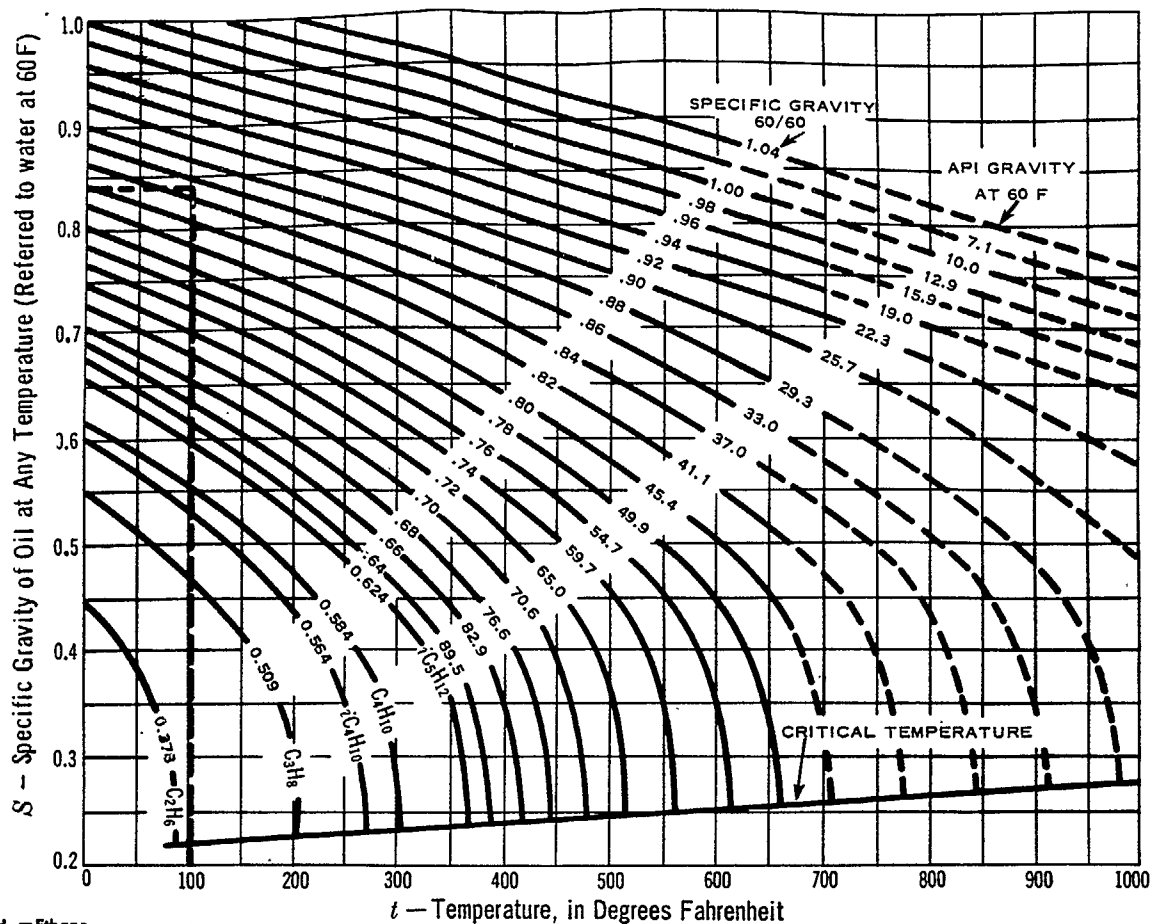
Specific gravity of water at 60 F = 1.00

Weight per gallon is based on 7.48 gallons per cubic foot.

All data on volume and pressure abstracted from Keenan and Keyes' Steam Tables (1936).

Specific Gravity-Temperature Relationship for Petroleum Oils¹²

(Reproduced by permission from the Oil and Gas Journal)



C_2H_6 = Ethane
 C_3H_8 = Propane iC_4H_{10} = Isobutane
 C_4H_{10} = Butane iC_5H_{12} = Isopentane

Example: The specific gravity of an oil at 60 F is 0.85. The specific gravity at 100 F = 0.83.

To find the weight density of a petroleum oil at its flowing temperature when the specific gravity at 60 F/60 F is known, multiply the specific gravity of the oil at flowing temperature (see chart above) by 62.4, the density of water at 60 F.

Weight Density and Specific Gravity* of Various Liquids

Liquid	Temp. <i>t</i> Deg. Fahr.	Weight Density ρ Lbs. per Cu. Ft.	Specific Gravity <i>S</i>	Liquid	Temp. <i>t</i> Deg. Fahr.	Weight Density ρ Lbs. per Cu. Ft.	Specific Gravity <i>S</i>
Acetone	60	49.4	0.792	Mercury	20	849.74	...
Ammonia, Saturated	10	40.9	...	Mercury	40	848.03	...
Benzene	32	56.1	...	Mercury	60	846.32	13.570
Brine, 10% Ca Cl	32	68.05	...	Mercury	80	844.62	...
Brine, 10% Na Cl	32	67.24	...	Mercury	100	842.93	...
Bunkers C Fuel Max.	60	63.25	1.014	Milk	...	...	...
Carbon Disulphide	32	80.6	...	Olive Oil	59	57.3	0.919
Distillate	60	52.99	0.850	Pentane	59	38.9	0.624
Fuel 3 Max.	60	56.02	0.898	SAE 10 Lube†	60	54.64	0.876
Fuel 5 Min.	60	60.23	0.966	SAE 30 Lube†	60	56.02	0.898
Fuel 5 Max.	60	61.92	0.993	SAE 70 Lube†	60	57.12	0.916
Fuel 6 Min.	60	61.92	0.993	Salt Creek Crude	60	52.56	0.843
Gasoline	60.	46.81	0.751	32.6° API Crude	60	53.77	0.862
Gasoline, Natural	60	42.42	0.680	35.6° API Crude	60	52.81	0.847
Kerosene	60	50.85	0.815	40° API Crude	60	51.45	0.825
M. C. Residuum	60	58.32	0.935	48° API Crude	60	49.16	0.788

*Liquid at 60 F referred to water at 60 F.

†Milk has a weight density of 64.2 to 64.6.

‡100 Viscosity Index.

Values in the table at the left were taken from *Smithsonian Physical Tables*, *Mark's Engineers' Handbook*, and ¹²*Nelson's Petroleum Refinery Engineering*.

Physical Properties of Gases¹³

c_p = specific heat at constant pressure
 c_v = specific heat at constant volume

Name of Gas	Chemical Formula or Symbol	Approx. Molecular Weight M	Weight Density, Pounds per Cubic Foot* ρ	Specific Gravity Relative To Air S_g	Individual Gas Constant R	Specific Heat Per Pound at Room Temperature		Heat Capacity Per Cubic Foot at Atmospheric Pressure and 68 F		k equal to c_p/c_v
						c_p	c_v	c_p	c_v	
Acetylene	C ₂ H ₂	26.0	.06754	.897	59.4	.350	.2737	.0236	.0185	1.28
Air	—	29.0	.07528	1.000	53.3	.241	.1725	.0181	.0130	1.40
Ammonia	NH ₃	17.0	.04420	.587	90.8	.523	.4064	.0231	.0179	1.29
Argon	A	40.0	.1037	1.377	38.7	.124	.0743	.0129	.0077	1.67
Carbon Dioxide	CO ₂	44.0	.1142	1.516	35.1	.205	.1599	.0234	.0183	1.28
Carbon Monoxide	CO	28.0	.07269	.965	55.2	.243	.1721	.0177	.0125	1.41
Ethylene	C ₂ H ₄	28.0	.0728	.967	55.1	.40	.3292	.0291	.0240	1.22
Helium	He	4.0	.01039	.138	386.	1.25	.754	.0130	.0078	1.66
Hydrochloric Acid	HCl	36.5	.09460	1.256	42.4	.191	.1365	.0181	.0129	1.40
Hydrogen	H ₂	2.0	.005234	.0695	767.	3.42	2.435	.0179	.0127	1.40
Methane	CH ₄	16.0	.04163	.553	96.4	.593	.4692	.0247	.0195	1.26
Methyl Chloride	CH ₃ Cl	50.5	.1309	1.738	30.6	.24	.2006	.0314	.0263	1.20
Nitrogen	N ₂	28.0	.07274	.966	55.2	.247	.1761	.0179	.0128	1.40
Nitric Oxide	NO	30.0	.07788	1.034	51.5	.231	.1648	.0180	.0128	1.40
Nitrous Oxide	N ₂ O	44.0	.1143	1.518	35.1	.221	.1759	.0253	.0201	1.26
Oxygen	O ₂	32.0	.08305	1.103	48.3	.217	.1549	.0180	.0129	1.40
Sulphur Dioxide	SO ₂	64.0	.1663	2.208	24.1	.154	.1230	.0256	.0204	1.25

*Weight density values are at atmospheric pressure and 68 F.
 For values at 60 F, multiply by 1.0154.

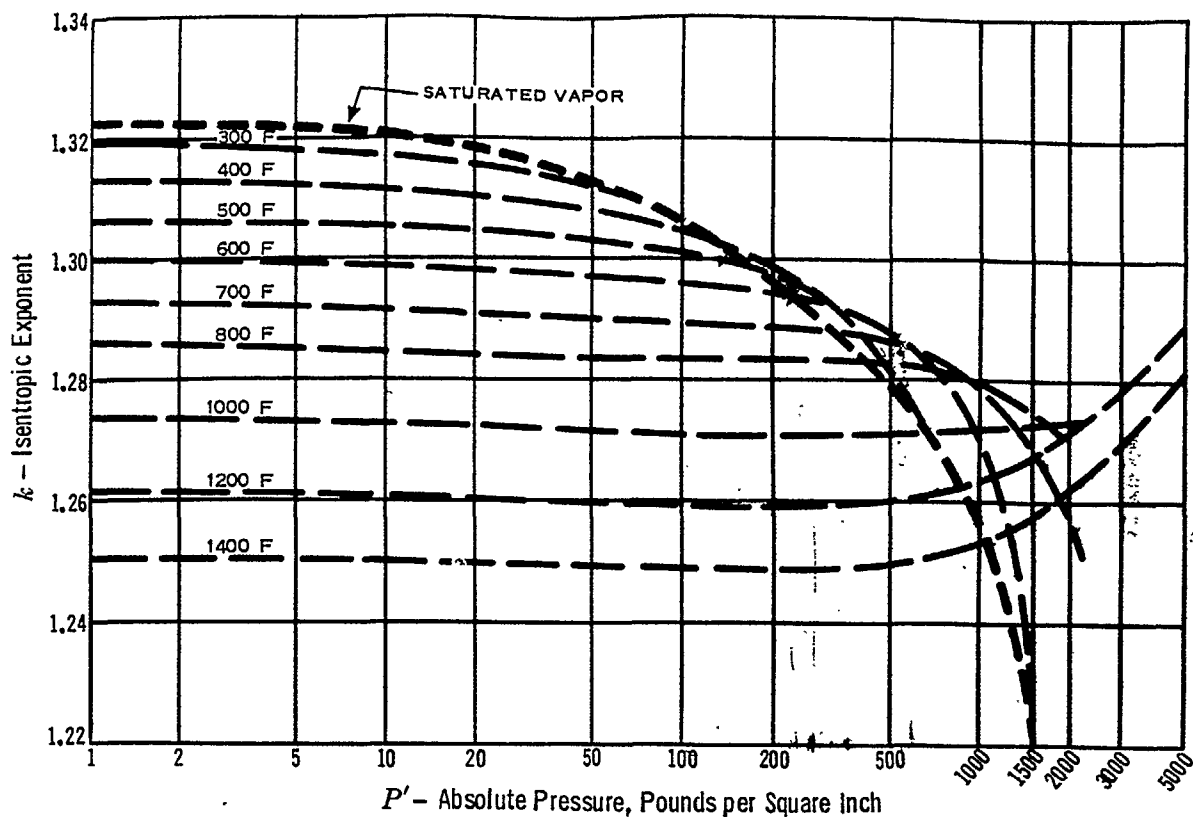
Volumetric Composition and Specific Gravity of Gaseous Fuels¹³

Type of Gas	Chemical Composition Percent by Volume								Specific Gravity Relative to Air S_g	
	Hydro- gen	Carbon Mon- oxide	Paraffin Hydrocarbons		Illuminants		Oxy- gen	Nitro- gen		Carbon Diox- ide
			Meth- ane	Eth- ane	Ethyl- ene	Ben- zene				
Natural Gas, Pittsburgh	...	...	83.4	15.8	...	...	...	0.8	...	0.61
Producer Gas from Bituminous Coal	14.0	27.0	3.0	...	...	...	0.6	50.9	4.5	0.86
Blast Furnace Gas	1.0	27.5	...	...	...	...	...	60.0	11.5	1.02
Blue Water Gas from Coke	47.3	37.0	1.3	...	...	...	0.7	8.3	5.4	0.57
Carbureted Water Gas	40.5	34.0	10.2	...	6.1	2.8	0.5	2.9	3.0	0.63
Coal Gas (Cont. Vertical Retorts)	54.5	10.9	24.2	...	1.5	1.3	0.2	4.4	3.0	0.42
Coke-Oven Gas	46.5	6.3	32.1	...	3.5	0.5	0.8	8.1	2.2	0.44
Refinery Oil Gas (Vapor Phase)	13.1	1.2	23.3	21.7	39.6	...	1.0	...	0.1	0.89
Oil Gas, Pacific Coast	48.6	12.7	26.3	...	2.7	1.1	0.3	3.6	4.7	0.47

Steam — Values of k

Ratio of Specific Heat at Constant Pressure
to Specific Heat at Constant Volume

$$k = c_p / c_v$$



For small changes in pressure (or volume) along an isentropic, $pv^k = \text{constant}$

Reprinted from "Thermodynamic Properties of Steam"
by J. H. Keenan and F. G. Keyes, 1936 edition, by
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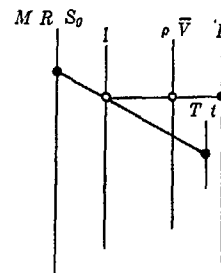
72
580
542.37

Weight Density and Specific Volume Of Gases and Vapors

The chart on page A-11 is based on the formula:

$$\rho = \frac{144 P'}{RT} = \frac{MP'}{10.73 T} = \frac{2.70 P' S_g}{T}$$

where: $P' = 14.7 + P$



Problem: What is the density of dry CH_4 if the temperature is 100 F and the gauge pressure is 15 pounds per square inch?

Solution: Refer to the table on page A-8 for molecular weight, specific gravity, or individual gas constant. Connect 96.4 of the R scale with 100 on the temperature scale, t , and mark the intersection with the index scale. Connect this point with 15 on the pressure scale, P . Read the answer, 0.08 pounds per cubic foot, on the weight density scale ρ .

Weight Density of Air

Air Temp. Deg F.		Weight Density of Air, in Pounds per Cubic Foot For Gauge Pressures Indicated (Based on an atmospheric pressure of 14.696 and a molecular weight of 28.97)																
	0 psi	5 psi	10 psi	20 psi	30 psi	40 psi	50 psi	60 psi	70 psi	80 psi	90 psi	100 psi	110 psi	120 psi	130 psi	140 psi	150 psi	
30°	.0811	.1087	.1363	.1915	.247	.302	.357	.412	.467	.522	.578	.633	.688	.743	.798	.853	.909	
40	.0795	.1065	.1335	.1876	.242	.295	.350	.404	.458	.512	.566	.620	.674	.728	.782	.836	.890	
50	.0782	.1048	.1314	.1846	.238	.291	.344	.397	.451	.504	.557	.610	.663	.717	.770	.823	.877	
60	.0764	.1024	.1284	.1804	.232	.284	.336	.388	.440	.492	.544	.596	.648	.700	.752	.804	.856	
70	.0750	.1005	.1260	.1770	.228	.279	.330	.381	.432	.483	.534	.585	.636	.687	.738	.789	.840	
80	.0736	.0986	.1236	.1737	.224	.274	.324	.374	.424	.474	.524	.574	.624	.674	.724	.774	.824	
90	.0722	.0968	.1214	.1705	.220	.269	.318	.367	.416	.465	.515	.564	.613	.662	.711	.760	.809	
100	.0709	.0951	.1192	.1675	.216	.264	.312	.361	.409	.457	.505	.554	.602	.650	.698	.747	.795	
110	.0697	.0934	.1171	.1645	.212	.259	.307	.354	.402	.449	.497	.544	.591	.639	.686	.734	.781	
120	.0685	.0918	.1151	.1617	.208	.255	.302	.348	.395	.441	.488	.535	.581	.628	.674	.721	.768	
130	.0673	.0902	.1131	.1590	.205	.251	.296	.342	.388	.434	.480	.525	.571	.617	.663	.709	.755	
140	.0662	.0887	.1113	.1563	.201	.246	.291	.337	.382	.427	.472	.517	.562	.607	.652	.697	.742	
150	.0651	.0873	.1094	.1537	.1981	.242	.287	.331	.375	.420	.464	.508	.553	.597	.641	.686	.730	
175	.0626	.0834	.1051	.1477	.1903	.233	.275	.318	.361	.403	.446	.488	.531	.573	.616	.659	.701	
200	.0602	.0807	.1011	.1421	.1831	.224	.265	.306	.347	.388	.429	.470	.511	.552	.593	.634	.675	
225	.0580	.0777	.0974	.1369	.1764	.216	.255	.295	.334	.374	.413	.453	.492	.531	.571	.610	.650	
250	.0559	.0750	.0940	.1321	.1702	.208	.246	.284	.322	.361	.399	.437	.475	.513	.551	.589	.627	
275	.0540	.0724	.0908	.1276	.1644	.201	.238	.275	.311	.348	.385	.422	.459	.495	.532	.569	.606	
300	.0523	.0700	.0878	.1234	.1590	.1945	.230	.266	.301	.337	.372	.408	.443	.479	.515	.550	.586	
350	.0490	.0657	.0824	.1158	.1491	.1825	.216	.249	.283	.316	.349	.383	.416	.449	.483	.516	.550	
400	.0462	.0619	.0776	.1090	.1405	.1719	.203	.235	.266	.298	.329	.360	.392	.423	.455	.486	.518	
450	.0436	.0585	.0733	.1030	.1327	.1624	.1921	.222	.252	.281	.311	.341	.370	.400	.430	.459	.489	
500	.0414	.0555	.0695	.0977	.1258	.1540	.1821	.210	.238	.267	.295	.323	.351	.379	.407	.436	.464	
550	.0393	.0527	.0661	.0928	.1196	.1464	.1731	.1999	.227	.253	.280	.307	.334	.360	.387	.414	.441	
600	.0375	.0502	.0630	.0885	.1140	.1395	.1649	.1904	.216	.241	.267	.292	.318	.343	.369	.394	.420	
	175 psi	200 psi	225 psi	250 psi	300 psi	400 psi	500 psi	600 psi	700 psi	800 psi	900 psi	1000 psi						
30°	1.047	1.185	1.323	1.460	1.736	2.29	2.84	3.39	3.94	4.49	5.05	5.60						
40	1.026	1.161	1.296	1.431	1.702	2.24	2.78	3.32	3.86	4.40	4.95	5.49						
50	1.009	1.142	1.275	1.408	1.674	2.21	2.74	3.27	3.80	4.33	4.87	5.40						
60	.986	1.116	1.246	1.376	1.636	2.16	2.68	3.20	3.72	4.24	4.76	5.28						
70	.968	1.095	1.223	1.350	1.605	2.12	2.63	3.14	3.65	4.16	4.67	5.18						
80	.950	1.075	1.200	1.325	1.575	2.08	2.58	3.08	3.58	4.08	4.58	5.08						
90	.932	1.055	1.178	1.301	1.547	2.04	2.53	3.02	3.51	4.00	4.50	4.99						
100	.916	1.036	1.157	1.278	1.519	2.00	2.48	2.97	3.45	3.93	4.42	4.90						
110	.900	1.018	1.137	1.255	1.492	1.967	2.44	2.92	3.39	3.86	4.34	4.81						
120	.884	1.001	1.117	1.234	1.467	1.933	2.40	2.86	3.33	3.80	4.26	4.73						
130	.869	.984	1.098	1.213	1.442	1.900	2.36	2.82	3.27	3.73	4.19	4.65						
140	.855	.967	1.080	1.193	1.418	1.868	2.32	2.77	3.22	3.67	4.12	4.57						
150	.841	.951	1.062	1.173	1.395	1.838	2.28	2.72	3.17	3.61	4.05	4.50						
175	.807	.914	1.020	1.127	1.340	1.765	2.19	2.62	3.04	3.47	3.89	4.32						
200	.777	.879	.982	1.084	1.289	1.698	2.11	2.52	2.93	3.34	3.75	4.16						
225	.749	.847	.946	1.044	1.242	1.636	2.03	2.43	2.82	3.21	3.61	4.00						
250	.722	.817	.913	1.088	1.198	1.579	1.959	2.34	2.72	3.10	3.48	3.86						
275	.698	.790	.881	.973	1.157	1.525	1.893	2.26	2.63	3.00	3.36	3.73						
300	.675	.764	.852	.941	1.119	1.475	1.830	2.19	2.54	2.90	3.25	3.61						
350	.633	.716	.800	.883	1.050	1.384	1.717	2.05	2.38	2.72	3.05	3.39						
400	.596	.675	.753	.832	.989	1.303	1.618	1.932	2.25	2.56	2.87	3.19						
450	.563	.638	.712	.786	.934	1.232	1.529	1.826	2.12	2.42	2.72	3.01						
500	.534	.604	.675	.745	.886	1.167	1.449	1.731	2.01	2.29	2.58	2.86						
550	.508	.575	.641	.708	.842	1.110	1.377	1.645	1.912	2.18	2.45	2.72						
600	.484	.547	.611	.675	.802	1.057	1.312	1.567	1.822	2.08	2.33	2.59						

Air Density Table

The table at the left is calculated for the perfect gas law shown at the top of the page. Correction for super-compressibility, the deviation from the perfect gas law, would be less than three percent and has not been applied.

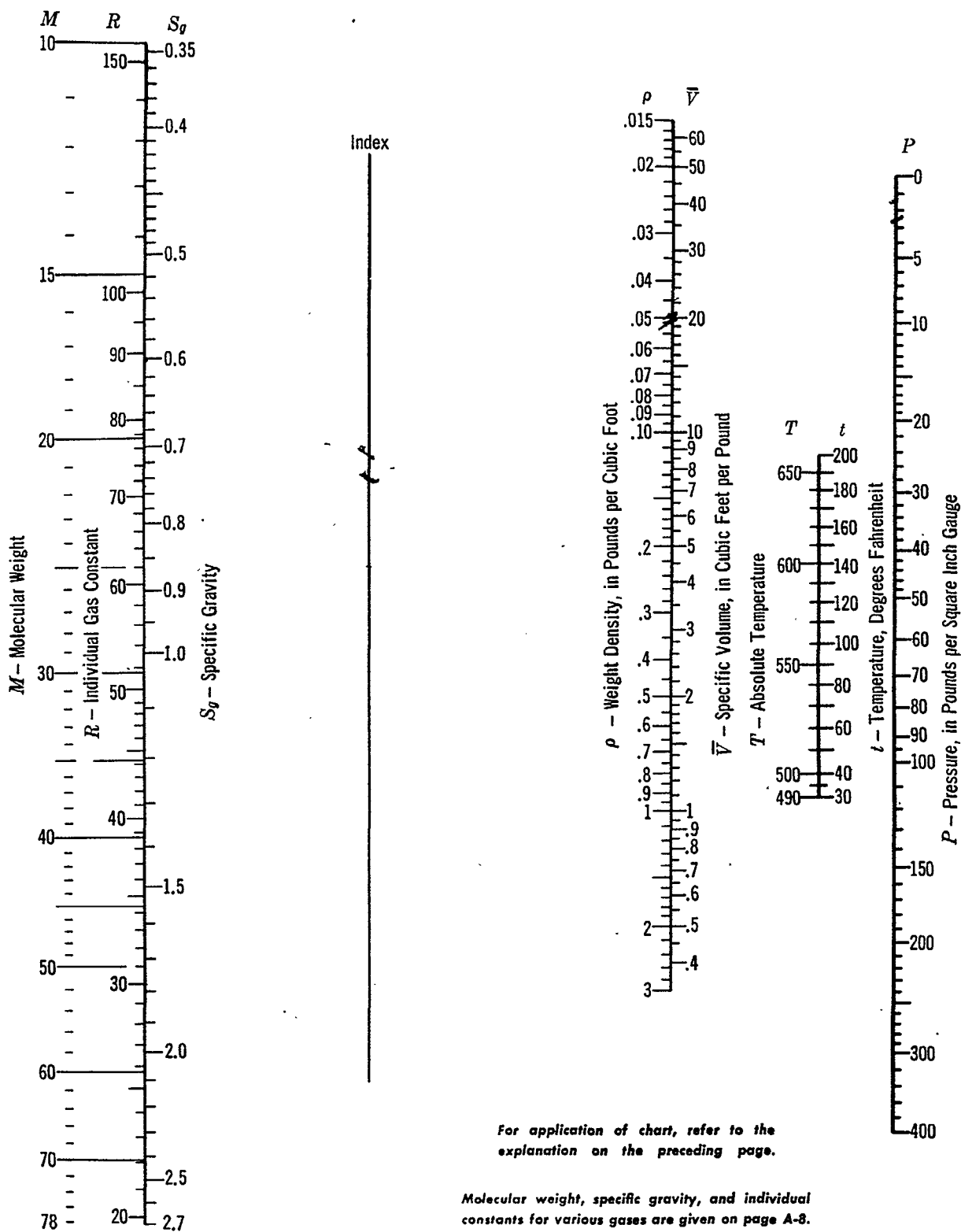
The weight density of gases other than air can be determined from this table by multiplying the density listed for air by the specific gravity of the gas relative to air, as listed in the tables on page A-8.

Air Density Table

The table at the left is calculated for the perfect gas law shown at the top of the page. Correction for supercompressibility, the deviation from the perfect gas law, would be less than three percent and has not been applied.

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-8.

Weight Density and Specific Volume Of Gases and Vapors — continued



Properties of Saturated Steam*

Absolute Pressure		Vacuum Inches of Hg	Temperature t Degrees F.	Heat of the Liquid Btu/lb.	Latent Heat of Evaporation Btu/lb.	Total Heat of Steam h _g Btu/lb.	Specific Volume V Cu. ft. per lb.
Lbs. per Sq. In. P'	Inches of Hg						
0.20	0.41	29.51	53.14	21.21	1063.8	1085.0	1526.0
0.25	0.51	29.41	59.30	27.36	1060.3	1087.7	1235.3
0.30	0.61	29.31	64.47	32.52	1057.4	1090.0	1039.5
0.35	0.71	29.21	68.93	36.97	1054.9	1091.9	898.5
0.40	0.81	29.11	72.86	40.89	1052.7	1093.6	791.9
0.45	0.92	29.00	76.38	44.41	1050.7	1095.1	708.5
0.50	1.02	28.90	79.58	47.60	1048.8	1096.4	641.4
0.60	1.22	28.70	85.21	53.21	1045.7	1098.9	540.0
0.70	1.43	28.49	90.08	58.07	1042.9	1101.0	466.9
0.80	1.63	28.29	94.38	62.36	1040.4	1102.8	411.7
0.90	1.83	28.09	98.24	66.21	1038.3	1104.5	368.4
1.0	2.04	27.88	101.74	69.70	1036.3	1106.0	333.6
1.2	2.44	27.48	107.92	75.87	1032.7	1108.6	280.9
1.4	2.85	27.07	113.26	81.20	1029.6	1110.8	243.0
1.6	3.26	26.66	117.99	85.91	1026.9	1112.8	214.3
1.8	3.66	26.26	122.23	90.14	1024.5	1114.6	191.8
2.0	4.07	25.85	126.08	93.99	1022.2	1116.2	173.73
2.2	4.48	25.44	129.62	97.52	1020.2	1117.7	158.85
2.4	4.89	25.03	132.89	100.79	1018.3	1119.1	146.38
2.6	5.29	24.63	135.94	103.83	1016.5	1120.3	135.78
2.8	5.70	24.22	138.79	106.68	1014.8	1121.5	126.65
3.0	6.11	23.81	141.48	109.37	1013.2	1122.6	118.71
3.5	7.13	22.79	147.57	115.46	1009.6	1125.1	102.72
4.0	8.14	21.78	152.97	120.86	1006.4	1127.3	90.63
4.5	9.16	20.76	157.83	125.71	1003.6	1129.3	81.16
5.0	10.18	19.74	162.24	130.13	1001.0	1131.1	73.52
5.5	11.20	18.72	166.30	134.19	998.5	1132.7	67.24
6.0	12.22	17.70	170.06	137.96	996.2	1134.2	61.98
6.5	13.23	16.69	173.56	141.47	994.1	1135.6	57.50
7.0	14.25	15.67	176.85	144.76	992.1	1136.9	53.64
7.5	15.27	14.65	179.94	147.86	990.2	1138.1	50.29
8.0	16.29	13.63	182.86	150.79	988.5	1139.3	47.34
8.5	17.31	12.61	185.64	153.57	986.8	1140.4	44.73
9.0	18.32	11.60	188.28	156.22	985.2	1141.4	42.40
9.5	19.34	10.58	190.80	158.75	983.6	1142.3	40.31
10.0	20.36	9.56	193.21	161.17	982.1	1143.3	38.42
11.0	22.40	7.52	197.75	165.73	979.3	1145.0	35.14
12.0	24.43	5.49	201.96	169.96	976.6	1146.6	32.40
13.0	26.47	3.45	205.88	173.91	974.2	1148.1	30.06
14.0	28.50	1.42	209.56	177.61	971.9	1149.5	28.04

Pressure Lbs. per Sq. In.		Temperature t Degrees F.	Heat of the Liquid Btu/lb.	Latent Heat of Evaporation Btu/lb.	Total Heat of Steam h _g Btu/lb.	Specific Volume V Cu. ft. per lb.
Absolute P'	Gage P					
14.696	0.0	212.00	180.07	970.3	1150.4	26.80
15.0	0.3	213.03	181.11	969.7	1150.8	26.29
16.0	1.3	216.32	184.42	967.6	1152.0	24.75
17.0	2.3	219.44	187.56	965.5	1153.1	23.39
18.0	3.3	222.41	190.56	963.6	1154.2	22.17
19.0	4.3	225.24	193.42	961.9	1155.3	21.08
20.0	5.3	227.96	196.16	960.1	1156.3	20.089
21.0	6.3	230.57	198.79	958.4	1157.2	19.192
22.0	7.3	233.07	201.33	956.8	1158.1	18.375
23.0	8.3	235.49	203.78	955.2	1159.0	17.627
24.0	9.3	237.82	206.14	953.7	1159.8	16.938
25.0	10.3	240.07	208.42	952.1	1160.6	16.303
26.0	11.3	242.25	210.62	950.7	1161.3	15.715
27.0	12.3	244.36	212.75	949.3	1162.0	15.170
28.0	13.3	246.41	214.83	947.9	1162.7	14.663
29.0	14.3	248.40	216.86	946.5	1163.4	14.189
30.0	15.3	250.33	218.82	945.3	1164.1	13.746
31.0	16.3	252.22	220.73	944.0	1164.7	13.330
32.0	17.3	254.05	222.59	942.8	1165.4	12.940
33.0	18.3	255.84	224.41	941.6	1166.0	12.572
34.0	19.3	257.58	226.18	940.3	1166.5	12.226

* Abstracted from Thermodynamic Properties of Steam by J. H. Keenan and F. G. Keyes, 1936 Edition, by permission of the publishers, John Wiley & Sons, Inc.

(continued on
the next page)

Properties of Saturated Steam — continued

Pressure Lbs. per Sq. In.		Temperature	Heat of the Liquid	Latent Heat of Evaporation	Total Heat of Steam	Specific Volume
Absolute P'	Gage P	t Degrees F.	Btu/lb.	Btu/lb.	h_g Btu/lb.	$\bar{V}$ Cu. ft. per lb.
35.0	20.3	259.28	227.91	939.2	1167.1	11.898
36.0	21.3	260.95	229.60	938.0	1167.6	11.588
37.0	22.3	262.57	231.26	936.9	1168.2	11.294
38.0	23.3	264.16	232.89	935.8	1168.7	11.015
39.0	24.3	265.72	234.48	934.7	1169.2	10.750
40.0	25.3	267.25	236.03	933.7	1169.7	10.498
41.0	26.3	268.74	237.55	932.6	1170.2	10.258
42.0	27.3	270.21	239.04	931.6	1170.7	10.029
43.0	28.3	271.64	240.51	930.6	1171.1	9.810
44.0	29.3	273.05	241.95	929.6	1171.6	9.601
45.0	30.3	274.44	243.36	928.6	1172.0	9.401
46.0	31.3	275.80	244.75	927.7	1172.4	9.209
47.0	32.3	277.13	246.12	926.7	1172.9	9.025
48.0	33.3	278.45	247.47	925.8	1173.3	8.848
49.0	34.3	279.74	248.79	924.9	1173.7	8.678
50.0	35.3	281.01	250.09	924.0	1174.1	8.515
51.0	36.3	282.26	251.37	923.0	1174.4	8.359
52.0	37.3	283.49	252.63	922.2	1174.8	8.208
53.0	38.3	284.70	253.87	921.3	1175.2	8.062
54.0	39.3	285.90	255.09	920.5	1175.6	7.922
55.0	40.3	287.07	256.30	919.6	1175.9	7.787
56.0	41.3	288.23	257.50	918.8	1176.3	7.656
57.0	42.3	289.37	258.67	917.9	1176.6	7.529
58.0	43.3	290.50	259.82	917.1	1176.9	7.407
59.0	44.3	291.61	260.96	916.3	1177.3	7.289
60.0	45.3	292.71	262.09	915.5	1177.6	7.175
61.0	46.3	293.79	263.20	914.7	1177.9	7.064
62.0	47.3	294.85	264.30	913.9	1178.2	6.957
63.0	48.3	295.90	265.38	913.1	1178.5	6.853
64.0	49.3	296.94	266.45	912.3	1178.8	6.752
65.0	50.3	297.97	267.50	911.6	1179.1	6.655
66.0	51.3	298.99	268.55	910.8	1179.4	6.560
67.0	52.3	299.99	269.58	910.1	1179.7	6.468
68.0	53.3	300.98	270.60	909.4	1180.0	6.378
69.0	54.3	301.96	271.61	908.7	1180.3	6.291
70.0	55.3	302.92	272.61	907.9	1180.6	6.206
71.0	56.3	303.88	273.60	907.2	1180.8	6.124
72.0	57.3	304.83	274.57	906.5	1181.1	6.044
73.0	58.3	305.76	275.54	905.8	1181.3	5.966
74.0	59.3	306.68	276.49	905.1	1181.6	5.890
75.0	60.3	307.60	277.43	904.5	1181.9	5.816
76.0	61.3	308.50	278.37	903.7	1182.1	5.743
77.0	62.3	309.40	279.30	903.1	1182.4	5.673
78.0	63.3	310.29	280.21	902.4	1182.6	5.604
79.0	64.3	311.16	281.12	901.7	1182.8	5.537
80.0	65.3	312.03	282.02	901.1	1183.1	5.472
81.0	66.3	312.89	282.91	900.4	1183.3	5.408
82.0	67.3	313.74	283.79	899.7	1183.5	5.346
83.0	68.3	314.59	284.66	899.1	1183.8	5.285
84.0	69.3	315.42	285.53	898.5	1184.0	5.226
85.0	70.3	316.25	286.39	897.8	1184.2	5.168
86.0	71.3	317.07	287.24	897.2	1184.4	5.111
87.0	72.3	317.88	288.08	896.5	1184.6	5.055
88.0	73.3	318.68	288.91	895.9	1184.8	5.001
89.0	74.3	319.48	289.74	895.3	1185.1	4.948
90.0	75.3	320.27	290.56	894.7	1185.3	4.896
91.0	76.3	321.06	291.38	894.1	1185.5	4.845
92.0	77.3	321.83	292.18	893.5	1185.7	4.796
93.0	78.3	322.60	292.98	892.9	1185.9	4.747
94.0	79.3	323.36	293.78	892.3	1186.1	4.699
95.0	80.3	324.12	294.56	891.7	1186.2	4.652
96.0	81.3	324.87	295.34	891.1	1186.4	4.606
97.0	82.3	325.61	296.12	890.5	1186.6	4.561
98.0	83.3	326.35	296.89	889.9	1186.8	4.517
99.0	84.3	327.08	297.65	889.4	1187.0	4.474
100.0	85.3	327.81	298.40	888.8	1187.2	4.432
101.0	86.3	328.53	299.15	888.2	1187.4	4.391
102.0	87.3	329.25	299.90	887.6	1187.5	4.350
103.0	88.3	329.96	300.64	887.1	1187.7	4.310
104.0	89.3	330.66	301.37	886.5	1187.9	4.271
105.0	90.3	331.36	302.10	886.0	1188.1	4.232
106.0	91.3	332.05	302.82	885.4	1188.2	4.194
107.0	92.3	332.74	303.54	884.9	1188.4	4.157
108.0	93.3	333.42	304.26	884.3	1188.6	4.120
109.0	94.3	334.10	304.97	883.7	1188.7	4.084

Properties of Saturated Steam — continued

Pressure Lbs. Per Sq. In.		Temperature	Heat of the Liquid	Latent Heat of Evaporation	Total Heat of Steam	Specific Volume
Absolute P'	Gage P	t Degrees F.	Btu/lb.	Btu/lb.	h _g Btu/lb.	V Cu. ft. per lb.
110.0	95.3	334.77	305.66	883.2	1188.9	4.049
111.0	96.3	335.44	306.37	882.6	1189.0	4.015
112.0	97.3	336.11	307.06	882.1	1189.2	3.981
113.0	98.3	336.77	307.75	881.6	1189.4	3.947
114.0	99.3	337.42	308.43	881.1	1189.5	3.914
115.0	100.3	338.07	309.11	880.6	1189.7	3.882
116.0	101.3	338.72	309.79	880.0	1189.8	3.850
117.0	102.3	339.36	310.46	879.5	1190.0	3.819
118.0	103.3	339.99	311.12	879.0	1190.1	3.788
119.0	104.3	340.62	311.78	878.4	1190.2	3.758
120.0	105.3	341.25	312.44	877.9	1190.4	3.728
121.0	106.3	341.88	313.10	877.4	1190.5	3.699
122.0	107.3	342.50	313.75	876.9	1190.7	3.670
123.0	108.3	343.11	314.40	876.4	1190.8	3.642
124.0	109.3	343.72	315.04	875.9	1190.9	3.614
125.0	110.3	344.33	315.68	875.4	1191.1	3.587
126.0	111.3	344.94	316.31	874.9	1191.2	3.560
127.0	112.3	345.54	316.94	874.4	1191.3	3.533
128.0	113.3	346.13	317.57	873.9	1191.5	3.507
129.0	114.3	346.73	318.19	873.4	1191.6	3.481
130.0	115.3	347.32	318.81	872.9	1191.7	3.455
131.0	116.3	347.90	319.43	872.5	1191.9	3.430
132.0	117.3	348.48	320.04	872.0	1192.0	3.405
133.0	118.3	349.06	320.65	871.5	1192.1	3.381
134.0	119.3	349.64	321.25	871.0	1192.2	3.357
135.0	120.3	350.21	321.85	870.6	1192.4	3.333
136.0	121.3	350.78	322.45	870.1	1192.5	3.310
137.0	122.3	351.35	323.05	869.6	1192.6	3.287
138.0	123.3	351.91	323.64	869.1	1192.7	3.264
139.0	124.3	352.47	324.23	868.7	1192.9	3.242
140.0	125.3	353.02	324.82	868.2	1193.0	3.220
141.0	126.3	353.57	325.40	867.7	1193.1	3.198
142.0	127.3	354.12	325.98	867.2	1193.2	3.177
143.0	128.3	354.67	326.56	866.7	1193.3	3.155
144.0	129.3	355.21	327.13	866.3	1193.4	3.134
145.0	130.3	355.76	327.70	865.8	1193.5	3.114
146.0	131.3	356.29	328.27	865.3	1193.6	3.094
147.0	132.3	356.83	328.83	864.9	1193.8	3.074
148.0	133.3	357.36	329.39	864.5	1193.9	3.054
149.0	134.3	357.89	329.95	864.0	1194.0	3.034
150.0	135.3	358.42	330.51	863.6	1194.1	3.015
152.0	137.3	359.46	331.61	862.7	1194.3	2.977
154.0	139.3	360.49	332.70	861.8	1194.5	2.940
156.0	141.3	361.52	333.79	860.9	1194.7	2.904
158.0	143.3	362.53	334.86	860.0	1194.9	2.869
160.0	145.3	363.53	335.93	859.2	1195.1	2.834
162.0	147.3	364.53	336.98	858.3	1195.3	2.801
164.0	149.3	365.51	338.02	857.5	1195.5	2.768
166.0	151.3	366.48	339.05	856.6	1195.7	2.736
168.0	153.3	367.45	340.07	855.7	1195.8	2.705
170.0	155.3	368.41	341.09	854.9	1196.0	2.675
172.0	157.3	369.35	342.10	854.1	1196.2	2.645
174.0	159.3	370.29	343.10	853.3	1196.4	2.616
176.0	161.3	371.22	344.09	852.4	1196.5	2.587
178.0	163.3	372.14	345.06	851.6	1196.7	2.559
180.0	165.3	373.06	346.03	850.8	1196.9	2.532
182.0	167.3	373.96	347.00	850.0	1197.0	2.505
184.0	169.3	374.86	347.96	849.2	1197.2	2.479
186.0	171.3	375.75	348.92	848.4	1197.3	2.454
188.0	173.3	376.64	349.86	847.6	1197.5	2.429
190.0	175.3	377.51	350.79	846.8	1197.6	2.404
192.0	177.3	378.38	351.72	846.1	1197.8	2.380
194.0	179.3	379.24	352.64	845.3	1197.9	2.356
196.0	181.3	380.10	353.55	844.5	1198.1	2.333
198.0	183.3	380.95	354.46	843.7	1198.2	2.310
200.0	185.3	381.79	355.36	843.0	1198.4	2.288
205.0	190.3	383.86	357.58	841.1	1198.7	2.234
210.0	195.3	385.90	359.77	839.2	1199.0	2.183
215.0	200.3	387.89	361.91	837.4	1199.3	2.134
220.0	205.3	389.86	364.02	835.6	1199.6	2.087
225.0	210.3	391.79	366.09	833.8	1199.9	2.0422
230.0	215.3	393.68	368.13	832.0	1200.1	1.9992
235.0	220.3	395.54	370.14	830.3	1200.4	1.9579
240.0	225.3	397.37	372.12	828.5	1200.6	1.9183
245.0	230.3	399.18	374.08	826.8	1200.9	1.8803

Properties of Saturated Steam — concluded

Pressure Lbs. Per Sq. In.		Temperature	Heat of the Liquid	Latent Heat of Evaporation	Total Heat of Steam	Specific Volume
Absolute P'	Gage P	t Degrees F.	Btu/lb.	Btu/lb.	h_g Btu/lb.	$\bar{V}$ Cu. ft. per lb.
250.0	235.3	400.95	376.00	825.1	1201.1	1.8438
255.0	240.3	402.70	377.89	823.4	1201.3	1.8086
260.0	245.3	404.42	379.76	821.8	1201.5	1.7748
265.0	250.3	406.11	381.60	820.1	1201.7	1.7422
270.0	255.3	407.78	383.42	818.5	1201.9	1.7107
275.0	260.3	409.43	385.21	816.9	1202.1	1.6804
280.0	265.3	411.05	386.98	815.3	1202.3	1.6511
285.0	270.3	412.65	388.73	813.7	1202.4	1.6228
290.0	275.3	414.23	390.46	812.1	1202.6	1.5954
295.0	280.3	415.79	392.16	810.5	1202.7	1.5689
300.0	285.3	417.33	393.84	809.0	1202.8	1.5433
320.0	305.3	423.29	400.39	803.0	1203.4	1.4485
340.0	325.3	428.97	406.66	797.1	1203.7	1.3645
360.0	345.3	434.40	412.67	791.4	1204.1	1.2895
380.0	365.3	439.60	418.45	785.8	1204.3	1.2222
400.0	385.3	444.59	424.0	780.5	1204.5	1.1613
420.0	405.3	449.39	429.4	775.2	1204.6	1.1061
440.0	425.3	454.02	434.6	770.0	1204.6	1.0556
460.0	445.3	458.50	439.7	764.9	1204.6	1.0094
480.0	465.3	462.82	444.6	759.9	1204.5	0.9670
500.0	485.3	467.01	449.4	755.0	1204.4	0.9278
520.0	505.3	471.07	454.1	750.1	1204.2	0.8915
540.0	525.3	475.01	458.6	745.4	1204.0	0.8578
560.0	545.3	478.85	463.0	740.8	1203.8	0.8265
580.0	565.3	482.58	467.4	736.1	1203.5	0.7973
600.0	585.3	486.21	471.6	731.6	1203.2	0.7698
620.0	605.3	489.75	475.7	727.2	1202.9	0.7440
640.0	625.3	493.21	479.8	722.7	1202.5	0.7198
660.0	645.3	496.58	483.8	718.3	1202.1	0.6971
680.0	665.3	499.88	487.7	714.0	1201.7	0.6757
700.0	685.3	503.10	491.5	709.7	1201.2	0.6554
720.0	705.3	506.25	495.3	705.4	1200.7	0.6362
740.0	725.3	509.34	499.0	701.2	1200.2	0.6180
760.0	745.3	512.36	502.6	697.1	1199.7	0.6007
780.0	765.3	515.33	506.2	692.9	1199.1	0.5843
800.0	785.3	518.23	509.7	688.9	1198.6	0.5687
820.0	805.3	521.08	513.2	684.8	1198.0	0.5538
840.0	825.3	523.88	516.6	680.8	1197.4	0.5396
860.0	845.3	526.63	520.0	676.8	1196.8	0.5260
880.0	865.3	529.33	523.3	672.8	1196.1	0.5130
900.0	885.3	531.98	526.6	668.8	1195.4	0.5006
920.0	905.3	534.59	529.8	664.9	1194.7	0.4886
940.0	925.3	537.16	533.0	661.0	1194.0	0.4772
960.0	945.3	539.68	536.2	657.1	1193.3	0.4663
980.0	965.3	542.17	539.3	653.3	1192.6	0.4557
1000.0	985.3	544.61	542.4	649.4	1191.8	0.4456
1050.0	1035.3	550.57	550.0	639.9	1189.9	0.4218
1100.0	1085.3	556.31	557.4	630.4	1187.8	0.4001
1150.0	1135.3	561.86	564.6	621.0	1185.6	0.3802
1200.0	1185.3	567.22	571.7	611.7	1183.4	0.3619
1250.0	1235.3	572.42	578.6	602.4	1181.0	0.3450
1300.0	1285.3	577.46	585.4	593.2	1178.6	0.3293
1350.0	1335.3	582.35	592.1	584.0	1176.1	0.3148
1400.0	1385.3	587.10	598.7	574.7	1173.4	0.3012
1450.0	1435.3	591.73	605.2	565.5	1170.7	0.2884
1500.0	1485.3	596.23	611.6	556.3	1167.9	0.2765
1600.0	1585.3	604.90	624.1	538.0	1162.1	0.2548
1700.0	1685.3	613.15	636.3	519.6	1155.9	0.2354
1800.0	1785.3	621.03	648.3	501.1	1149.4	0.2179
1900.0	1885.3	628.58	660.1	482.4	1142.4	0.2021
2000.0	1985.3	635.82	671.7	463.4	1135.1	0.1878
2100.0	2085.3	642.77	683.3	444.1	1127.4	0.1746
2200.0	2185.3	649.46	694.8	424.4	1119.2	0.1625
2300.0	2285.3	655.91	706.5	403.9	1110.4	0.1513
2400.0	2385.3	662.12	718.4	382.7	1101.1	0.1407
2500.0	2485.3	668.13	730.6	360.5	1091.1	0.1307
2600.0	2585.3	673.94	743.0	337.2	1080.2	0.1213
2700.0	2685.3	679.55	756.2	312.1	1068.3	0.1123
2800.0	2785.3	684.99	770.1	284.7	1054.8	0.1035
2900.0	2885.3	690.26	785.4	253.6	1039.0	0.0947
3000.0	2985.3	695.36	802.5	217.8	1020.3	0.0858
3100.0	3085.3	700.31	825.0	168.1	993.1	0.0753
3200.0	3185.3	705.11	872.4	62.0	934.4	0.0580
3206.2	3191.5	705.40	902.7	0.0	902.7	0.0503

Properties of Superheated Steam*

 $\bar{V}$ = specific volume, cubic feet per pound h_g = total heat of steam, Btu per pound

Pressure Lbs. per Sq. In.		Sat. Temp. t		Total Temperature—Degrees Fahrenheit (t)										
Abs. P'	Gage P			360°	400°	440°	480°	500°	600°	700°	800°	900°	1000°	1200°
14.696	0.0	212.00	$\bar{V}_{h_g}$	33.03 1221.1	34.68 1239.9	36.32 1258.8	37.96 1277.6	38.78 1287.1	42.86 1334.8	46.94 1383.2	51.00 1432.3	55.07 1482.3	59.13 1533.1	67.25 1637.5
20.0	5.3	227.96	$\bar{V}_{h_g}$	24.21 1220.3	25.43 1239.2	26.65 1258.2	27.86 1277.1	28.46 1286.6	31.47 1334.4	34.47 1382.9	37.46 1432.1	40.45 1482.1	43.44 1533.0	49.41 1637.4
30.0	15.3	250.33	$\bar{V}_{h_g}$	16.072 1218.6	16.897 1237.9	17.714 1257.0	18.528 1276.2	18.933 1285.7	20.95 1333.8	22.96 1382.4	24.96 1431.7	26.95 1481.8	28.95 1532.7	32.93 1637.2
40.0	25.3	267.25	$\bar{V}_{h_g}$	12.001 1216.9	12.628 1236.5	13.247 1255.9	13.862 1275.2	14.168 1284.8	15.688 1333.1	17.198 1381.9	18.702 1431.3	20.20 1481.4	21.70 1532.4	24.69 1637.0
50.0	35.3	281.01	$\bar{V}_{h_g}$	9.557 1215.2	10.065 1235.1	10.567 1254.7	11.062 1274.2	11.309 1283.9	12.532 1332.5	13.744 1381.4	14.950 1430.9	16.152 1481.1	17.352 1532.1	19.747 1636.8
60.0	45.3	292.71	$\bar{V}_{h_g}$	7.927 1213.4	8.357 1233.6	8.779 1253.5	9.196 1273.2	9.403 1283.0	10.427 1331.8	11.441 1380.9	12.449 1430.5	13.452 1480.8	14.454 1531.9	16.451 1636.6
70.0	55.3	302.92	$\bar{V}_{h_g}$	6.762 1211.5	7.136 1232.1	7.502 1252.3	7.863 1272.2	8.041 1282.0	8.924 1331.1	9.796 1380.4	10.662 1430.1	11.524 1480.5	12.383 1531.6	14.097 1636.3
80.0	65.3	312.03	$\bar{V}_{h_g}$	5.888 1209.7	6.220 1230.7	6.544 1251.1	6.862 1271.1	7.020 1281.1	7.797 1330.5	8.562 1379.9	9.322 1429.7	10.077 1480.1	10.830 1531.3	12.332 1636.2
90.0	75.3	320.27	$\bar{V}_{h_g}$	5.208 1207.7	5.508 1229.1	5.799 1249.8	6.084 1270.1	6.225 1280.1	6.920 1329.8	7.603 1379.4	8.279 1429.3	8.952 1479.8	9.623 1531.0	10.959 1635.9
100.0	85.3	327.81	$\bar{V}_{h_g}$	4.663 1205.7	4.937 1227.6	5.202 1248.6	5.462 1269.0	5.589 1279.1	6.218 1329.1	6.835 1378.9	7.446 1428.9	8.052 1479.5	8.656 1530.8	9.860 1635.7
120.0	105.3	341.25	$\bar{V}_{h_g}$	3.844 1201.6	4.081 1224.4	4.307 1246.0	4.527 1266.9	4.636 1277.2	5.165 1327.7	5.683 1377.8	6.195 1428.1	6.702 1478.8	7.207 1530.2	8.212 1635.3
140.0	125.3	353.02	$\bar{V}_{h_g}$	3.258 1197.3	3.468 1221.1	3.667 1243.3	3.860 1264.7	3.954 1275.2	4.413 1326.4	4.861 1376.8	5.301 1427.3	5.738 1478.2	6.172 1529.7	7.035 1634.9
160.0	145.3	363.53	$\bar{V}_{h_g}$	...	3.008 1217.6	3.187 1240.6	3.359 1262.4	3.443 1273.1	3.849 1325.0	4.244 1375.7	4.631 1426.4	5.015 1477.5	5.396 1529.1	6.152 1634.5
180.0	165.3	373.06	$\bar{V}_{h_g}$	...	2.649 1214.0	2.813 1237.8	2.969 1260.2	3.044 1271.0	3.411 1323.5	3.764 1374.7	4.110 1425.6	4.452 1476.8	4.792 1528.6	5.466 1634.1
200.0	185.3	381.79	$\bar{V}_{h_g}$	...	2.361 1210.3	2.513 1234.9	2.656 1257.8	2.726 1268.9	3.060 1322.1	3.380 1373.6	3.693 1424.8	4.002 1476.2	4.309 1528.0	4.917 1633.7
220.0	205.3	389.86	$\bar{V}_{h_g}$	...	2.125 1206.5	2.267 1231.9	2.400 1255.4	2.465 1266.7	2.772 1320.7	3.066 1372.6	3.352 1424.0	3.634 1475.5	3.913 1527.5	4.467 1633.3
240.0	225.3	397.37	$\bar{V}_{h_g}$	...	1.9276 1202.5	2.062 1228.8	2.187 1253.0	2.247 1264.5	2.533 1319.2	2.804 1371.5	3.068 1423.2	3.327 1474.8	3.584 1526.9	4.093 1632.9
260.0	245.3	404.42	$\bar{V}_{h_g}$	...	...	1.8882 1225.7	2.006 1250.5	2.063 1262.3	2.330 1317.7	2.582 1370.4	2.827 1422.3	3.067 1474.2	3.305 1526.3	3.776 1632.5
280.0	265.3	411.05	$\bar{V}_{h_g}$	...	...	1.7388 1222.4	1.8512 1247.9	1.9047 1260.0	2.156 1316.2	2.392 1369.4	2.621 1421.5	2.845 1473.5	3.066 1525.8	3.504 1632.1
300.0	285.3	417.33	$\bar{V}_{h_g}$	...	...	1.6090 1219.1	1.7165 1245.3	1.7675 1257.6	2.005 1314.7	2.227 1368.3	2.442 1420.6	2.652 1472.8	2.859 1525.2	3.269 1631.7
320.0	305.3	423.29	$\bar{V}_{h_g}$	...	...	1.4950 1215.6	1.5985 1242.6	1.6472 1255.2	1.8734 1313.2	2.083 1367.2	2.285 1419.8	2.483 1472.1	2.678 1524.7	3.063 1631.3
340.0	325.3	428.97	$\bar{V}_{h_g}$	...	...	1.3941 1212.1	1.4941 1239.9	1.5410 1252.8	1.7569 1311.6	1.9562 1366.1	2.147 1419.0	2.334 1471.5	2.518 1524.1	2.881 1630.9
360.0	345.3	434.40	$\bar{V}_{h_g}$	...	...	1.3041 1208.4	1.4012 1237.1	1.4464 1250.3	1.6533 1310.1	1.8431 1365.0	2.025 1418.1	2.202 1470.8	2.376 1523.5	2.719 1630.5

* Abstracted from Thermodynamic Properties of Steam by J. H. Keenan and F. G. Keyes, 1936 Edition, by permission of the publishers, John Wiley & Sons, Inc

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the next page)

Properties of Superheated Steam — continued

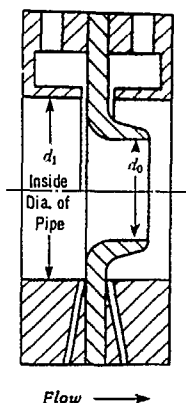
 $\bar{V}$ = specific volume, cubic feet per pound h_g = total heat of steam, Btu per pound

Pressure Lbs. per Sq. In.		Sat. Temp. t		Total Temperature — Degrees Fahrenheit (t)										
Abs. P'	Gage P			500°	540°	600°	640°	660°	700°	740°	800°	900°	1000°	1200°
380.0	365.3	439.60	$\bar{V}$ h_g	1.3616 1247.7	1.4444 1273.1	1.5605 1308.5	1.6345 1331.0	1.6707 1342.0	1.7419 1363.8	1.8118 1385.3	1.9149 1417.3	2.083 1470.1	2.249 1523.0	2.575 1630.0
400.0	385.3	444.59	$\bar{V}$ h_g	1.2851 1245.1	1.3652 1271.0	1.4770 1306.9	1.5480 1329.6	1.5827 1340.8	1.6508 1362.7	1.7177 1384.3	1.8161 1416.4	1.9767 1469.4	2.134 1522.4	2.445 1629.6
420.0	405.3	449.39	$\bar{V}$ h_g	1.2158 1242.5	1.2935 1268.9	1.4014 1305.3	1.4697 1328.3	1.5030 1339.5	1.5684 1361.6	1.6324 1383.3	1.7267 1415.5	1.8802 1468.7	2.031 1521.9	2.327 1629.2
440.0	425.3	454.02	$\bar{V}$ h_g	1.1526 1239.8	1.2282 1266.7	1.3327 1303.6	1.3984 1326.9	1.4306 1338.2	1.4934 1360.4	1.5549 1382.3	1.6454 1414.7	1.7925 1468.1	1.9368 1521.3	2.220 1628.8
460.0	445.3	458.50	$\bar{V}$ h_g	1.0948 1237.0	1.1685 1264.5	1.2698 1302.0	1.3334 1325.4	1.3644 1336.9	1.4250 1359.3	1.4842 1381.3	1.5711 1413.8	1.7124 1467.4	1.8508 1520.7	2.122 1628.4
480.0	465.3	462.82	$\bar{V}$ h_g	1.0417 1234.2	1.1138 1262.3	1.2122 1300.3	1.2737 1324.0	1.3038 1335.6	1.3622 1358.2	1.4193 1380.3	1.5031 1412.9	1.6390 1466.7	1.7720 1520.2	2.033 1628.0
500.0	485.3	467.01	$\bar{V}$ h_g	0.9927 1231.3	1.0633 1260.0	1.1591 1298.6	1.2188 1322.6	1.2478 1334.2	1.3044 1357.0	1.3596 1379.3	1.4405 1412.1	1.5715 1466.0	1.6996 1519.6	1.9504 1627.6
520.0	505.3	471.07	$\bar{V}$ h_g	0.9473 1228.3	1.0166 1257.7	1.1101 1296.9	1.1681 1321.1	1.1962 1332.9	1.2511 1355.8	1.3045 1378.2	1.3826 1411.2	1.5091 1465.3	1.6326 1519.0	1.8743 1627.2
540.0	525.3	475.01	$\bar{V}$ h_g	0.9052 1225.3	0.9733 1255.4	1.0646 1295.2	1.1211 1319.7	1.1485 1331.5	1.2017 1354.6	1.2535 1377.2	1.3291 1410.3	1.4514 1464.6	1.5707 1518.5	1.8039 1626.8
560.0	545.3	478.85	$\bar{V}$ h_g	0.8659 1222.2	0.9330 1253.0	1.0224 1293.4	1.0775 1318.2	1.1041 1330.2	1.1558 1353.5	1.2060 1376.1	1.2794 1409.4	1.3978 1463.9	1.5132 1517.9	1.7385 1626.4
580.0	565.3	482.58	$\bar{V}$ h_g	0.8291 1219.0	0.8954 1250.5	0.9830 1291.7	1.0368 1316.7	1.0627 1328.8	1.1131 1352.3	1.1619 1375.1	1.2331 1408.6	1.3479 1463.2	1.4596 1517.3	1.6776 1626.0
600.0	585.3	486.21	$\bar{V}$ h_g	0.7947 1215.7	0.8602 1248.1	0.9463 1289.9	0.9988 1315.2	1.0241 1327.4	1.0732 1351.1	1.1207 1374.0	1.1899 1407.7	1.3013 1462.5	1.4096 1516.7	1.6208 1625.5
620.0	605.3	489.75	$\bar{V}$ h_g	0.7624 1212.4	0.8272 1245.5	0.9118 1288.1	0.9633 1313.7	0.9880 1326.0	1.0358 1349.9	1.0821 1373.0	1.1494 1406.8	1.2577 1461.8	1.3628 1516.2	1.5676 1625.1
640.0	625.3	493.21	$\bar{V}$ h_g	0.7319 1209.0	0.7962 1243.0	0.8795 1286.2	0.9299 1312.2	0.9541 1324.6	1.0008 1348.6	1.0459 1371.9	1.1115 1405.9	1.2168 1461.1	1.3190 1515.6	1.5178 1624.7
660.0	645.3	496.58	$\bar{V}$ h_g	0.7032 1205.4	0.7670 1240.4	0.8491 1284.4	0.8985 1310.6	0.9222 1323.2	0.9679 1347.4	1.0119 1370.8	1.0759 1405.0	1.1784 1460.4	1.2778 1515.0	1.4709 1624.3
680.0	665.3	499.88	$\bar{V}$ h_g	0.6759 1201.8	0.7395 1237.7	0.8205 1282.5	0.8690 1309.1	0.8922 1321.7	0.9369 1346.2	0.9800 1369.8	1.0424 1404.1	1.1423 1459.7	1.2390 1514.5	1.4269 1623.9
700.0	685.3	503.10	$\bar{V}$ h_g	...	0.7134 1235.0	0.7934 1280.6	0.8411 1307.5	0.8639 1320.3	0.9077 1345.0	0.9498 1368.7	1.0108 1403.2	1.1082 1459.0	1.2024 1513.9	1.3853 1623.5
750.0	735.3	510.86	$\bar{V}$ h_g	...	0.6540 1227.9	0.7319 1275.7	0.7778 1303.5	0.7996 1316.6	0.8414 1341.8	0.8813 1366.0	0.9391 1400.9	1.0310 1457.2	1.1196 1512.4	1.2912 1622.4
800.0	785.3	518.23	$\bar{V}$ h_g	...	0.6015 1220.5	0.6779 1270.7	0.7223 1299.4	0.7433 1312.9	0.7833 1338.6	0.8215 1363.2	0.8763 1398.6	0.9633 1455.4	1.0470 1511.0	1.2088 1621.4
850.0	835.3	525.26	$\bar{V}$ h_g	...	0.5546 1212.7	0.6301 1265.5	0.6732 1295.2	0.6934 1309.0	0.7320 1335.4	0.7685 1360.4	0.8209 1396.3	0.9037 1453.6	0.9830 1509.5	1.1360 1620.4
900.0	885.3	531.98	$\bar{V}$ h_g	...	0.5124 1204.4	0.5873 1260.1	0.6294 1290.9	0.6491 1305.1	0.6863 1332.1	0.7215 1357.5	0.7716 1393.9	0.8506 1451.8	0.9262 1508.1	1.0714 1619.3
950.0	935.3	538.42	$\bar{V}$ h_g	...	0.4740 1195.5	0.5489 1254.6	0.5901 1286.4	0.6092 1301.1	0.6453 1328.7	0.6793 1354.7	0.7275 1391.6	0.8031 1450.0	0.8753 1506.6	1.0136 1618.3
1000.0	985.3	544.61	$\bar{V}$ h_g	...	...	0.5140 1248.8	0.5546 1281.9	0.5733 1297.0	0.6084 1325.3	0.6413 1351.7	0.6878 1389.2	0.7604 1448.2	0.8294 1505.1	0.9615 1617.3

Properties of Superheated Steam — concluded

$\bar{V}$ = specific volume, cubic feet per pound
 h_g = total heat of steam, Btu per pound

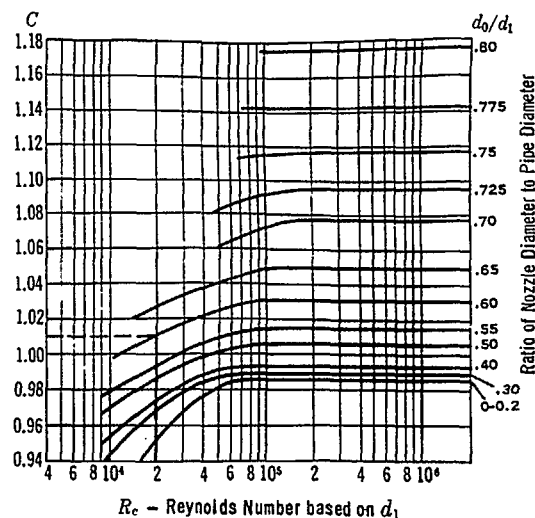
Pressure Lbs. per Sq. In.		Sat. Temp. t		Total Temperature — Degrees Fahrenheit (t)											
Abs. P'	Gage P			660°	700°	740°	760°	780°	800°	860°	900°	1000°	1100°	1200°	
1100.0	1085.3	556.31	$\bar{V}$ h_g	0.5110 1288.5	0.5445 1318.3	0.5755 1345.8	0.5904 1358.9	0.6049 1371.7	0.6191 1384.3	0.6601 1420.8	0.6866 1444.5	0.7503 1502.2	0.8117 1558.8	0.8716 1615.2	
1200.0	1185.3	567.22	$\bar{V}$ h_g	0.4586 1279.6	0.4909 1311.0	0.5206 1339.6	0.5347 1353.2	0.5484 1366.4	0.5617 1379.3	0.6003 1416.7	0.6250 1440.7	0.6843 1499.2	0.7412 1556.4	0.7967 1613.1	
1300.0	1285.3	577.46	$\bar{V}$ h_g	0.4139 1270.2	0.4454 1303.4	0.4739 1333.3	0.4874 1347.3	0.5004 1361.0	0.5131 1374.3	0.5496 1412.5	0.5728 1437.0	0.6284 1496.2	0.6816 1553.9	0.7333 1611.0	
1400.0	1385.3	587.10	$\bar{V}$ h_g	0.3753 1260.3	0.4062 1295.5	0.4338 1326.7	0.4468 1341.3	0.4593 1355.4	0.4714 1369.1	0.5061 1408.2	0.5281 1433.1	0.5805 1493.2	0.6305 1551.4	0.6789 1608.9	
1500.0	1485.3	596.23	$\bar{V}$ h_g	0.3413 1249.8	0.3719 1287.2	0.3989 1320.0	0.4114 1335.2	0.4235 1349.7	0.4352 1363.8	0.4684 1403.9	0.4893 1429.3	0.5390 1490.1	0.5862 1548.9	0.6318 1606.8	
1600.0	1585.3	604.90	$\bar{V}$ h_g	0.3112 1238.7	0.3417 1278.7	0.3682 1313.0	0.3804 1328.8	0.3921 1343.9	0.4034 1358.4	0.4353 1399.5	0.4553 1425.3	0.5027 1487.0	0.5474 1546.4	0.5906 1604.6	
1700.0	1685.3	613.15	$\bar{V}$ h_g	0.2842 1226.8	0.3148 1269.7	0.3410 1305.8	0.3529 1322.3	0.3643 1337.9	0.3753 1352.9	0.4061 1395.0	0.4253 1421.4	0.4706 1484.0	0.5132 1543.8	0.5542 1602.5	
1800.0	1785.3	621.03	$\bar{V}$ h_g	0.2597 1214.0	0.2907 1260.3	0.3166 1298.4	0.3284 1315.5	0.3395 1331.8	0.3502 1347.2	0.3801 1390.4	0.3986 1417.4	0.4421 1480.8	0.4828 1541.3	0.5218 1600.4	
1900.0	1885.3	628.58	$\bar{V}$ h_g	0.2371 1200.2	0.2688 1250.4	0.2947 1290.6	0.3063 1308.6	0.3173 1325.4	0.3277 1341.5	0.3568 1385.8	0.3747 1413.3	0.4165 1477.7	0.4556 1538.8	0.4929 1598.2	
2000.0	1985.3	635.82	$\bar{V}$ h_g	0.2161 1184.9	0.2489 1240.0	0.2748 1282.6	0.2863 1301.4	0.2972 1319.0	0.3074 1335.5	0.3358 1381.2	0.3532 1409.2	0.3935 1474.5	0.4311 1536.2	0.4668 1596.1	
2100.0	2085.3	642.77	$\bar{V}$ h_g	0.1962 1167.7	0.2306 1229.0	0.2567 1274.3	0.2682 1294.0	0.2789 1312.3	0.2890 1329.5	0.3167 1376.4	0.3337 1405.0	0.3727 1471.4	0.4089 1533.6	0.4433 1593.9	
2200.0	2185.3	649.46	$\bar{V}$ h_g	0.1768 1147.8	0.2135 1217.4	0.2400 1265.7	0.2514 1286.3	0.2621 1305.4	0.2721 1323.3	0.2994 1371.5	0.3159 1400.8	0.3538 1468.2	0.3887 1531.1	0.4218 1591.8	
2300.0	2285.3	655.91	$\bar{V}$ h_g	0.1575 1123.8	0.1978 1204.9	0.2247 1256.7	0.2362 1278.4	0.2468 1298.4	0.2567 1316.9	0.2835 1366.6	0.2997 1396.5	0.3365 1464.9	0.3703 1528.5	0.4023 1589.6	
2400.0	2385.3	662.12	$\bar{V}$ h_g	...	0.1828 1191.5	0.2105 1247.3	0.2221 1270.2	0.2327 1291.1	0.2425 1310.3	0.2689 1361.6	0.2848 1392.2	0.3207 1461.7	0.3534 1525.9	0.3843 1587.4	
2500.0	2485.3	668.13	$\bar{V}$ h_g	...	0.1686 1176.8	0.1973 1237.6	0.2090 1261.8	0.2196 1283.6	0.2294 1303.6	0.2555 1356.5	0.2710 1387.8	0.3061 1458.4	0.3379 1523.2	0.3678 1585.3	
2600.0	2585.3	673.94	$\bar{V}$ h_g	...	0.1549 1160.6	0.1849 1227.3	0.1967 1252.9	0.2074 1275.8	0.2172 1296.8	0.2431 1351.4	0.2584 1383.4	0.2926 1455.1	0.3236 1520.6	0.3526 1583.1	
2700.0	2685.3	679.55	$\bar{V}$ h_g	...	0.1415 1142.5	0.1732 1216.5	0.1853 1243.8	0.1960 1267.9	0.2059 1289.7	0.2315 1346.1	0.2466 1378.9	0.2801 1451.8	0.3103 1518.0	0.3385 1580.9	
2800.0	2785.3	684.99	$\bar{V}$ h_g	...	0.1281 1121.4	0.1622 1205.1	0.1745 1234.2	0.1854 1259.6	0.1953 1282.4	0.2208 1340.8	0.2356 1374.3	0.2685 1448.5	0.2979 1515.4	0.3254 1578.7	
2900.0	2885.3	690.26	$\bar{V}$ h_g	...	0.1143 1095.9	0.1517 1193.0	0.1644 1224.3	0.1754 1251.1	0.1853 1274.9	0.2108 1335.3	0.2254 1369.7	0.2577 1445.1	0.2864 1512.7	0.3132 1576.5	
3000.0	2985.3	695.36	$\bar{V}$ h_g	...	0.0984 1060.7	0.1416 1180.1	0.1548 1213.8	0.1660 1242.2	0.1760 1267.2	0.2014 1329.7	0.2159 1365.0	0.2476 1441.8	0.2757 1510.0	0.3018 1574.3	
3100.0	3085.3	700.31	$\bar{V}$ h_g	...	...	0.1320 1166.2	0.1456 1202.9	0.1571 1233.0	0.1672 1259.3	0.1926 1324.1	0.2070 1360.3	0.2382 1438.4	0.2657 1507.4	0.2911 1572.1	
3200.0	3185.3	705.11	$\bar{V}$ h_g	...	...	0.1226 1151.1	0.1369 1191.4	0.1486 1223.5	0.1589 1251.1	0.1843 1318.3	0.1986 1355.5	0.2293 1434.9	0.2563 1504.7	0.2811 1569.9	
3206.2	3191.5	705.40	$\bar{V}$ h_g	...	...	0.1220 1150.2	0.1363 1190.6	0.1480 1222.9	0.1583 1250.5	0.1838 1317.9	0.1981 1355.2	0.2288 1434.7	0.2557 1504.5	0.2806 1569.8	

Flow Coefficient C for Nozzles⁷

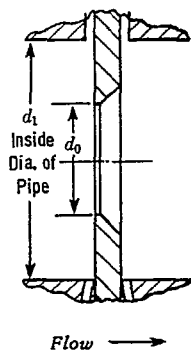
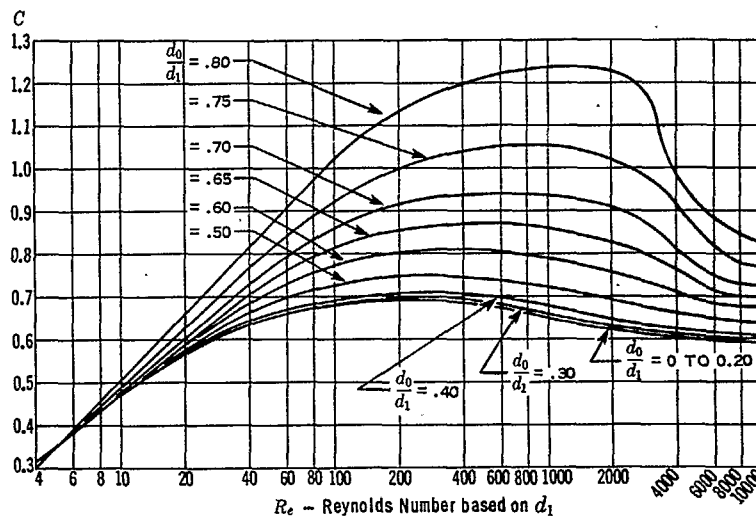
Data from *Regeln fuer die Durchflussmessung mit genormten Duesen und Blenden*. VDI-Verlag G. m.b. H., Berlin, SNW, 7, 1937. Published as Technical Memorandum 952 by the NACA.

$$C = \frac{C_d}{\sqrt{1 - \left(\frac{d_0}{d_1}\right)^4}}$$

Example: The flow coefficient C for a diameter ratio d_0/d_1 of 0.60 at a Reynolds number of 20,000 (2×10^4) equals 1.01.

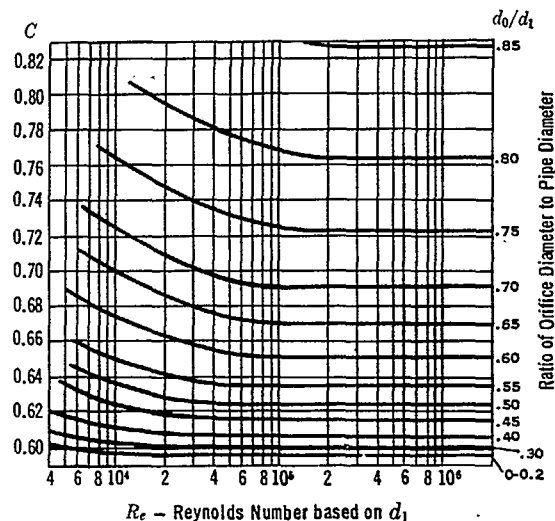
Flow Coefficient C for Square Edged Orifices^{7, 17}

$$C = \frac{C_d}{\sqrt{1 - \left(\frac{d_0}{d_1}\right)^4}}$$

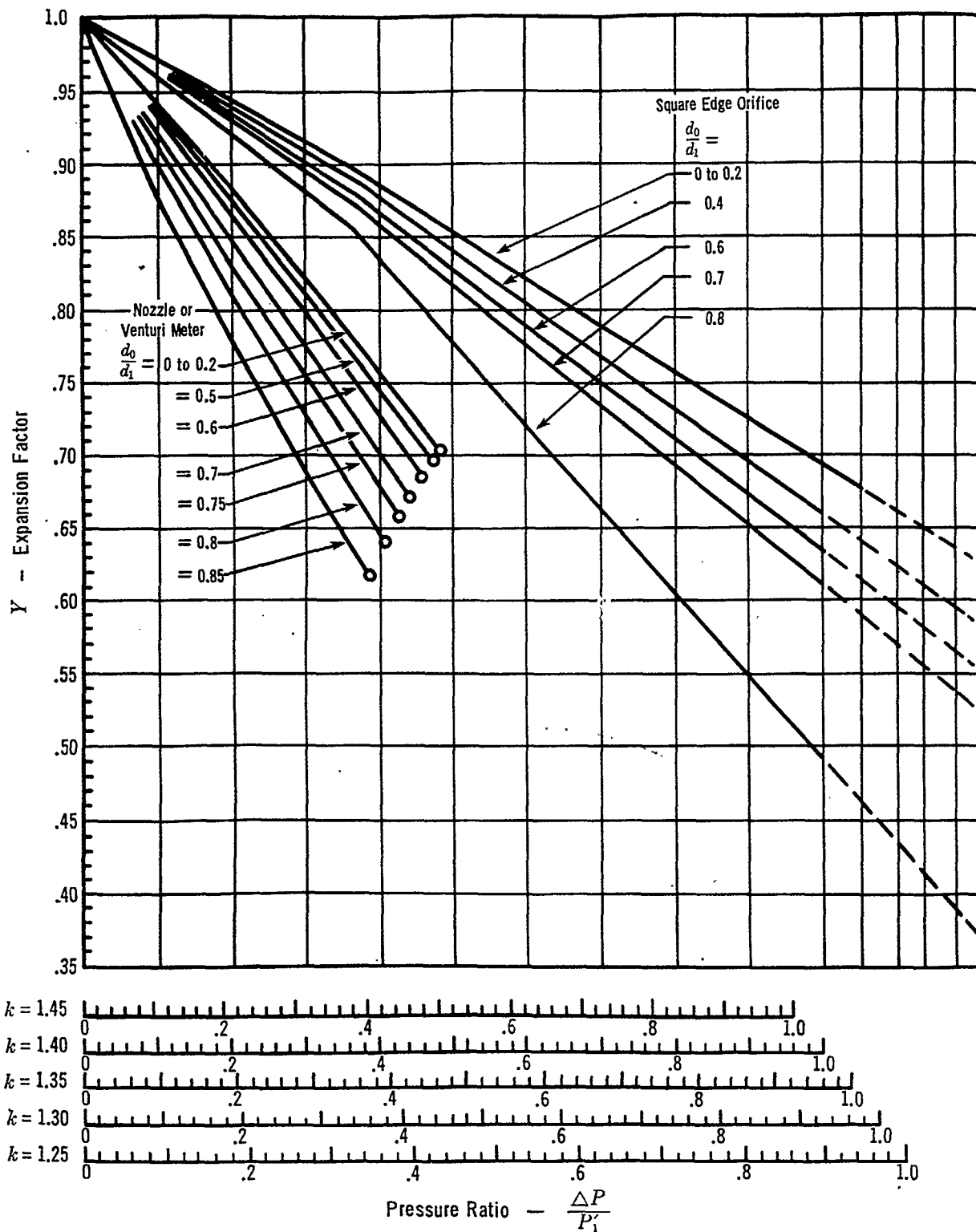


$$C = \frac{C_d}{\sqrt{1 - \left(\frac{d_0}{d_1}\right)^4}}$$

Lower chart data from *Regeln fuer die Durchflussmessung mit genormten Duesen und Blenden*. VDI-Verlag G. m.b.H., Berlin, SNW, 7, 1937. Published as Technical Memorandum 952 by the NACA.

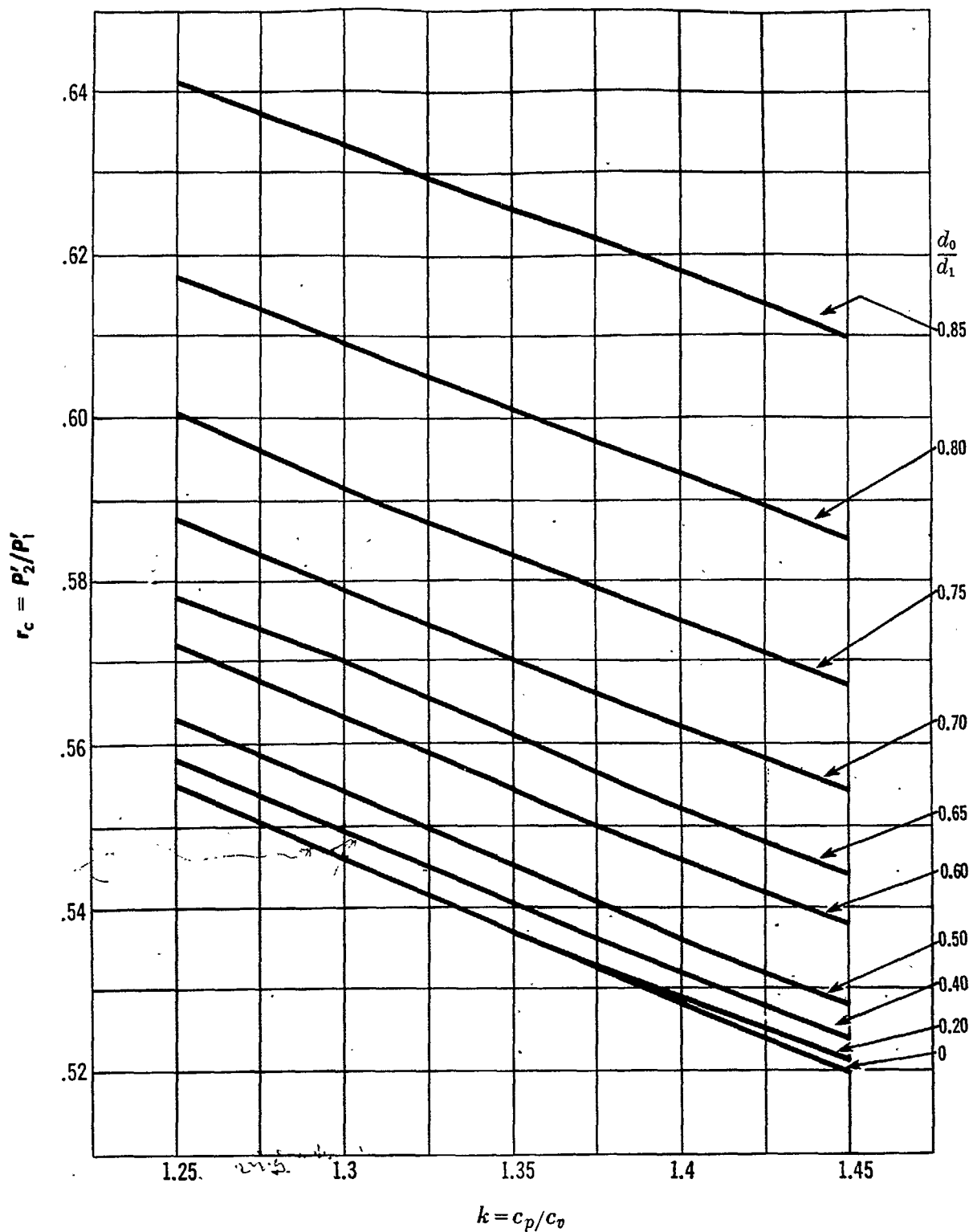


Net Expansion Factor, Y For Compressible Flow through Nozzles and Orifices^{9, 10}



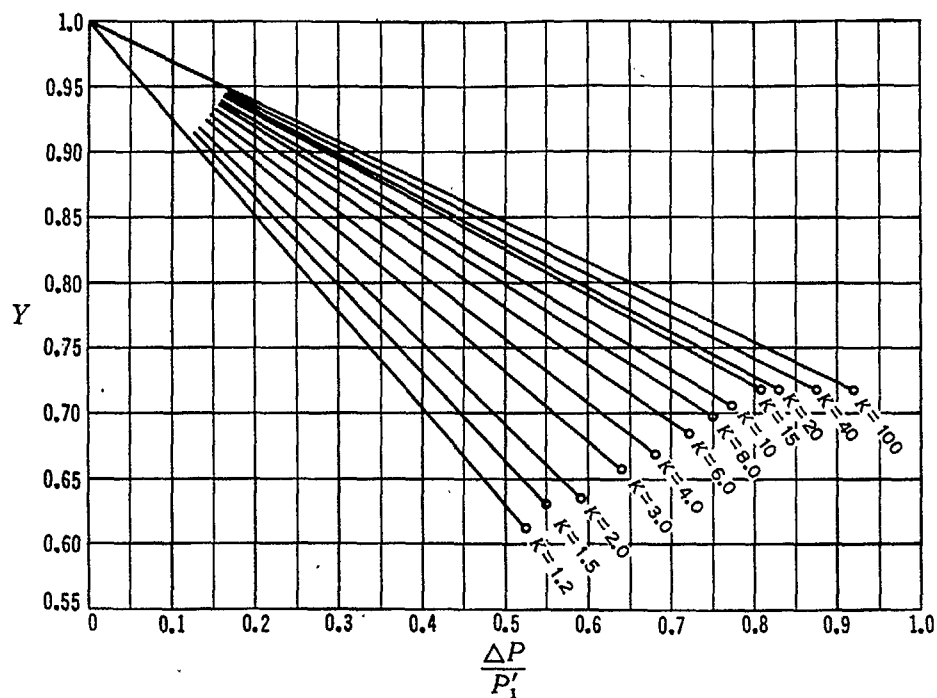
Data extracted from *Fluid Meters, Their Theory and Application*, Fourth Edition, 1937, and *Orifice Meters with Supercritical Flow* by R. G. Cunningham, with permission of the publisher, The American Society of Mechanical Engineers, 29 West 39th Street, New York 18.

Critical Pressure Ratio, r_c
For Compressible Flow through Nozzles and Venturi Tubes⁹



Net Expansion Factor Y for Compressible Flow Through Pipe to a Larger Flow Area

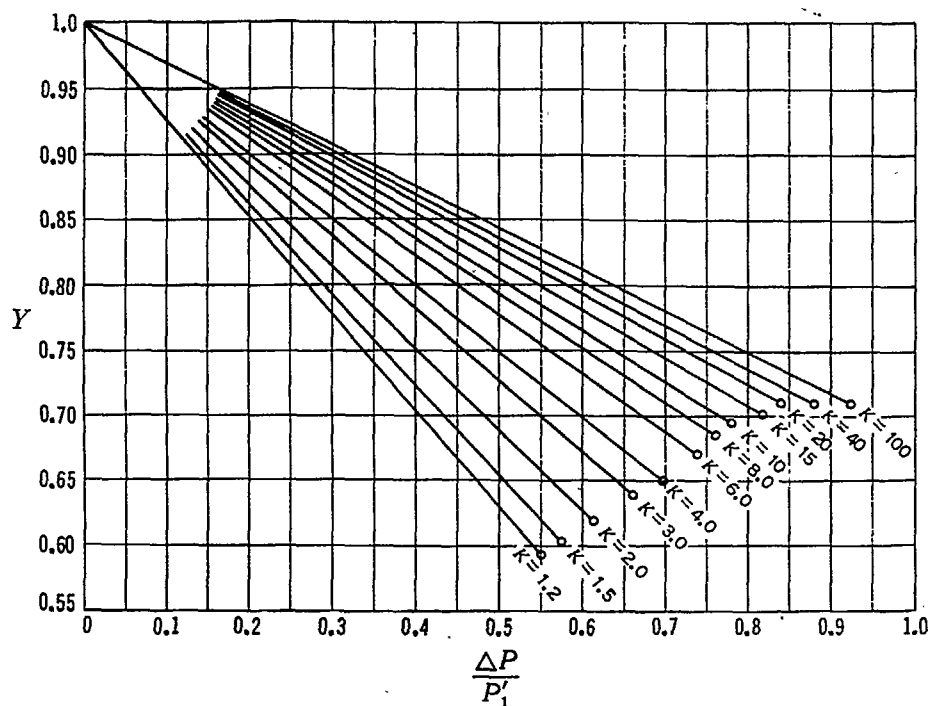
$k = 1.3$



Limiting Factors
For Sonic Velocity
 $k = 1.3$

K	$\frac{\Delta P}{P_1}$	Y
1.2	.525	.612
1.5	.550	.631
2.0	.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

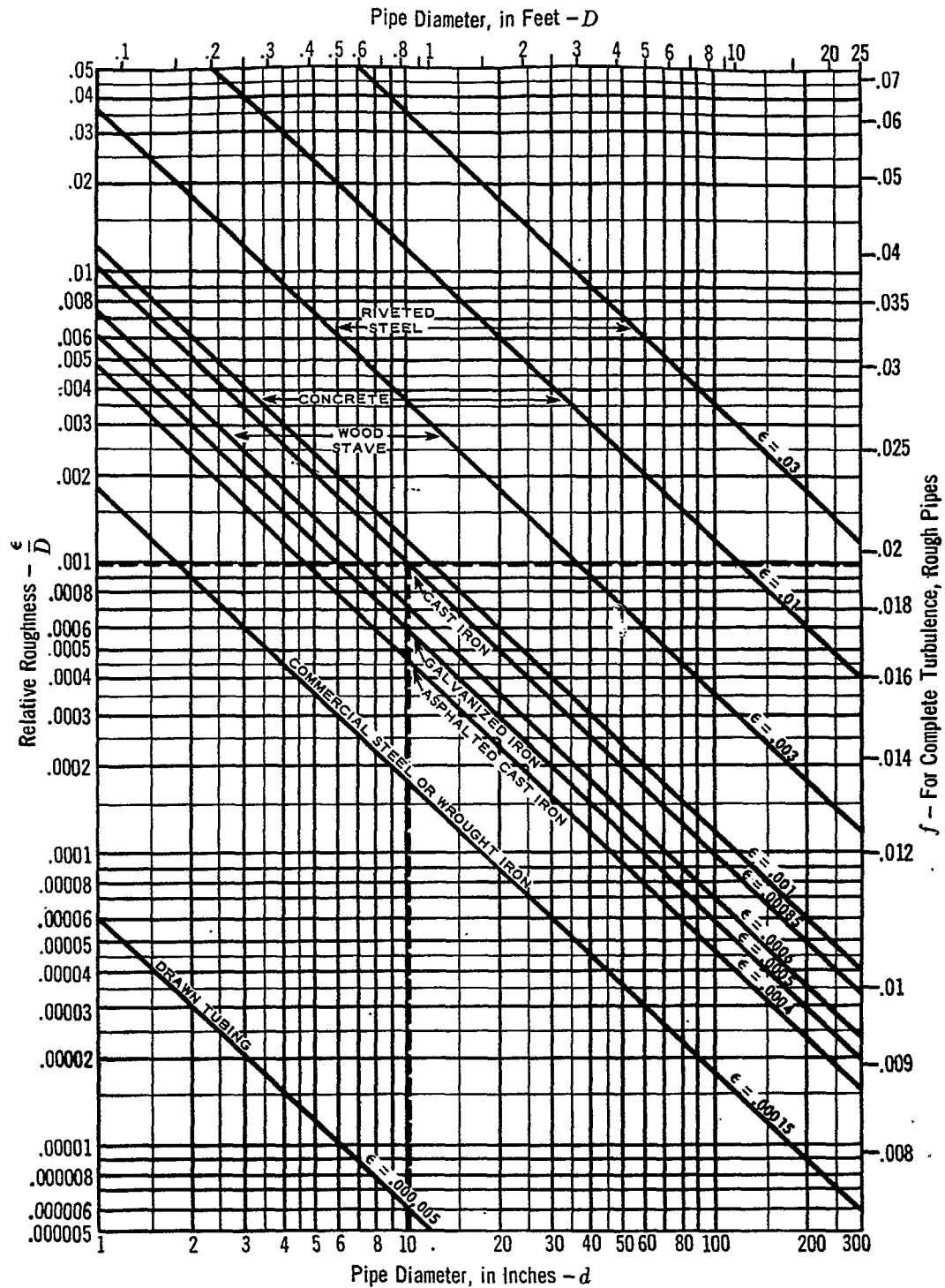
$k = 1.4$



Limiting Factors
For Sonic Velocity
 $k = 1.4$

K	$\frac{\Delta P}{P_1}$	Y
1.2	.552	.588
1.5	.576	.606
2.0	.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

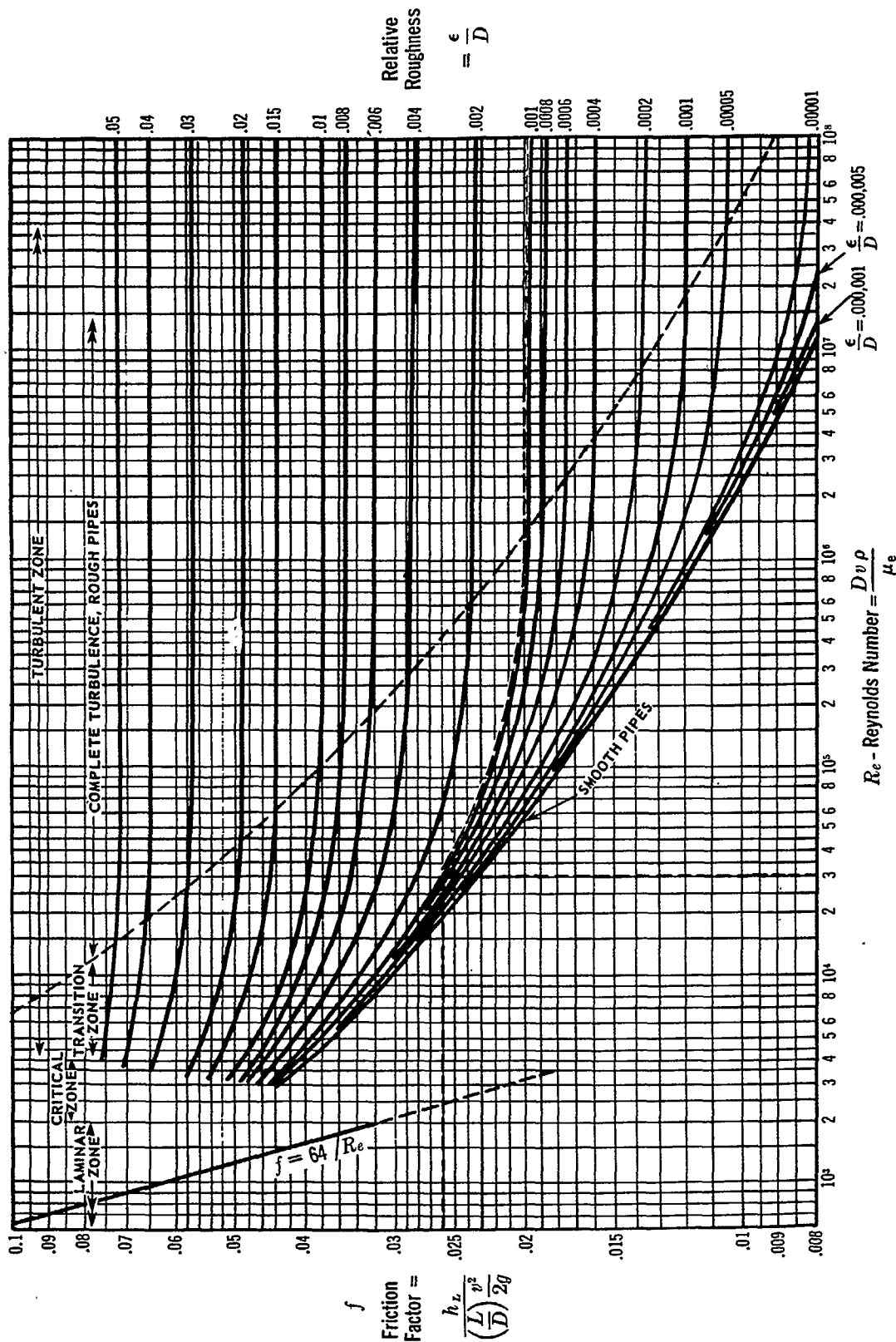
Relative Roughness of Pipe Materials and Friction Factors For Complete Turbulence¹⁸



Data extracted from *Friction Factors for Pipe Flow* by L.F. Moody, with permission of the publisher, The American Society of Mechanical Engineers, 29 West 39th Street, New York.

Problem: Determine absolute and relative roughness, and friction factor, for fully turbulent flow in 10-inch cast iron pipe (I.D. = 10.16").

Solution: Absolute roughness (ϵ) = 0.00085.... Relative roughness (ϵ/D) = 0.001... Friction factor at fully turbulent flow (f) = 0.0196.

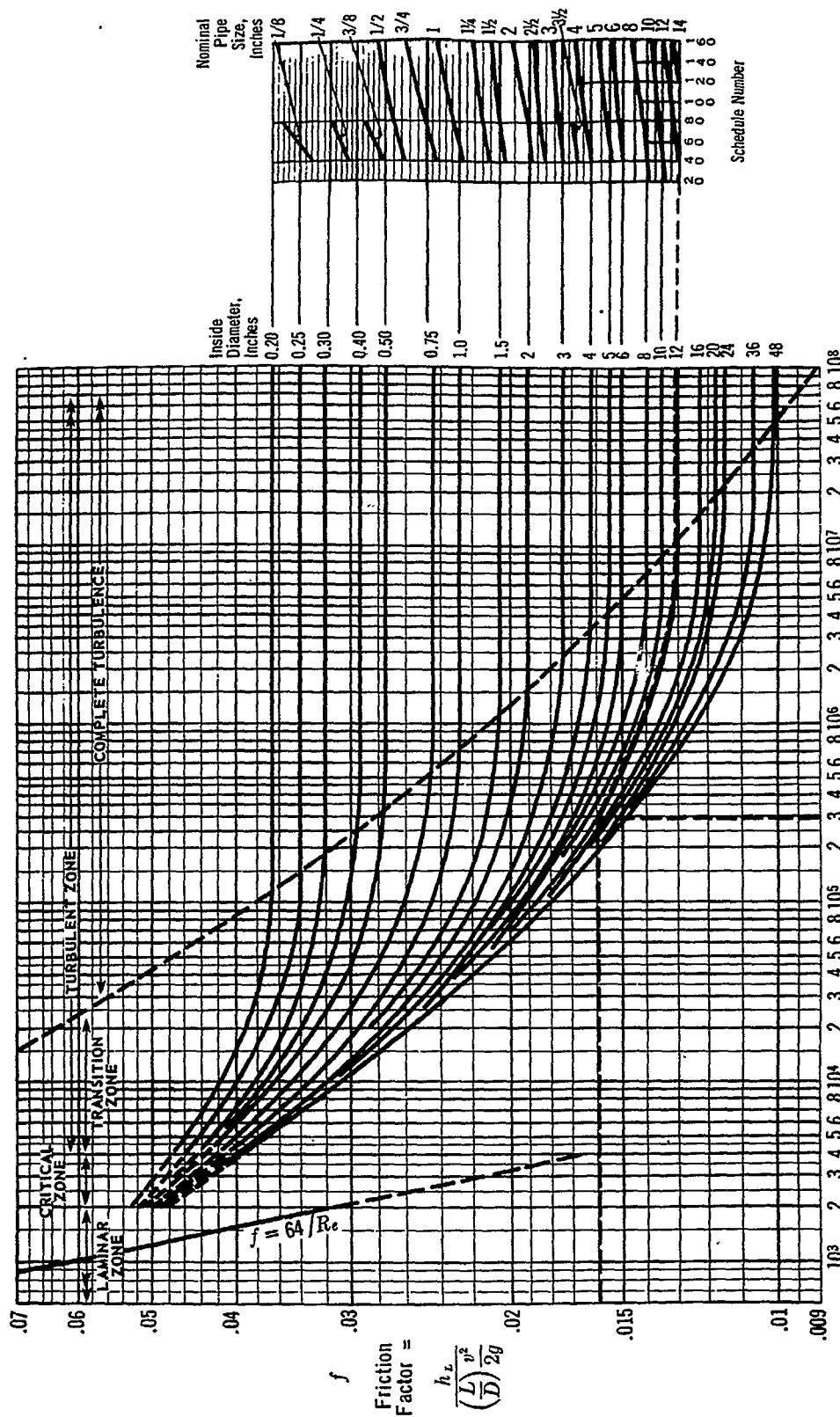
Friction Factors for Any Type of Commercial Pipe¹⁸**Problem:**

Determine the friction factor for 10-inch cast iron pipe (10.16" I.D.) at a Reynolds number flow of 30,000.

Solution: The relative roughness (see page A-23) is 0.001. Then, the friction factor (f) equals 0.026.

For other forms of the R_e equation, see page 3-2.

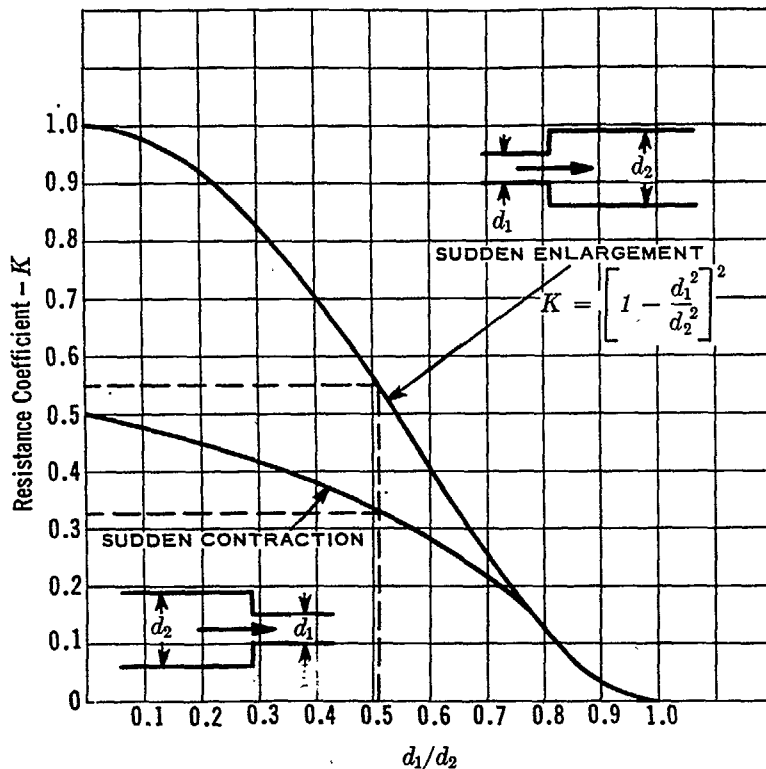
Data extracted from *Friction Factors for Pipe Flow* by L. F. Moody, with permission of the publisher, The American Society of Mechanical Engineers, 29 West 39th Street, New York 18, N. Y.

Friction Factors for Clean Commercial Steel and Wrought Iron Pipe¹⁸

Problem: Determine the friction factor for 12-inch Schedule 40 pipe at a flow having a Reynolds number of 300,000.

Solution: The friction factor (f) equals 0.016.

Resistance in Pipe

Resistance Due to Sudden Enlargements and Contractions²⁰

Sudden enlargement: The resistance coefficient K for a sudden enlargement from 6-inch Schedule 40 pipe to 12-inch Schedule 40 pipe is 0.55, based on the 6-inch pipe size.

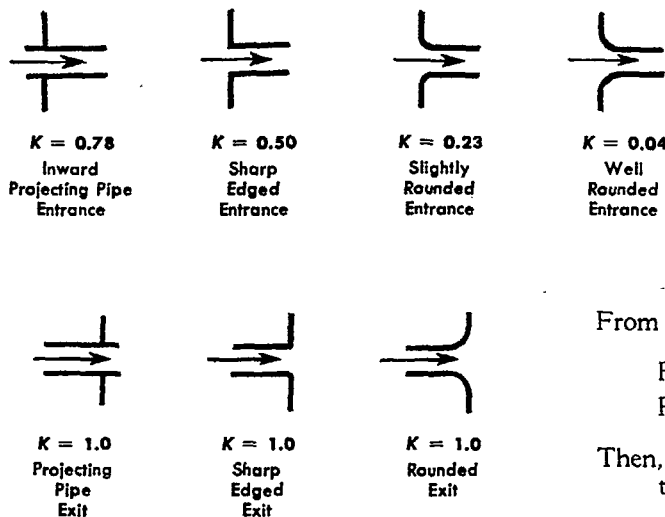
$$\frac{d_1}{d_2} = \frac{6.065}{11.938} = 0.51$$

Sudden contraction: The resistance coefficient K for a sudden contraction from 12-inch Schedule 40 pipe to 6-inch Schedule 40 pipe is 0.33, based on the 6-inch pipe size.

$$\frac{d_1}{d_2} = \frac{6.065}{11.938} = 0.51$$

Note: The values for the resistance coefficient, K , are based on velocity in the small pipe. To determine K values in terms of the greater diameter, multiply the chart values by $(d_2/d_1)^4$.

Resistance Due to Pipe Entrance and Exit



Problem: Determine the total resistance coefficient for a pipe one diameter long having a sharp edged entrance and a sharp edged exit.

Solution: The resistance of pipe one diameter long is small and can be neglected ($K = fL/D$).

From the diagrams, note:

Resistance for a sharp edged entrance = 0.5

Resistance for a sharp edged exit = 1.0

Then, the total resistance, K , for the pipe = 1.5

Resistance of Bends

Resistance of 90 Degree Bends²¹

The chart at the right shows the resistance of 90 degree bends to the flow of fluids in terms of equivalent lengths of straight pipe.

Resistance of bends greater than 90 degrees is found using the formula:

$$\frac{L}{D} = R_t + (n - 1) \left(R_t + \frac{R_b}{2} \right)$$

n = total number of 90° bends in coil

R_t = total resistance due to one 90° bend, in L/D

R_l = resistance due to length of one 90° bend, in L/D

R_b = bend resistance due to one 90° bend, in L/D

Problem: Determine the equivalent lengths in pipe diameters of a 90 degree bend and a 270 degree bend having a relative radius of 12.

Solution: Referring to the "Total Resistance" curve, the equivalent length for a 90 degree bend is 34.5 pipe diameters.

The equivalent length of a 270 degree bend is:

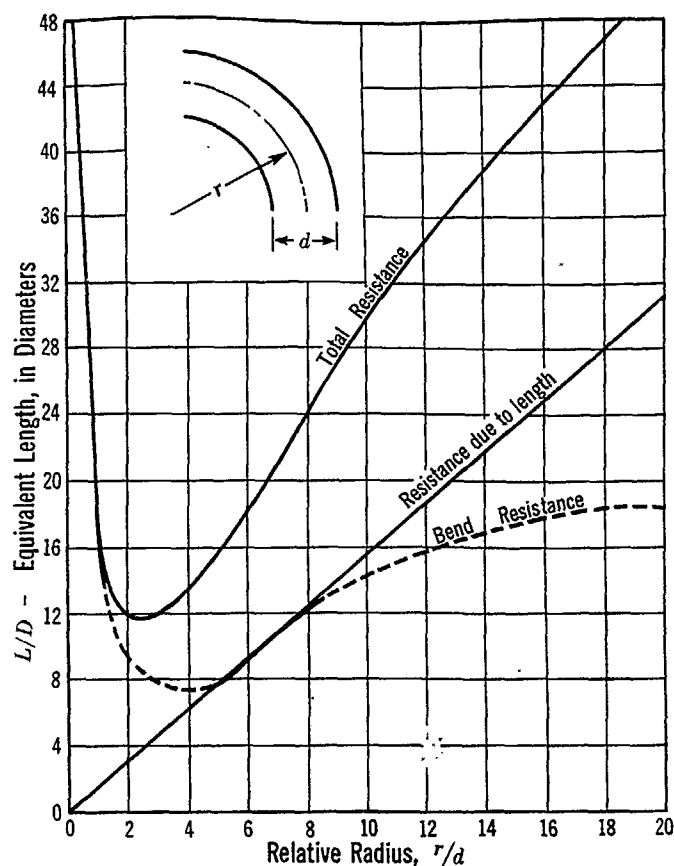
$$L/D = 34.5 + (3 - 1) [18.7 + (15.8 \div 2)]$$

$$L/D = 87.7 \text{ pipe diameters}$$

Note: This loss is less than the sum of losses through three 90 degree bends separated by tangents. For "resistance of bends theory", see page 2-12.

Chart for Resistance of 90 Degree Bends

From *Pressure Losses for Fluid Flow in 90 Degree Pipe Bends* by K. H. Beij. Courtesy of *Journal of Research of National Bureau of Standards*, Vol. 21, July, 1938.

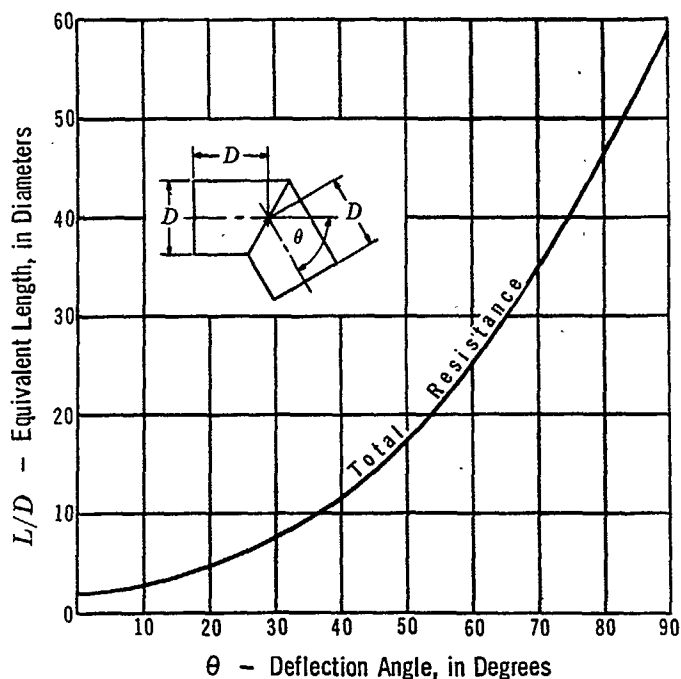
Resistance of Miter Bends⁴

The chart at the lower right shows the resistance of miter bends to the flow of fluids. The chart is based on data published by the American Society of Mechanical Engineers (ASME).

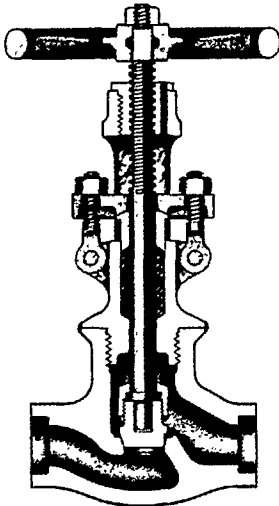
Problem: Determine the equivalent length in pipe diameters of a 40 degree miter bend.

Solution: Referring to the "Total Resistance" curve in the chart, the equivalent length is 12 pipe diameters.

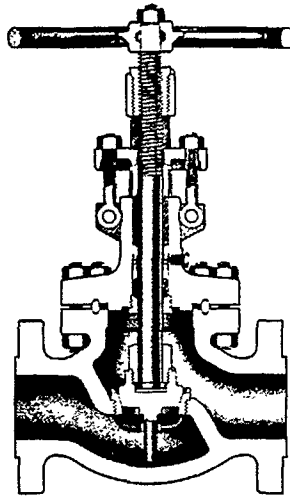
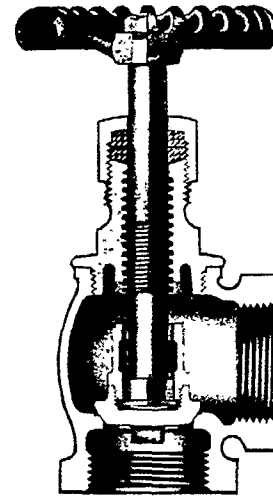
Chart for Resistance of Miter Bends



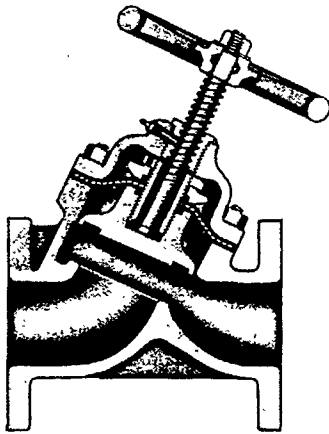
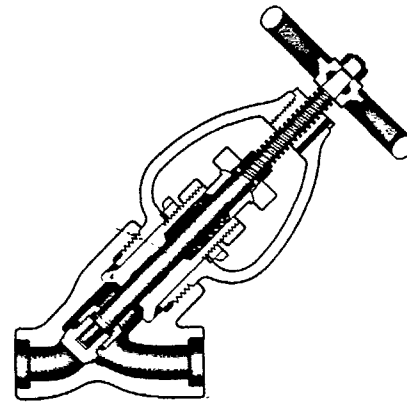
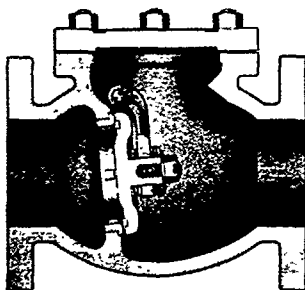
Types of Valves



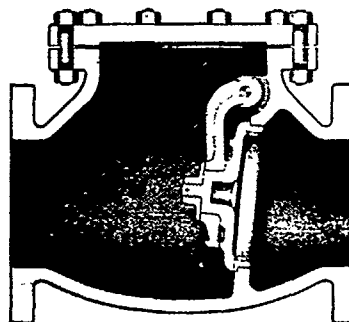
Conventional Globe Valve

Conventional Globe Valve
With Disc Guide

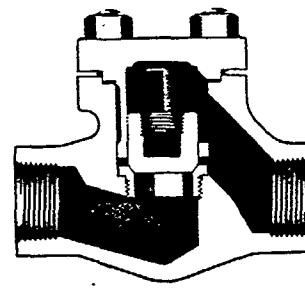
Conventional Angle Valve

Y-Pattern Globe Valve
With Stem 60 degrees from RunY-Pattern Globe Valve
With Stem 45 degrees from Run

Conventional Swing Check Valve

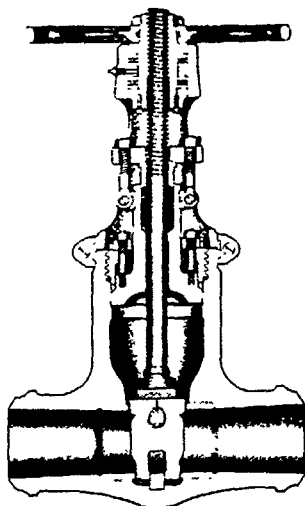


Clearway Swing Check Valve

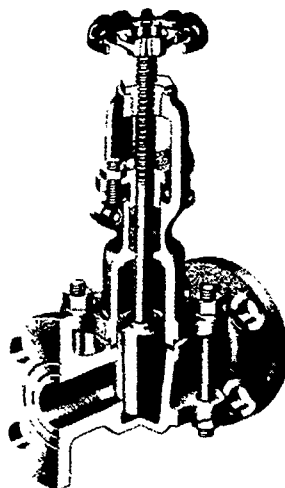


Globe Type Lift Check Valve

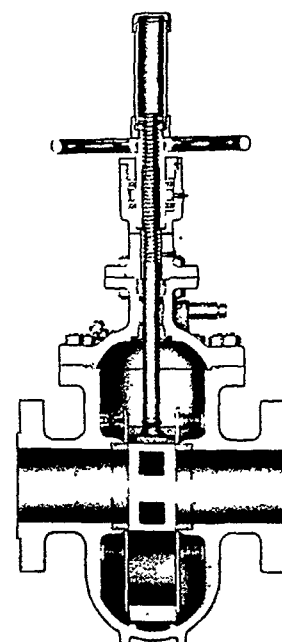
Types of Valves



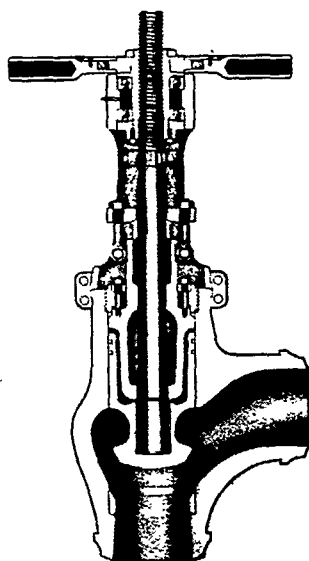
Conventional Gate Valve



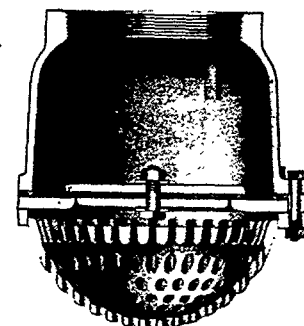
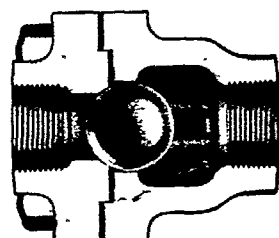
Plug Gate Valve



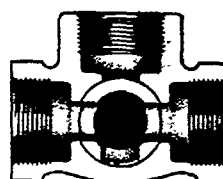
Pipe Line Gate Valve



Angle Stop-Check Valve

Foot Valve
With Strainer
Poppet TypeFoot Valve
With Strainer
Hinged Type

In-Line Ball Check Valve

Three-Way Cock
Sectional and Outside Views

Schedule (Thickness) of Steel Pipe Used in Obtaining Resistance Of Valves and Fittings of Various Pressure Classes by Test*

Valve or Fitting ASA Pressure Classification (Steam Rating)		Schedule No. of Pipe (Thickness)
250-Pound and Lower		Schedule 40
300-Pound to 600-Pound		Schedule 80
900-Pound		Schedule 120
1500-Pound		Schedule 160
2500-Pound	Sizes ½ to 6-inch Sizes 8-inch and larger	xx (Double Extra Strong) Schedule 160

*These schedule numbers have been arbitrarily selected only for the purpose of identifying the various pressure classes of valves and fittings with specific pipe dimensions for the interpretation of flow test data; they should not be construed as a recommendation for installation purposes.

Representative Equivalent Length[†] in Pipe Diameters (L/D) Of Various Valves and Fittings

Description of Product				Equivalent Length In Pipe Diameters (L/D)	
Globe Valves	Conventional	With no obstruction in flat, bevel, or plug type seat	Fully open	340	
		With wing or pin guided disc	Fully open	450	
	Y-Pattern	(No obstruction in flat, bevel, or plug type seat)			
		- With stem 60 degrees from run of pipe line	Fully open	175	
Angle Valves	Conventional	With no obstruction in flat, bevel, or plug type seat	Fully open	145	
		With wing or pin guided disc	Fully open	200	
	Gate Valves	Conventional Wedge Disc, Double Disc, or Plug Disc		Fully open	13
				Three-quarters open	35
			One-half open	160	
			One-quarter open	900	
Pulp Stock			Fully open	17	
			Three-quarters open	50	
			One-half open	260	
			One-quarter open	1200	
Conduit Pipe Line		Fully open	3**		
Check Valves	Conventional Swing	0.5†... Fully open	135		
	Clearway Swing	0.5†... Fully open	50		
	Globe Lift or Stop	2.0†... Fully open	Same as Globe		
	Angle Lift or Stop	2.0†... Fully open	Same as Angle		
	In-Line Ball	2.5 vertical and 0.25 horizontal†... Fully open	150		
	Foot Valves with Strainer	With poppet lift-type disc	0.3†... Fully open	420	
	With leather-hinged disc	0.4†... Fully open	75		
Butterfly Valves (6-inch and larger)			Fully open	20	
Cocks	Straight-Through	Rectangular plug port area equal to 100% of pipe area	Fully open	18	
	Three-Way	Rectangular plug port area equal to 80% of pipe area (fully open)	Flow straight through	44	
			Flow through branch	140	
Fittings	90 Degree Standard Elbow			30	
	45 Degree Standard Elbow			16	
	90 Degree Long Radius Elbow			20	
	90 Degree Street Elbow			50	
	45 Degree Street Elbow			26	
	Square Corner Elbow			57	
	Standard Tee	With flow through run		20	
		With flow through branch		60	
Close Pattern Return Bend			50		
Pipe	90 Degree Pipe Bends			See Page A-27	
	Miter Bends			See Page A-27	
	Sudden Enlargements and Contractions			See Page A-26	
	Entrance and Exit Losses			See Page A-26	

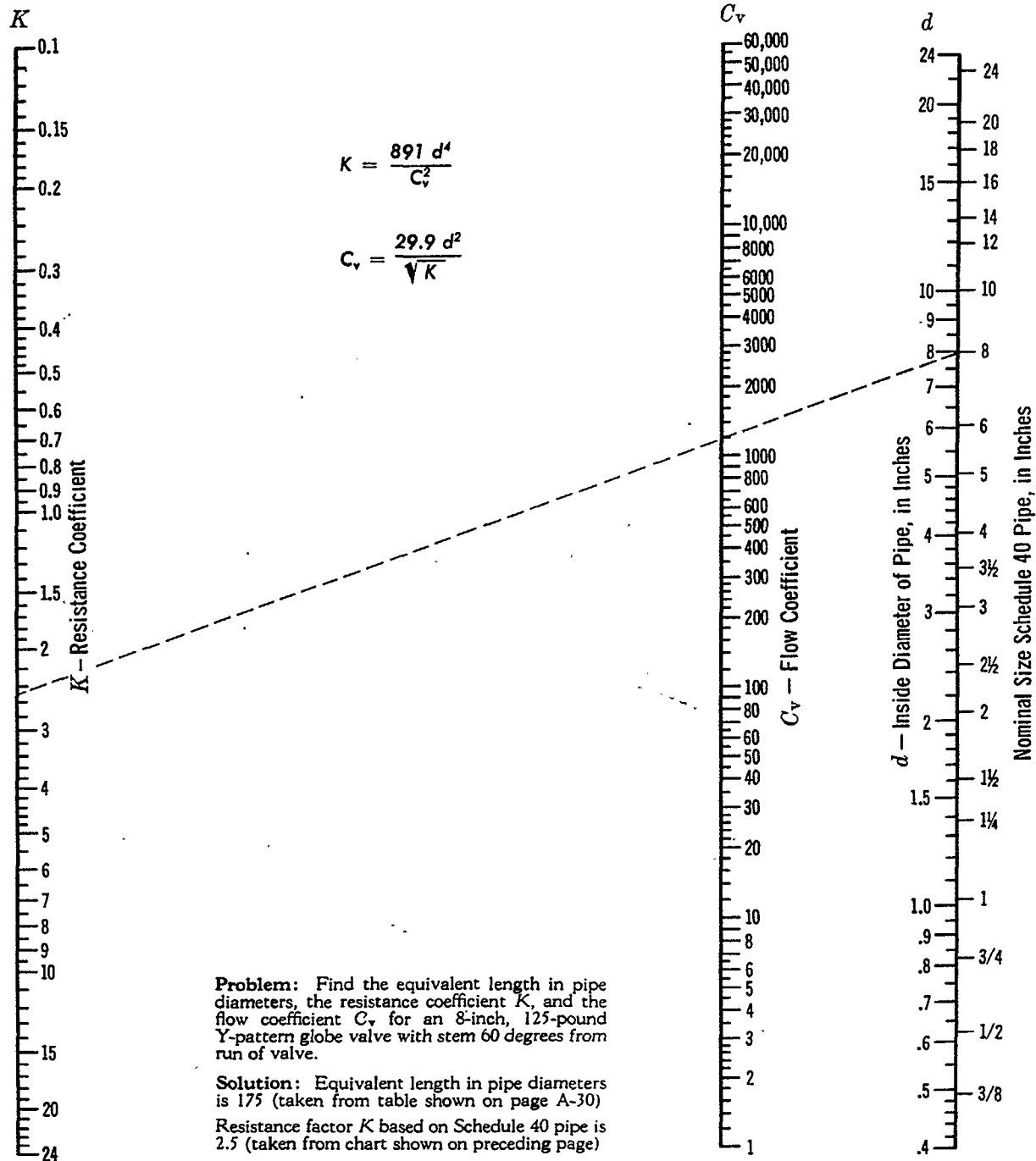
**Exact equivalent length is equal to the length between flange faces or welding ends.

†Minimum calculated pressure drop (psi) across valve to provide sufficient flow to lift disc fully.

‡For limitations, see page 2-11. For effect of end connections, see page 2-10.

For resistance factor "K", equivalent length in feet of pipe, and equivalent flow coefficient "Cv", see pages A-31 and A-32.

*Equivalents of Resistance Coefficient K And Flow Coefficient C_v



*For limitations, see page 2-11.

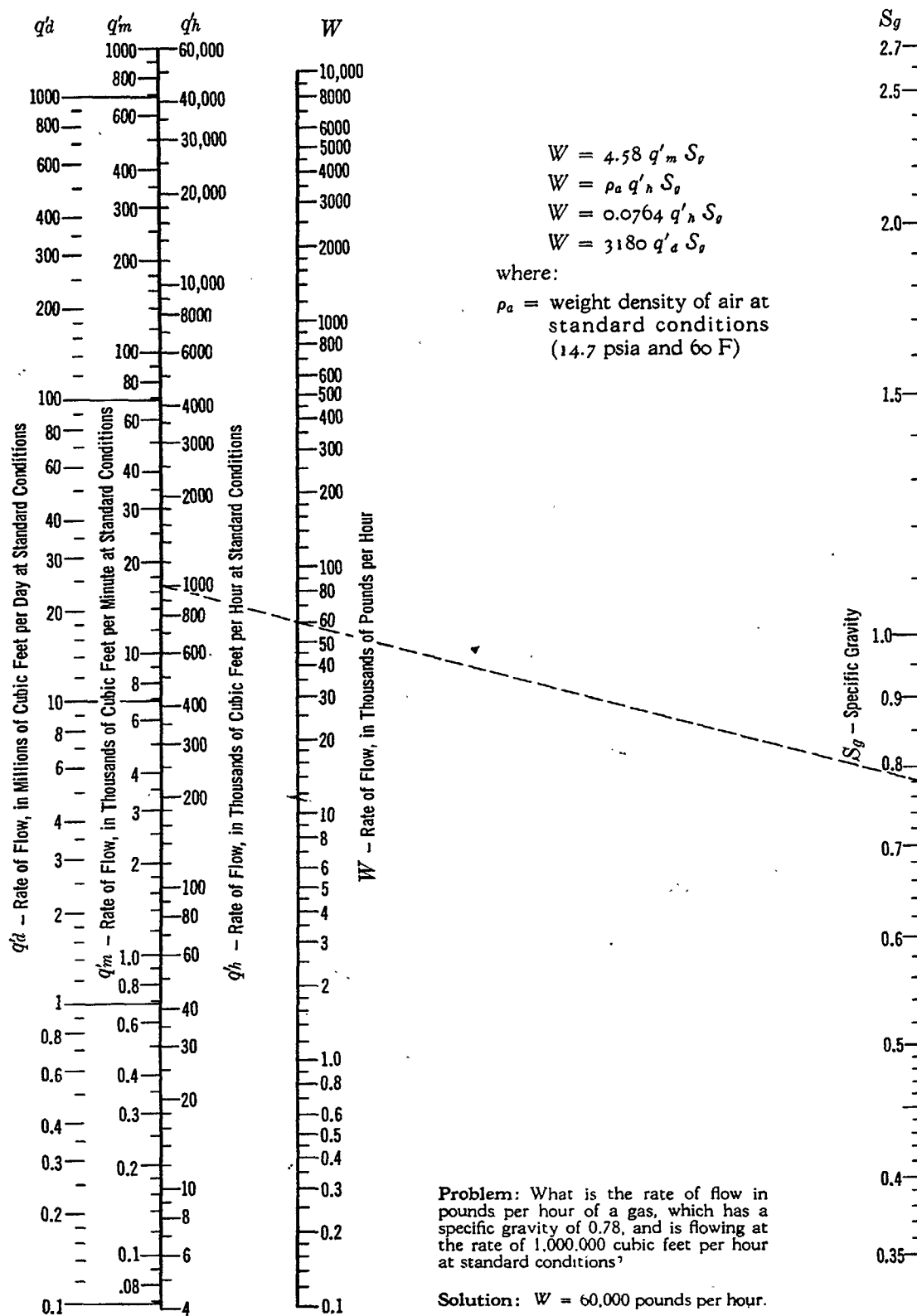
Engineering Data

APPENDIX B

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 and nomographs 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.

Equivalent Volume and Weight Flow Rates of Compressible Fluids



Equivalents of Absolute Viscosity

Absolute or Dynamic Viscosity		Centipoise	Poise	Slugs Ft Sec	†Pound _m Ft Sec
			$\frac{\text{Gram}}{\text{Cm Sec}}$ $\frac{\text{Dyne Sec}}{\text{Cm}^2}$	$\frac{*Pound_f \text{ Sec}}{\text{Ft}^2}$	$\frac{\text{Poundal Sec}}{\text{Ft}^2}$
		(μ)	(100 μ)	(μ'_s)	(μ_s)
Centipoise	(μ)	1	0.01	$2.09 (10^{-5})$	$6.72 (10^{-4})$
Poise $\frac{\text{Gram}}{\text{Cm Sec}}$ $\frac{\text{Dyne Sec}}{\text{Cm}^2}$	(100 μ)	100	1	$2.09 (10^{-3})$	0.0672
Slugs Ft Sec $\frac{*Pound_f \text{ Sec}}{\text{Ft}^2}$	(μ'_s)	47 900	479	1	g or 32.2
†Pound _m Ft Sec $\frac{\text{Poundal Sec}}{\text{Ft}^2}$	(μ_s)	1487	14.87	$\frac{1}{g}$ or .0311	1

*Pound_f = Pound of Force†Pound_m = Pound of Mass

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 $2.09 (10^{-3})$. Then, 2 (poise) times $2.09 (10^{-3}) = 4.18 (10^{-3}) = 0.00418$ slugs/foot second.

Equivalents of Kinematic Viscosity

Kinematic Viscosity		Centistokes	Stokes	$\frac{\text{Ft}^2}{\text{Sec}}$
		(ν)	$\frac{\text{Cm}^2}{\text{Sec}}$ (100 ν)	(ν')
Centistokes	(ν)	1	0.01	$1.076 (10^{-5})$
Stokes $\frac{\text{Cm}^2}{\text{Sec}}$	(100 ν)	100	1	$1.076 (10^{-3})$
$\frac{\text{Ft}^2}{\text{Sec}}$	(ν')	92 900	929	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 92,900. Then, 0.5 (sq ft/sec) times 92,900 = 46,450 centistokes.

For conversion from kinematic to absolute viscosity, see page B-5.

Equivalents of Kinematic and Saybolt Universal Viscosity

Kinematic Viscosity	Equivalent Saybolt Universal Viscosity, Sec		Kinematic Viscosity	Equivalent Saybolt Universal Viscosity, Sec	
	At 100 F Basic Values	At 210 F		At 100 F Basic Values	At 210 F
Centistokes ν			Centistokes ν		
2.0	32.6	32.9	29	136.9	137.9
2.5	34.4	34.7	30	141.3	142.3
3.0	36.0	36.3	31	145.7	146.8
3.5	37.6	37.9	32	150.2	151.2
4.0	39.1	39.4	33	154.7	155.8
4.5	40.8	41.0	34	159.2	160.3
5	42.4	42.7	35	163.7	164.9
6	45.6	45.9	36	168.2	169.4
7	48.8	49.1	37	172.7	173.9
8	52.1	52.5	38	177.3	178.5
9	55.5	55.9	39	181.8	183.0
10	58.9	59.3	40	186.3	187.6
11	62.4	62.9	41	190.8	192.1
12	66.0	66.5	42	195.3	196.7
13	69.8	70.3	43	199.8	201.2
14	73.6	74.1	44	204.4	205.9
15	77.4	77.9	45	209.1	210.5
16	81.3	81.9	46	213.7	215.2
17	85.3	85.9	47	218.3	219.8
18	89.4	90.1	48	222.9	224.5
19	93.6	94.2	49	227.5	229.1
20	97.8	98.5	50	232.1	233.8
21	102.0	102.8	55	255.2	257.0
22	106.4	107.1	60	278.3	280.2
23	110.7	111.4	65	301.4	303.5
24	115.0	115.8	70	324.4	326.7
25	119.3	120.1	Over 70	Saybolt Seconds equal	
26	123.7	124.5		Centistokes	
27	128.1	129.0		$\times 4.635$ $\times 4.667$	
28	132.5	133.4			

Note: To obtain the Saybolt Universal viscosity, equivalent to a kinematic viscosity determined at t , multiply the equivalent Saybolt Universal viscosity at 100 F by $1 + (t - 100) 0.000064$.

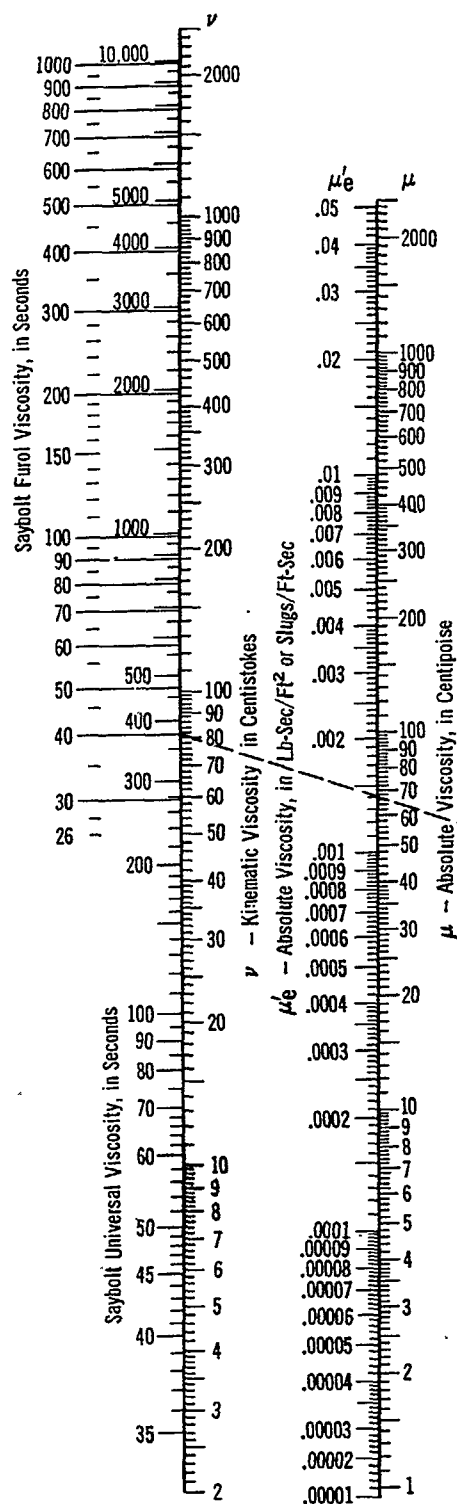
For example, 10 ν at 210 F are equivalent to 58.9 multiplied by 1.0070 or 59.3 sec Saybolt Universal at 210 F.

Equivalents of Kinematic and Saybolt Furoi Viscosity at 122 F

Kinematic Viscosity	Equivalent Saybolt Furoi Viscosity, Sec	Kinematic Viscosity	Equivalent Saybolt Furoi Viscosity, Sec
Centistokes ν		Centistokes ν	
48	25.3	150	71.7
50	26.1	155	74.0
52	27.0	160	76.3
54	27.9	165	78.7
56	28.8	170	81.0
58	29.7	175	83.3
60	30.6	180	85.6
62	31.5	185	88.0
64	32.4	190	90.3
66	33.3	195	92.6
68	34.2	200	95.0
70	35.1	210	99.7
72	36.0	220	104.3
74	36.9	230	109.0
76	37.8	240	113.7
78	38.7	250	118.4
80	39.6	260	123.0
82	40.5	270	127.7
84	41.4	280	132.4
86	42.3	290	137.1
88	43.2	300	141.8
90	44.1	310	146.5
92	45.0	320	151.2
94	45.9	330	155.9
96	46.8	340	160.6
98	47.7	350	165.3
100	48.6	360	170.0
105	50.9	370	174.7
110	53.2	380	179.4
115	55.5	390	184.1
120	57.8	400	188.8
125	60.1	Over 400	Saybolt Furoi Seconds = Centistokes $\times 0.4717$
130	62.4		
135	64.7		
140	67.0		
145	69.4		

These tables are reprinted with the permission of the American Society for Testing Materials (ASTM). The table at the left was abstracted from D446-53, as shown on page 226 of ASTM Standards, Part 5, 1955 edition. The table at the right was abstracted from D 666-53, as shown on page 302 of ASTM Standards, Part 5, 1955 edition.

Equivalents of Kinematic, Saybolt Universal, Saybolt Furol, and Absolute Viscosity



$$\mu = \nu S$$

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. D446-53 and D666-53. At other temperatures, the Saybolt Viscosities vary only slightly.

Saybolt Viscosities above those shown are given by the relationships:

$$\text{Saybolt Universal Seconds} = \text{Centistokes} \times 4.635$$

$$\text{Saybolt Furol Seconds} = \text{Centistokes} \times 0.470$$

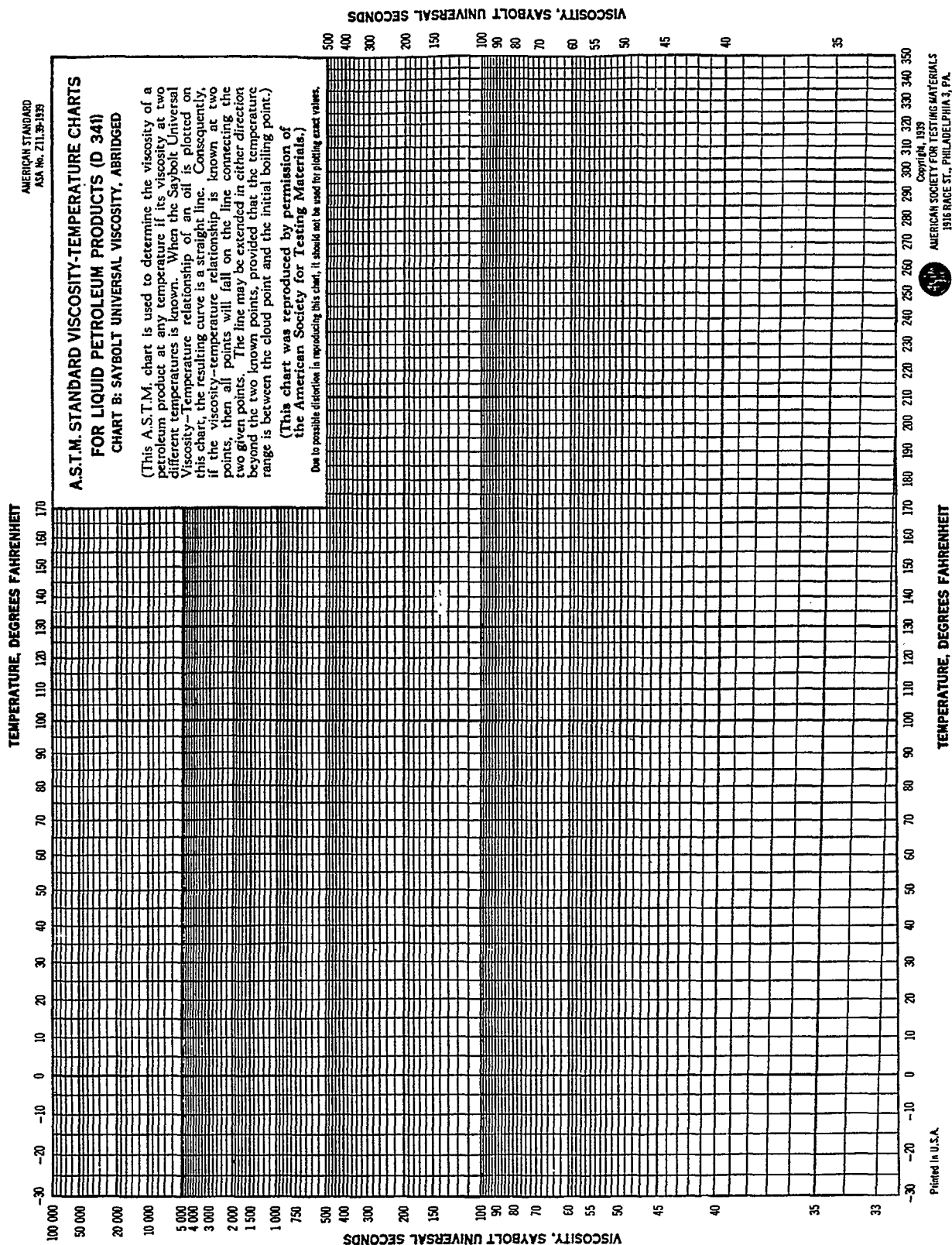
Problem 1: Determine the absolute viscosity of an oil which has a kinematic viscosity of 82 centistokes and a specific gravity of 0.83.

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.

Saybolt Universal Viscosity Chart



**Equivalents of Degrees API, Degrees Baumé,
Specific Gravity, Weight Density, and
Pounds Per Gallon at 60F/60F**

Degrees on API or Baumé Scale	Values for API Scale			Values for Baumé Scale					
	Oil			Liquids Lighter Than Water			Liquids Heavier Than Water		
	Specific Gravity	Weight Density, Lb/Ft ³	Pounds per Gallon	Specific Gravity	Weight Density, Lb/Ft ³	Pounds per Gallon	Specific Gravity	Weight Density, Lb/Ft ³	Pounds per Gallon
	<i>S</i>	<i>ρ</i>		<i>S</i>	<i>ρ</i>		<i>S</i>	<i>ρ</i>	
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	0.9279	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.

Steam Data

Boiler Capacity

The output of a steam generating plant is often expressed in pounds of steam delivered per hour. Since the steam output may vary in temperature and pressure, the boiler capacity is more completely expressed as the heat transferred in Btu per hour. Boiler capacity is usually expressed as kilo Btu (kB)/hour which is 1000 Btu/hour, or mega Btu (mB)/hour which is 1,000,000 Btu/hour. The boiler capacity is:

$$\frac{W(h_g - h_f)}{1000} \text{ in kilo Btu/hour}$$

$$h_g - h_f = \text{change in enthalpy, Btu/lb}$$

An older expression of boiler capacity in terms of an irrational unit called "boiler horsepower" may be expressed:

$$\frac{W(h_g - h_f)}{970.3 \times 34.5} = 33570$$

That is, one boiler horsepower is equivalent to 34.5 pounds of water evaporated per hour at Standard Atmospheric Pressure and a temperature of 212 F.

$$1 \text{ boiler horsepower} = \text{horsepower}/13.1547$$

$$\text{horsepower} = 550 \text{ ft-lb/sec.}$$

$$1 \text{ Btu} = 778.2 \text{ ft-lb} \quad 1 \text{ Btu} = 252 \text{ calories}$$

$$1 \text{ kw-hr} = 3412.20 \text{ Btu}$$

Horsepower of an Engine

P = Mean effective pressure per square inch of the steam on the piston

L = Length of stroke, in feet

A = Area of piston, in square inches

N = Number of strokes per minute

then,

$$\text{Horsepower} = \frac{PLAN}{33000}$$

The approximate mean effective pressure in the cylinder when the valve cuts off at:

$\frac{1}{4}$ stroke, equals steam pressure $\times .597$

$\frac{1}{3}$ stroke, equals steam pressure $\times .670$

$\frac{3}{8}$ stroke, equals steam pressure $\times .743$

$\frac{1}{2}$ stroke, equals steam pressure $\times .847$

$\frac{5}{8}$ stroke, equals steam pressure $\times .919$

$\frac{2}{3}$ stroke, equals steam pressure $\times .937$

$\frac{3}{4}$ stroke, equals steam pressure $\times .966$

$\frac{7}{8}$ stroke, equals steam pressure $\times .992$

Ranges in Steam Consumption by Prime Movers (For Estimating Purposes)

Simple Non-Condensing Engines	29 to 45 pounds per H. P. hour
Simple Non-Condensing Automatic Engines	26 to 40 pounds per H. P. hour
Simple Non-Condensing Corliss Engines	26 to 35 pounds per H. P. hour
Compound Non-Condensing Engines	19 to 28 pounds per H. P. hour
Compound Condensing Engines	12 to 22 pounds per H. P. hour
Simple Duplex Steam Pumps	120 to 200 pounds per H. P. hour
Turbines, Non-Condensing	21 to 45 pounds per H. P. hour
Turbines, Condensing	9 to 32 pounds per H. P. hour

$$\text{Quality of Steam} \dots x = \frac{(h_g - h_f) 100}{h_{fg}}$$

where,

h_f = heat of liquid, in Btu/lb

h_{fg} = latent heat of evaporation, in Btu/lb

h_g = total heat of steam, in Btu/lb

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

Specific gravity of water.....page A-6
 Specific gravity of liquids other than water....page A-7

$$\begin{aligned}\text{HORSEPOWER} &= 33\,000 \dots \text{ft-lb/min} \\ &= 550 \dots \text{ft-lb/sec} \\ &= 2544.48 \dots \text{Btu/hr} \\ &= 745.7 \dots \text{watts}\end{aligned}$$

$$\begin{aligned}(whp) &= QH\rho \div 247\,000 = QP \div 1714 \\ (bhp) &= (whp) \div e_p = QH\rho \div 247\,000 e_p \\ (e_p) &= QH\rho \div 247\,000 (bhp)\end{aligned}$$

where: (whp) = water horsepower
 H = pump head in feet
 (bhp) = brake horsepower
 e_p = pump efficiency

Overall efficiency (e_o) takes into account all losses in the pump and driver.

$$e_o = e_p e_D e_T$$

where: e_D = driver efficiency
 e_T = transmission efficiency

$$e_v = \text{volumetric efficiency}$$

$$e_v(\%) = \frac{\text{actual pump displacement } (Q) (100)}{\text{theoretical pump displacement } (Q)}$$

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.

Equivalents

Measure

1 in.	= 25.4 mm
1 in.	= 2.54 cm
1 mm	= 0.03937 in.
1 mm	= 0.00328 ft
1 micron	= 0.000001 meter

1 ft	= 304.8 mm
1 ft	= 30.48 cm

1 sq in.	= 6.4516 sq cm
1 sq cm	= 0.155 sq in.
1 sq cm	= 0.00108 sq ft
1 sq ft	= 929.03 sq cm

Circumference
of a circle = $2\pi r = \pi d$

Area of a circle = $\pi r^2 = \frac{\pi d^2}{4}$

Weight

1 kg	= 2.205 lb
1 cu in. of water (60 F)	= 0.073551 cu in. of mercury (32 F)
1 cu in. of mercury (32 F)	= 13.596 cu in. of water (60 F)
1 cu in. of mercury (32 F)	= 0.4905 lb

Velocity

1 ft per sec	= 0.3048 m per sec
1 m per sec	= 3.2808 ft per sec

Density

1 lb per cu in.	= 27.68 gram per cu cm
1 gr per cu cm	= 0.03613 lb per cu in.
1 lb per cu ft	= 16.0184 kg per cu m
1 kg per cu m	= 0.06243 lb per cu ft

Physical Constants

Base of Natural Logarithms (e)	2.7182818285
Acceleration of Gravity (g)	32.174 ft/sec ² (980.665 cm/sec ²)
Pi (π)	3.1415926536

	Degrees Kelvin	Degrees Rankine	Degrees Centigrade	Degrees Fahrenheit
Absolute Zero	0	0	- 273.16	- 459.69
Water Freezing Point (14.696 psia)	273.16	459.69	0	32
Water Boiling Point (14.696 psia)	373.16	671.69	100	212

Equivalents of Temperature

To convert degrees Centigrade to degrees Fahrenheit:

$$t = 1.8 t_c + 32$$

To convert degrees Fahrenheit to degrees Centigrade:

$$t_c = \frac{t - 32}{1.8}$$

Where: t_c = temperature, in degrees Centigrade

Prefixes

Micro	one-millionth	0.000 001	10^{-6}
Milli	one-thousandth	0.001	10^{-3}
Centi	one-hundredth	0.01	10^{-2}
Deci	one-tenth	0.1	10^{-1}
*Unity	one	1.0	10^0
Deka	ten	10.0	10^1
Hecto	one hundred	100.0	10^2
Kilo	one thousand	1000.0	10^3
Mega	one million	1 000 000.0	10^6

*Unity is not a prefix

Equivalents of Liquid Measures and Weights

Liquid Measure or Weight	U.S. Gallon	Imperial Gallon	U.S. Pint	U.S. Pound Water*	U.S. Cubic Foot	U.S. Cubic Inch	Liter	Cubic Meter
U.S. Gallon	1	0.833	8	8.337	0.13368	231	3.78533	0.003785
Imperial Gallon	1.2009	1	9.60752	10.	0.16054	277.42	4.54596	0.004546
U.S. Pint	0.125	0.1501	1	1.042	0.01671	28.875	0.473166	0.000473
U.S. Pound Water*	0.11995	0.09992	0.09596	1	0.016035	27.708	0.45405	0.000454
U.S. Cubic Foot	7.48052	6.22888	59.8442	62.365	1	1728	28.31702	0.028317
U.S. Cubic Inch	0.004329	0.00361	0.034632	0.03609	0.0005787	1	0.016387	0.0000164
Liter	0.2641779	0.2199756	2.113423	2.202	0.0353154	61.02509	1	0.001000
Cubic Meter	264.170	219.969	2113.34	2202.	35.31446	61023.38	999.972	1

*Water at 60 F.

1 Barrel = 42 gallons (petroleum measure)

Problem:

How many British Imperial Gallons are equivalent to 10 U.S. Gallons?

Solution:The Imperial Gallon equivalent of 1 U.S. Gallon is 0.833. Then, $10 \times 0.833 = 8.33$ Imperial Gallons.

Equivalents of Pressure and Head

Pressure or Head	lb/in. ²	lb/ft ²	Atmos- pheres	kg/cm ²	kg/m ²	in. water (60 F)	ft water (60 F)	in. mercury (32 F)	mm mercury (32 F)
lb/in. ²	1	144	0.068046	0.070307	703.067	27.707	2.3039	2.03601	51.7148
lb/ft ²	0.0069445	1	0.000473	0.000488	4.88241	0.19241	0.01603	0.014139	0.35913
Atmospheres	14.696	2116.22	1	1.0332	10332.27	407.17	33.931	29.921	760.
kg/cm ²	14.2234	2048.17	0.96784	1	10000.	394.08	32.840	28.959	735.559
kg/m ²	0.001422	0.204817	0.0000968	0.0001	1	0.03941	0.003284	0.002896	0.073556
in. water*	0.036092	5.1972	0.002456	0.00253	25.375	1	0.08333	0.073483	1.8665
ft water*	0.433103	62.3668	0.029471	0.03045	304.50	12.	1	0.88180	22.3980
in. mercury†	0.491157	70.7266	0.033421	0.03453	345.316	13.608	1.1340	1	25.40005
mm mercury†	0.0193368	2.78450	0.0013158	0.0013595	13.59509	0.535764	0.044647	0.03937	1

*Water at 60 F.

†Mercury at 32 F.

Problem:

How many inches of mercury are equivalent to 10 inches of water?

Solution:1 inch of water is equal to 0.073483 inch mercury. Then, $10 \times 0.073483 = 0.73483$ inch mercury.

To convert from one set of units to another, locate the given unit in the left hand column, and multiply the numerical value by the factor shown horizontally to the right, under the set of units desired.

Four-Place Logarithms to Base 10

The logarithms to base 10 of numbers between 1 and 10, correct to four places, are given in the tables shown on this and the following page.

If the decimal point in the number is moved n places to the right (or left), the value of n (or $-n$) is added to the logarithm, thus:

$$\begin{aligned}\log 3.14 &= 0.4969 \\ \log 314 &= 0.4969 + 2 \text{ or } 2.4969 \\ \log .0314 &= 0.4969 - 2, \text{ which may be written } \bar{2}.4969 \\ &\text{ or } 8.4969 - 10\end{aligned}$$

If the given number has more than four significant figures, it should be reduced to four figures, since those beyond four figures will not affect the result in four-place computations.

The logarithm of a number having four significant figures must be interpolated by adding to the logarithm of the three figure number, the amount under

the fourth figure, as read in the proportional parts section of the table.

Thus, the logarithm of 3.1416 is found as follows:

- Reduce the number to four significant figures: 3.142
- The log of 3.14 is .4969
- The value of the proportional part under 2 (the fourth figure) is 3
- Then, the log 3.142 = 0.4969 + .0003 or 0.4972

Natural logarithms: Many calculations make use of natural logarithms (Base $e = 2.7183$). To convert base 10 (common) logarithms to natural logarithms, multiply the value for the former by 2.30258.

Natural logarithms are also called Hyperbolic or Napierian logarithms.

$$\log ab = \log a + \log b$$

$$\log a^n = n \log a$$

$$\log \frac{a}{b} = \log a - \log b$$

$$\log \sqrt[n]{a} = \frac{\log a}{n}$$

N	0	1	2	3	4	5	6	7	8	9	Proportional Parts								
											1	2	3	4	5	6	7	8	9
1.0	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	4	8	12	17	21	25	29	33	37
1.1	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	4	8	11	15	19	23	26	30	34
1.2	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	3	7	10	14	17	21	24	28	31
1.3	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	3	6	10	13	16	19	23	26	29
1.4	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	3	6	9	12	15	18	21	24	27
1.5	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	3	6	8	11	14	17	20	22	25
1.6	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	3	5	8	11	13	16	18	21	24
1.7	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	2	5	7	10	12	15	17	20	22
1.8	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	2	5	7	9	12	14	16	19	21
1.9	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	2	4	7	9	11	13	16	18	20
2.0	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	2	4	6	8	11	13	15	17	19
2.1	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	2	4	6	8	10	12	14	16	18
2.2	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	2	4	6	8	10	12	14	15	17
2.3	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	2	4	6	7	9	11	13	15	17
2.4	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	2	4	5	7	9	11	12	14	16
2.5	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	2	3	5	7	9	10	12	14	15
2.6	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	2	3	5	7	8	10	11	13	15
2.7	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	2	3	5	6	8	9	11	13	14
2.8	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	2	3	5	6	8	9	11	12	14
2.9	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	1	3	4	6	7	9	10	12	13
3.0	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	1	3	4	6	7	9	10	11	13
3.1	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	1	3	4	6	7	8	10	11	12
3.2	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	1	3	4	5	7	8	9	11	12
3.3	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	1	3	4	5	6	8	9	10	12
3.4	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	1	3	4	5	6	8	9	10	11
3.5	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	1	2	4	5	6	7	9	10	11
3.6	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	1	2	4	5	6	7	8	10	11
3.7	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	1	2	3	5	6	7	8	9	10
3.8	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	1	2	3	5	6	7	8	9	10
3.9	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	1	2	3	4	5	7	8	9	10
N	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9

Four-Place Logarithms to Base 10 — continued

N	0	1	2	3	4	5	6	7	8	9	Proportional Parts									
											1	2	3	4	5	6	7	8	9	
4.0	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	1	2	3	4	5	6	8	9	10	
4.1	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	1	2	3	4	5	6	7	8	9	
4.2	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	1	2	3	4	5	6	7	8	9	
4.3	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	1	2	3	4	5	6	7	8	9	
4.4	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	1	2	3	4	5	6	7	8	9	
4.5	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	1	2	3	4	5	6	7	8	9	
4.6	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	1	2	3	4	5	6	7	7	8	
4.7	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	1	2	3	4	5	5	6	7	8	
4.8	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	1	2	3	4	4	5	6	7	8	
4.9	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	1	2	3	4	4	5	6	7	8	
5.0	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	1	2	3	3	4	5	6	7	8	
5.1	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	1	2	3	3	4	5	6	7	8	
5.2	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	1	2	2	3	4	5	6	7	7	
5.3	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	1	2	2	3	4	5	6	6	7	
5.4	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	1	2	2	3	4	5	6	6	7	
5.5	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	1	2	2	3	4	5	5	6	7	
5.6	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	1	2	2	3	4	5	5	6	7	
5.7	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	1	2	2	3	4	5	5	6	7	
5.8	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	1	1	2	3	4	4	5	6	7	
5.9	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	1	1	2	3	4	4	5	6	7	
6.0	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	1	1	2	3	4	4	5	6	6	
6.1	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	1	1	2	3	4	4	5	6	6	
6.2	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	1	1	2	3	3	4	5	6	6	
6.3	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	1	1	2	3	3	4	5	5	6	
6.4	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	1	1	2	3	3	4	5	5	6	
6.5	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	1	1	2	3	3	4	5	5	6	
6.6	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	1	1	2	3	3	4	5	5	6	
6.7	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	1	1	2	3	3	4	5	5	6	
6.8	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	1	1	2	3	3	4	4	5	6	
6.9	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	1	1	2	2	3	4	4	5	6	
7.0	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	1	1	2	2	3	4	4	5	6	
7.1	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	1	1	2	2	3	4	4	5	5	
7.2	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	1	1	2	2	3	4	4	5	5	
7.3	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	1	1	2	2	3	4	4	5	5	
7.4	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	1	1	2	2	3	4	4	5	5	
7.5	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	1	1	2	2	3	3	4	5	5	
7.6	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	1	1	2	2	3	3	4	5	5	
7.7	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	1	1	2	2	3	3	4	4	5	
7.8	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	1	1	2	2	3	3	4	4	5	
7.9	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	1	1	2	2	3	3	4	4	5	
8.0	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	1	1	2	2	3	3	4	4	5	
8.1	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	1	1	2	2	3	3	4	4	5	
8.2	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	1	1	2	2	3	3	4	4	5	
8.3	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	1	1	2	2	3	3	4	4	5	
8.4	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	1	1	2	2	3	3	4	4	5	
8.5	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	1	1	2	2	3	3	4	4	5	
8.6	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	1	1	2	2	3	3	4	4	5	
8.7	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	0	1	1	2	2	3	3	4	4	
8.8	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	0	1	1	2	2	3	3	4	4	
8.9	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	0	1	1	2	2	3	3	4	4	
9.0	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	0	1	1	2	2	3	3	4	4	
9.1	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	0	1	1	2	2	3	3	4	4	
9.2	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	0	1	1	2	2	3	3	4	4	
9.3	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	0	1	1	2	2	3	3	4	4	
9.4	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	0	1	1	2	2	3	3	4	4	
9.5	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	0	1	1	2	2	3	3	4	4	
9.6	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	0	1	1	2	2	3	3	4	4	
9.7	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	0	1	1	2	2	3	3	4	4	
9.8	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	0	1	1	2	2	3	3	4	4	
9.9	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	0	1	1	2	2	3	3	4	4	
N	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9	

Flow of Water Through Schedule 40 Steel Pipe

Discharge		Pressure Drop per 100 feet and Velocity in Schedule 40 Pipe for Water at 60 F.																			
		Velocity Feet per Second		Press. Drop Lbs. per Sq. In.		Velocity Feet per Second		Press. Drop Lbs. per Sq. In.		Velocity Feet per Second		Press. Drop Lbs. per Sq. In.		Velocity Feet per Second		Press. Drop Lbs. per Sq. In.		Velocity Feet per Second		Press. Drop Lbs. per Sq. In.	
Gallons per Minute	Cubic Ft. per Second	1/8"		1/4"		3/8"		1/2"		3/4"		1"		1 1/4"		1 1/2"		2"		2 1/2"	
.2	0.000446	1.13	1.86	0.616	0.359	0.504	0.159	0.317	0.061	3/4"											
.3	0.000668	1.69	4.22	0.924	0.903	0.504	0.159	0.317	0.061	3/4"											
.4	0.000891	2.26	6.98	1.23	1.61	0.672	0.345	0.422	0.086	3/4"											
.5	0.00111	2.82	10.5	1.54	2.39	0.840	0.539	0.528	0.167	0.301	0.033										
.6	0.00134	3.39	14.7	1.85	3.29	1.01	0.751	0.633	0.240	0.361	0.041										
.8	0.00178	4.52	25.0	2.46	5.44	1.34	1.25	0.844	0.408	0.481	0.102	1"		1 1/4"		1 1/2"					
1	0.00223	5.65	37.2	3.08	8.28	1.68	1.85	1.06	0.600	0.602	0.155	0.371	0.048	0.429	0.044	0.473	0.043				
2	0.00446	11.29	134.4	6.16	30.1	3.36	6.58	2.11	2.10	1.20	0.526	0.743	0.164	0.644	0.090	0.630	0.071				
3	0.00668	9.25	64.1	9.25	64.1	5.04	13.9	3.17	4.33	1.81	1.09	1.114	0.336	0.858	0.150	0.788	0.104				
4	0.00891	12.33	111.2	12.33	111.2	6.72	23.9	4.22	7.42	2.41	1.83	1.49	0.565	1.073	0.237	0.907	0.145				
5	0.01114	15.4	167.0	15.4	167.0	8.40	36.7	5.28	11.2	3.01	2.75	1.86	0.835	1.353	0.273	1.128	0.158				
6	0.01337	18.5	222.8	18.5	222.8	10.08	51.9	6.33	15.8	3.61	3.84	2.23	1.17	1.29	0.309	0.946	0.145				
8	0.01782	24.6	299.6	24.6	299.6	13.44	91.1	8.45	27.7	4.81	6.60	2.97	1.99	1.72	0.518	1.26	0.241				
10	0.02228	30.7	376.4	30.7	376.4	16.8	128.0	10.56	42.4	6.02	9.99	3.71	2.99	2.15	0.774	1.58	0.361				
15	0.03442	45.8	544.0	45.8	544.0	25.0	199.0	15.4	79.0	9.03	21.6	5.57	6.36	3.22	1.63	2.37	0.755				
20	0.04456	56.5	698.0	56.5	698.0	31.6	271.0	19.6	105.0	12.03	37.8	7.43	10.9	4.29	2.78	3.16	1.28				
25	0.05570	67.2	852.0	67.2	852.0	38.0	356.0	23.9	134.0	14.8	48.0	9.28	16.7	5.37	4.22	3.94	1.93				
30	0.06684	77.9	1006.0	77.9	1006.0	45.0	450.0	29.4	171.0	17.4	59.0	11.14	23.8	6.44	5.92	4.73	2.72				
35	0.07798	88.6	1160.0	88.6	1160.0	52.0	552.0	35.4	208.0	20.0	69.0	12.99	32.2	7.51	7.30	5.32	3.64				
40	0.08912	99.3	1314.0	99.3	1314.0	60.0	600.0	42.0	240.0	22.6	81.0	14.85	41.5	8.59	10.24	6.30	4.65				
45	0.1003	110.0	1468.0	110.0	1468.0	68.0	704.0	49.0	279.0	25.2	93.0	16.7	46.0	9.67	12.80	7.09	5.85				
50	0.1114	120.7	1622.0	120.7	1622.0	77.0	824.0	56.0	316.0	27.8	104.0	18.7	51.0	10.74	15.66	7.88	7.15				
60	0.1337	146.0	1996.0	146.0	1996.0	92.0	984.0	67.0	376.0	32.8	121.0	21.9	59.0	12.89	22.2	9.47	10.21				
70	0.1560	171.3	2370.0	171.3	2370.0	108.0	1152.0	79.0	441.0	37.8	138.0	24.1	67.0	14.4	26.0	11.05	13.71				
80	0.1782	196.6	2744.0	196.6	2744.0	125.0	1320.0	91.0	508.0	43.8	154.0	26.1	75.0	16.0	30.0	12.62	17.59				
90	0.2005	221.9	3118.0	221.9	3118.0	143.0	1504.0	104.0	576.0	50.0	171.0	28.1	83.0	17.4	34.0	14.20	22.0				
100	0.2228	247.2	3492.0	247.2	3492.0	162.0	1692.0	117.0	648.0	57.0	189.0	30.1	91.0	18.7	38.0	15.78	26.9				
125	0.2785	314.0	4376.0	314.0	4376.0	200.0	2000.0	146.0	800.0	67.0	227.0	32.1	103.0	20.0	42.0	17.72	41.4				
150	0.3342	380.8	5260.0	380.8	5260.0	240.0	2400.0	177.0	950.0	78.0	255.0	34.1	111.0	21.6	46.0	19.07	48.0				
175	0.3899	447.6	6144.0	447.6	6144.0	280.0	2800.0	210.0	1100.0	90.0	297.0	36.1	121.0	23.2	50.0	20.72	53.0				
200	0.4456	514.4	7028.0	514.4	7028.0	320.0	3200.0	244.0	1280.0	103.0	339.0	38.1	131.0	24.8	54.0	22.0	58.0				
225	0.5013	581.2	7912.0	581.2	7912.0	360.0	3600.0	280.0	1440.0	117.0	387.0	40.1	141.0	26.0	58.0	23.0	63.0				
250	0.557	648.0	8796.0	648.0	8796.0	400.0	4000.0	320.0	1600.0	132.0	432.0	42.1	151.0	27.0	62.0	24.0	68.0				
275	0.6127	714.8	9680.0	714.8	9680.0	440.0	4400.0	360.0	1800.0	147.0	477.0	44.1	161.0	28.0	64.0	25.0	73.0				
300	0.6684	781.6	10564.0	781.6	10564.0	480.0	4800.0	400.0	2000.0	162.0	516.0	46.1	171.0	29.0	66.0	26.0	78.0				
325	0.7241	848.4	11448.0	848.4	11448.0	520.0	5200.0	440.0	2200.0	177.0	557.0	48.1	181.0	30.0	68.0	27.0	83.0				
350	0.7798	915.2	12332.0	915.2	12332.0	560.0	5600.0	480.0	2400.0	192.0	598.0	50.1	191.0	31.0	70.0	28.0	88.0				
375	0.8355	982.0	13216.0	982.0	13216.0	600.0	6000.0	520.0	2600.0	207.0	639.0	52.1	201.0	32.0	72.0	29.0	93.0				
400	0.8912	1048.8	14100.0	1048.8	14100.0	640.0	6400.0	560.0	2800.0	222.0	680.0	54.1	211.0	33.0	74.0	30.0	98.0				
425	0.9469	1115.6	14984.0	1115.6	14984.0	680.0	6800.0	600.0	3000.0	237.0	721.0	56.1	221.0	34.0	76.0	31.0	103.0				
450	1.003	1182.4	15868.0	1182.4	15868.0	720.0	7200.0	640.0	3200.0	252.0	762.0	58.1	231.0	35.0	78.0	32.0	108.0				
475	1.059	1249.2	16752.0	1249.2	16752.0	760.0	7600.0	680.0	3400.0	267.0	803.0	60.1	241.0	36.0	80.0	33.0	113.0				
500	1.114	1316.0	17636.0	1316.0	17636.0	800.0	8000.0	720.0	3600.0	282.0	844.0	62.1	251.0	37.0	82.0	34.0	118.0				
550	1.225	1444.0	19416.0	1444.0	19416.0	920.0	9200.0	840.0	4080.0	328.0	968.0	68.1	271.0	40.0	86.0	36.0	128.0				
600	1.337	1572.0	21196.0	1572.0	21196.0	1040.0	10400.0	960.0	4560.0	374.0	1092.0	74.1	291.0	42.0	88.0	38.0	138.0				
650	1.448	1700.0	22976.0	1700.0	22976.0	1160.0	11600.0	1080.0	5040.0	420.0	1216.0	80.1	311.0	44.0	90.0	39.0	148.0				
700	1.560	1828.0	24756.0	1828.0	24756.0	1280.0	12800.0	1200.0	5520.0	466.0	1340.0	86.1	331.0	46.0	92.0	40.0	158.0				
750	1.671	1956.0	26536.0	1956.0	26536.0	1400.0	14000.0	1320.0	6000.0	512.0	1464.0	92.1	351.0	48.0	94.0	41.0	168.0				
800	1.782	2084.0	28316.0	2084.0	28316.0	1520.0	15200.0	1440.0	6480.0	558.0	1588.0	98.1	371.0	50.0	96.0	42.0	178.0				
850	1.894	2212.0	30096.0	2212.0	30096.0	1640.0	16400.0	1560.0	7040.0	604.0	1712.0	104.1	391.0	52.0	98.0	43.0	188.0				
900	2.005	2340.0	31876.0	2340.0	31876.0	1760.0	17600.0	1680.0	7600.0	650.0	1836.0	110.1	411.0	54.0	100.0	44.0	198.0				
950	2.117	2468.0	33656.0	2468.0	33656.0	1880.0	18800.0	1800.0	8160.0	696.0	1960.0	116.1	431.0	56.0	102.0	45.0	208.0				
1000	2.228	2596.0	35436.0	2596.0	35436.0	2000.0	20000.0	1920.0	8720.0	742.0	2084.0	122.1	451.0	58.0	104.0	46.0	218.0				
1100	2.451	2912.0	40216.0	2912.0	40216.0	2240.0	22400.0	2160.0	9840.0	816.0	2304.0	130.1	471.0	60.0							

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	Com- pressed Air	Pressure Drop of Air In Pounds per Square Inch Per 100 Feet of Schedule 40 Pipe											
Cubic Feet Per Minute at 60 F and 14.7 psia	Cubic Feet Per Minute at 60 F and 100 psig	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	0.020								
2	0.256	1.31	0.285	0.064	0.042	3/4"							
3	0.384	3.06	0.605	0.133	0.071								
4	0.513	4.83	1.04	0.226	0.106	0.027							
5	0.641	7.45	1.58	0.343									
6	0.769	10.6	2.23	0.408	0.148	0.037	1"						
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"	1 1/2"				
15	1.922	...	13.0	2.73	0.834	0.201	0.062						
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				
45	5.767	...	...	23.1	6.90	1.62	0.474	0.116	0.055			2"	
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			23.1	6.60	1.58	0.727	0.205			
200	25.63	0.107	0.036	3 1/2"		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		...	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		...	18.0	8.04	2.21			
650	83.30	1.04	0.342	0.163	0.086	5"	...	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	0.046		35.9	16.0	4.40			
900	115.3	1.95	0.642	0.305	0.160	0.051	6"	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			
1000	128.2	2.40	0.788	0.375	0.197	0.063	0.025	...	22.1	6.06			
1100	141.0	2.89	0.948	0.451	0.236	0.075	0.030	...	26.7	7.29			
1200	153.8	3.44	1.13	0.533	0.279	0.089	0.035	...	31.8	8.63			
1300	166.6	4.01	1.32	0.626	0.327	0.103	0.041	...	37.3	10.1			
1400	179.4	4.65	1.52	0.718	0.377	0.119	0.047			11.8			
1500	192.2	5.31	1.74	0.824	0.431	0.136	0.054			13.5			
1600	205.1	6.04	1.97	0.932	0.490	0.154	0.061			15.3			
1800	230.7	7.65	2.50	1.18	0.616	0.193	0.075			19.3			
2000	256.3	9.44	3.06	1.45	0.757	0.237	0.094	0.023	10"	23.9			
2500	320.4	14.7	4.76	2.25	1.17	0.366	0.143	0.035		37.3			
3000	384.5	21.1	6.82	3.20	1.67	0.524	0.204	0.051	0.016				
3500	448.6	28.8	9.23	4.33	2.26	0.709	0.276	0.068	0.022				
4000	512.6	37.6	12.1	5.66	2.94	0.919	0.358	0.088	0.028				
4500	576.7	47.6	15.3	7.16	3.69	1.16	0.450	0.111	0.035	12"			
5000	640.8	...	18.8	8.85	4.56	1.42	0.552	0.136	0.043	0.018			
6000	769.0	...	27.1	12.7	6.57	2.03	0.794	0.195	0.061	0.025			
7000	897.1	...	36.9	17.2	8.94	2.76	1.07	0.262	0.082	0.034			
8000	1025	...	...	22.5	11.7	3.59	1.39	0.339	0.107	0.044			
9000	1153	...	...	28.5	14.9	4.54	1.76	0.427	0.134	0.055			
10000	1282	...	...	35.2	18.4	5.60	2.16	0.526	0.164	0.067			
11000	1410	...	...	...	22.2	6.78	2.62	0.633	0.197	0.081			
12000	1538	...	...	...	26.4	8.07	3.09	0.753	0.234	0.096			
13000	1666	...	...	...	31.0	9.47	3.63	0.884	0.273	0.112			
14000	1794	...	...	...	36.0	11.0	4.21	1.02	0.316	0.129			
15000	1922	...	...	...	...	12.6	4.84	1.17	0.364	0.148			
16000	2051	...	...	...	...	14.3	5.50	1.33	0.411	0.167			
18000	2307	...	...	...	...	18.2	6.96	1.68	0.520	0.213			
20000	2563	...	...	...	...	22.4	8.60	2.01	0.642	0.260			
22000	2820	...	...	...	...	27.1	10.4	2.50	0.771	0.314			
24000	3076	...	...	...	...	32.3	12.4	2.97	0.918	0.371			
26000	3332	...	...	...	...	37.9	14.5	3.49	1.12	0.435			
28000	3588	...	...	...	...	...	16.9	4.04	1.25	0.505			
30000	3845	...	...	...	...	...	19.3	4.64	1.42	0.520			

Commercial Wrought Steel Pipe Data

Schedule Wall Thickness—Per ASA B36.10-1950

	Nominal Pipe Size	Outside Diam- eter	Thick- ness	Inside Diameter		Inside Diameter Functions (In Inches)				Transverse Internal Area	
				d Inches	D Feet	d ²	d ³	d ⁴	d ⁵	a Sq. In.	A Sq. Ft.
Schedule 10	14	14	0.250	13.5	1.125	182.25	2460.4	33215.	448400.	143.14	0.994
	16	16	0.250	15.5	1.291	240.25	3723.9	57720.	894660.	188.69	1.310
	18	18	0.250	17.5	1.4583	306.25	5359.4	93789.	1641309.	240.53	1.670
	20	20	0.250	19.5	1.625	380.25	7414.9	144590.	2819500.	298.65	2.074
	24	24	0.250	23.5	1.958	552.25	12977.	304980.	7167030.	433.74	3.012
	30	30	0.312	29.376	2.448	862.95	25350.	744288.	21864218.	677.76	4.707
Schedule 20	8	8.625	0.250	8.125	0.6771	66.02	536.38	4359.3	35409.	51.85	0.3601
	10	10.75	0.250	10.25	0.8542	105.06	1076.9	11038.	113141.	82.52	0.5731
	12	12.75	0.250	12.25	1.021	150.06	1838.3	22518.	275855.	117.86	0.8185
	14	14	0.312	13.376	1.111	178.92	2393.2	32012.	428185.	140.52	0.9758
	16	16	0.312	15.376	1.281	236.42	3635.2	55894.	859442.	185.69	1.290
	18	18	0.312	17.376	1.448	301.92	5246.3	91156.	1583978.	237.13	1.647
Schedule 30	20	20	0.375	19.250	1.604	370.56	7133.3	137317.	2643352.	291.04	2.021
	24	24	0.375	23.25	1.937	540.56	12568.	292205.	6793832.	424.56	2.948
	30	30	0.500	29.00	2.417	841.0	24389.	707281.	20511149.	660.52	4.587
	8	8.625	0.277	8.071	0.6726	65.14	525.75	4243.2	34248.	51.16	0.3553
	10	10.75	0.307	10.136	0.8447	102.74	1041.4	10555.	106987.	80.69	0.5603
	12	12.75	0.330	12.09	1.0075	146.17	1767.2	21366.	258304.	114.80	0.7972
Schedule 40	14	14	0.375	13.25	1.1042	175.56	2326.2	30821.	408394.	137.88	0.9575
	16	16	0.375	15.25	1.2708	232.56	3546.6	54084.	824801.	182.65	1.268
	18	18	0.438	17.124	1.4270	293.23	5021.3	85984.	1472397.	230.30	1.599
	20	20	0.500	19.00	1.5833	361.00	6859.0	130321.	2476099.	283.53	1.969
	24	24	0.562	22.876	1.9063	523.31	11971.	273853.	6264703.	411.00	2.854
	30	30	0.625	28.75	2.3958	826.56	23764.	683201.	19642160.	649.18	4.508
Schedule 60	1/8	0.405	0.068	0.269	0.0224	0.0724	0.0195	0.005242	0.00141	0.057	0.00040
	1/4	0.540	0.088	0.364	0.0303	0.1325	0.0482	0.01756	0.00639	0.104	0.00072
	3/8	0.675	0.091	0.493	0.0411	0.2430	0.1198	0.05905	0.02912	0.191	0.00133
	1/2	0.840	0.109	0.622	0.0518	0.3869	0.2406	0.1497	0.09310	0.304	0.00211
	3/4	1.050	0.113	0.824	0.0687	0.679	0.5595	0.4610	0.3799	0.533	0.00371
	1	1.315	0.133	1.049	0.0874	1.100	1.154	1.210	1.270	0.864	0.00600
	1 1/4	1.660	0.140	1.380	0.1150	1.904	2.628	3.625	5.005	1.495	0.01040
	1 1/2	1.900	0.145	1.610	0.1342	2.592	4.173	6.718	10.82	2.036	0.01414
	2	2.375	0.154	2.067	0.1722	4.272	8.831	18.250	37.72	3.355	0.02330
	2 1/2	2.875	0.203	2.469	0.2057	6.096	15.051	37.161	91.75	4.788	0.03322
	3	3.500	0.216	3.068	0.2557	9.413	28.878	88.605	271.8	7.393	0.05130
	3 1/2	4.000	0.226	3.548	0.2957	12.59	44.663	158.51	562.2	9.886	0.06870
	4	4.500	0.237	4.026	0.3355	16.21	65.256	262.76	1058.	12.730	0.08840
	5	5.563	0.258	5.047	0.4206	25.47	128.56	648.72	3275.	20.006	0.1390
	6	6.625	0.280	6.065	0.5054	36.78	223.10	1352.8	8206.	28.891	0.2006
	8	8.625	0.322	7.981	0.6651	63.70	508.36	4057.7	32380.	50.027	0.3474
	10	10.75	0.365	10.02	0.8350	100.4	1006.0	10080.	101000.	78.855	0.5475
	12	12.75	0.406	11.938	0.9965	142.5	1701.3	20306.	242470.	111.93	0.7773
	14	14.0	0.438	13.124	1.0937	172.24	2260.5	29666.	389340.	135.28	0.9394
Schedule 80	16	16.0	0.500	15.000	1.250	225.0	3375.0	50625.	759375.	176.72	1.2272
	18	18.0	0.562	16.876	1.4063	284.8	4806.3	81111.	1368820.	223.68	1.5533
	20	20.0	0.593	18.814	1.5678	354.0	6659.5	125320.	2357244.	278.00	1.9305
	24	24.0	0.687	22.626	1.8855	511.9	11583.	262040.	5929784.	402.07	2.7921
	8	8.625	0.406	7.813	0.6511	61.04	476.93	3725.9	29113.	47.94	0.3329
	10	10.75	0.500	9.750	0.8125	95.06	926.86	9036.4	88110.	74.66	0.5185
Schedule 100	12	12.75	0.562	11.626	0.9688	135.16	1571.4	18268.	212399.	106.16	0.7372
	14	14.0	0.593	12.814	1.0678	164.20	2104.0	26962.	345480.	128.96	0.8956
	16	16.0	0.656	14.688	1.2240	215.74	3168.8	46544.	683618.	169.44	1.1766
	18	18.0	0.750	16.500	1.3750	272.25	4492.1	74120.	1222982.	213.83	1.4849
	20	20.0	0.812	18.376	1.5313	337.68	6205.2	114028.	2095342.	265.21	1.8417
	24	24.0	0.968	22.064	1.8387	486.82	10741.	236994.	5229036.	382.35	2.6552
Schedule 120	1/8	0.405	0.095	0.215	0.0179	0.0462	0.00994	0.002134	0.000459	0.036	0.00025
	1/4	0.540	0.119	0.302	0.0252	0.0912	0.0275	0.008317	0.002513	0.072	0.00050
	3/8	0.675	0.126	0.423	0.0353	0.1789	0.0757	0.03200	0.01354	0.141	0.00098
	1/2	0.840	0.147	0.546	0.0455	0.2981	0.1628	0.08886	0.04852	0.234	0.00163
	3/4	1.050	0.154	0.742	0.0618	0.5506	0.4085	0.3032	0.2249	0.433	0.00300
	1	1.315	0.179	0.957	0.0797	0.9158	0.8765	0.8387	0.8027	0.719	0.00499
Schedule 150	1 1/4	1.660	0.191	1.278	0.1065	1.633	2.087	2.6667	3.409	1.283	0.00891

(continued on the next page)

Commercial Wrought Steel Pipe Data

Schedule Wall Thickness—Per ASA B36.10-1950

Nominal Pipe Size	Outside Diam- eter	Thick- ness	Inside Diameter		Inside Diameter Functions (In Inches)				Transverse Internal Area		
			<i>d</i>	<i>D</i>	<i>d</i> ²	<i>d</i> ³	<i>d</i> ⁴	<i>d</i> ⁵	<i>a</i>	<i>A</i>	
Inches	Inches	Inches	Inches	Feet					Sq. In.	Sq. Ft.	
Schedule 80—cont.	1½	1.900	0.200	1.500	0.1250	2.250	3.375	5.062	7.594	1.767	0.01225
	2	2.375	0.218	1.939	0.1616	3.760	7.290	14.136	27.41	2.953	0.02050
	2½	2.875	0.276	2.323	0.1936	5.396	12.536	29.117	67.64	4.238	0.02942
	3	3.5	0.300	2.900	0.2417	8.410	24.389	70.728	205.1	6.605	0.04587
	3½	4.0	0.318	3.364	0.2803	11.32	38.069	128.14	430.8	8.888	0.06170
	4	4.5	0.337	3.826	0.3188	14.64	56.006	214.33	819.8	11.497	0.07986
	5	5.563	0.375	4.813	0.4011	23.16	111.49	536.38	2583.	18.194	0.1263
	6	6.625	0.432	5.761	0.4801	33.19	191.20	1101.6	6346.	26.067	0.1810
	8	8.625	0.500	7.625	0.6354	58.14	443.32	3380.3	25775.	45.663	0.3171
	10	10.75	0.593	9.564	0.7970	91.47	874.82	8366.8	80020.	71.84	0.4989
	12	12.75	0.687	11.376	0.9480	129.41	1472.2	16747.	190523.	101.64	0.7058
	14	14.0	0.750	12.500	1.0417	156.25	1953.1	24414.	305176.	122.72	0.8522
	16	16.0	0.843	14.314	1.1928	204.89	2932.8	41980.	600904.	160.92	1.1175
	18	18.0	0.937	16.126	1.3438	260.05	4193.5	67626.	1090518.	204.24	1.4183
	20	20.0	1.031	17.938	1.4948	321.77	5771.9	103536.	1857248.	252.72	1.7550
	24	24.0	1.218	21.564	1.7970	465.01	10027.	216234.	4662798.	365.22	2.5362
Schedule 100	8	8.625	0.593	7.439	0.6199	55.34	411.66	3062.	22781.	43.46	0.3018
	10	10.75	0.718	9.314	0.7762	86.75	807.99	7526.	69357.	68.13	0.4732
	12	12.75	0.843	11.064	0.9220	122.41	1354.4	14985.	165791.	96.14	0.6677
	14	14.0	0.937	12.126	1.0105	147.04	1783.0	21621.	262173.	115.49	0.8020
	16	16.0	1.031	13.938	1.1615	194.27	2707.7	37740.	526020.	152.58	1.0596
	18	18.0	1.156	15.688	1.3057	246.11	3861.0	60572.	950250.	193.30	1.3423
	20	20.0	1.281	17.438	1.4532	304.08	5302.6	92467.	1612438.	238.83	1.6585
	24	24.0	1.531	20.938	1.7448	438.40	9179.2	192195.	4024179.	344.32	2.3911
Schedule 120	4	4.50	0.438	3.624	0.302	13.133	47.595	172.49	625.1	10.315	0.07163
	5	5.563	0.500	4.543	0.3802	20.82	95.006	433.5	1978.	16.35	0.1136
	6	6.625	0.562	5.501	0.4584	30.26	166.47	915.7	5037.	23.77	0.1650
	8	8.625	0.718	7.189	0.5991	51.68	371.54	2671.	19202.	40.59	0.2819
	10	10.75	0.843	9.064	0.7553	82.16	744.66	6750.	61179.	64.53	0.4481
	12	12.75	1.000	10.750	0.8959	115.56	1242.3	13355.	143563.	90.76	0.6303
	14	14.0	1.093	11.814	0.9845	139.57	1648.9	19480.	230137.	109.62	0.7612
	16	16.0	1.218	13.564	1.1303	183.98	2495.5	33849.	459133.	144.50	1.0035
	18	18.0	1.375	15.250	1.2708	232.56	3546.6	54086.	824804.	182.66	1.2684
	20	20.0	1.500	17.000	1.4166	289.00	4913.0	83521.	1419857.	226.98	1.5762
	24	24.0	1.812	20.376	1.6980	415.18	8459.7	172375.	3512313.	326.08	2.2645
	8	8.625	0.812	7.001	0.5834	49.01	343.15	2402.	16819.	38.50	0.2673
10	10.75	1.000	8.750	0.7292	76.56	669.92	5862.	51291.	60.13	0.4176	
12	12.75	1.125	10.500	0.8750	110.25	1157.6	12155.	127628.	86.59	0.6013	
14	14.0	1.250	11.500	0.9583	132.25	1520.9	17490.	201136.	103.87	0.7213	
16	16.0	1.438	13.124	1.0937	172.24	2260.5	29666.	389340.	135.28	0.9394	
18	18.0	1.562	14.876	1.2396	221.30	3292.0	48972.	728502.	173.80	1.2070	
20	20.0	1.750	16.5	1.3750	272.25	4492.1	74120.	1222981.	213.82	1.4849	
24	24.0	2.062	19.876	1.6563	395.06	7852.1	156069.	3102022.	310.28	2.1547	
Schedule 140	½	0.840	0.187	0.466	0.0388	0.2172	0.1012	0.04716	0.02197	0.1706	0.00118
	¾	1.050	0.218	0.614	0.0512	0.3770	0.2315	0.1421	0.08726	0.2961	0.00206
	1	1.315	0.250	0.815	0.0679	0.6642	0.5413	0.4412	0.3596	0.5217	0.00362
	1¼	1.660	0.250	1.160	0.0966	1.346	1.561	1.811	2.100	1.057	0.00734
	1½	1.900	0.281	1.338	0.1115	1.790	2.395	3.205	4.288	1.406	0.00976
	2	2.375	0.343	1.689	0.1407	2.853	4.818	8.138	13.74	2.241	0.01556
	2½	2.875	0.375	2.125	0.1771	4.516	9.596	20.39	43.33	3.546	0.02463
	3	3.50	0.437	2.626	0.2188	6.896	18.109	47.55	124.9	5.416	0.03761
	4	4.50	0.531	3.438	0.2865	11.82	40.637	139.7	480.3	9.283	0.06447
	5	5.563	0.625	4.313	0.3594	18.60	80.230	346.0	1492.	14.61	0.1015
	6	6.625	0.718	5.189	0.4324	26.93	139.72	725.0	3762.	21.15	0.1469
	8	8.625	0.906	6.813	0.5677	46.42	316.24	2155.	14679.	36.46	0.2532
	10	10.75	1.125	8.500	0.7083	72.25	614.12	5220.	44371.	56.75	0.3941
	12	12.75	1.312	10.126	0.8438	102.54	1038.3	10514.	106461.	80.53	0.5592
	14	14.0	1.406	11.188	0.9323	125.17	1400.4	15668.	175292.	98.31	0.6827
	16	16.0	1.593	12.814	1.0678	164.20	2104.0	26961.	345482.	128.96	0.8956
18	18.0	1.781	14.438	1.2032	208.45	3009.7	43454.	627387.	163.72	1.1369	
20	20.0	1.968	16.064	1.3387	258.05	4145.3	66590.	1069715.	202.67	1.4074	
24	24.0	2.343	19.314	1.6095	373.03	7204.7	139152.	2687582.	292.98	2.0346	

Commercial Wrought Steel Pipe Data

(Per ASA B36.10-1950)

Nominal Pipe Size Inches	Outside Diam- eter Inches	Thick- ness Inches	Inside Diameter		Inside Diameter Functions (In Inches)				Transverse Internal Area	
			d Inches	D Feet	d ²	d ³	d ⁴	d ⁵	A Sq. In.	A Sq. Ft.
Standard Wall Pipe										
1/8	0.405	0.068	0.269	0.0224	0.0724	0.0195	0.00524	0.00141	0.057	0.00040
1/4	0.540	0.088	0.364	0.0303	0.1325	0.0482	0.01756	0.00639	0.104	0.00072
3/8	0.675	0.091	0.493	0.0411	0.2430	0.1198	0.05905	0.02912	0.191	0.00133
1/2	0.840	0.109	0.622	0.0518	0.3869	0.2406	0.1497	0.0931	0.304	0.00211
3/4	1.050	0.113	0.824	0.0687	0.679	0.5595	0.4610	0.3799	0.533	0.00371
1	1.315	0.133	1.049	0.0874	1.100	1.154	1.210	1.270	0.864	0.00600
1 1/4	1.660	0.140	1.380	0.1150	1.904	2.628	3.625	5.005	1.495	0.01040
1 1/2	1.900	0.145	1.610	0.1342	2.592	4.173	6.718	10.82	2.036	0.01414
2	2.375	0.154	2.067	0.1722	4.272	8.831	18.250	37.72	3.355	0.02330
2 1/2	2.875	0.203	2.469	0.2057	6.096	15.051	37.161	91.75	4.788	0.03322
3	3.500	0.216	3.068	0.2557	9.413	28.878	88.605	271.8	7.393	0.05130
3 1/2	4.000	0.226	3.548	0.2957	12.59	44.663	158.51	562.2	9.886	0.06870
4	4.500	0.237	4.026	0.3355	16.21	65.256	262.76	1058.	12.730	0.08840
5	5.563	0.258	5.047	0.4206	25.47	128.56	648.72	3275.	20.006	0.1390
6	6.625	0.280	6.065	0.5054	36.78	223.10	1352.8	8206.	28.891	0.2006
8	8.625	0.277	8.071	0.6725	65.14	525.75	4243.0	34248.	51.161	0.3553
	8.625	0.322	7.981	0.6651	63.70	508.36	4057.7	32380.	50.027	0.3474
10	10.75	0.279	10.192	0.8493	103.88	1058.7	10789.	109876.	81.585	0.5666
	10.75	0.307	10.136	0.8446	102.74	1041.4	10555.	106987.	80.691	0.5604
	10.75	0.365	10.020	0.8350	100.4	1006.0	10080.	101000.	78.855	0.5475
12	12.75	0.330	12.090	1.0075	146.17	1767.2	21366.	258300.	114.80	0.7972
	12.75	0.375	12.000	1.000	144.0	1728.0	20736.	248800.	113.10	0.7854
Extra Strong Pipe										
1/8	0.405	0.095	0.215	0.0179	0.0462	0.00994	0.002134	0.000459	0.036	0.00025
1/4	0.540	0.119	0.302	0.0252	0.0912	0.0275	0.008317	0.002513	0.072	0.0005
3/8	0.675	0.126	0.423	0.0353	0.1789	0.0757	0.03201	0.01354	0.141	0.00098
1/2	0.840	0.147	0.546	0.0455	0.2981	0.1628	0.08886	0.04852	0.234	0.00163
3/4	1.050	0.154	0.742	0.0618	0.5506	0.4085	0.3032	0.2249	0.433	0.00300
1	1.315	0.179	0.957	0.0797	0.9158	0.8765	0.8387	0.8027	0.719	0.00499
1 1/4	1.660	0.191	1.278	0.1065	1.633	2.087	2.6667	3.409	1.283	0.00891
1 1/2	1.900	0.200	1.500	0.1250	2.250	3.375	5.062	7.594	1.767	0.01225
2	2.375	0.218	1.939	0.1616	3.760	7.290	14.136	27.41	2.953	0.02050
2 1/2	2.875	0.276	2.323	0.1936	5.396	12.536	29.117	67.64	4.238	0.02942
3	3.500	0.300	2.900	0.2417	8.410	24.389	70.728	205.1	6.605	0.04587
3 1/2	4.000	0.318	3.364	0.2803	11.32	38.069	128.14	430.8	8.888	0.06170
4	4.500	0.337	3.826	0.3188	14.64	56.006	214.33	819.8	11.497	0.07986
5	5.563	0.375	4.813	0.4011	23.16	111.49	536.6	2583.	18.194	0.1263
6	6.625	0.432	5.761	0.4801	33.19	191.20	1101.6	6346.	26.067	0.1810
8	8.625	0.500	7.625	0.6354	58.14	443.32	3380.3	25775.	45.663	0.3171
10	10.75	0.500	9.750	0.8125	95.06	926.86	9036.4	88110.	74.662	0.5185
12	12.75	0.500	11.750	0.9792	138.1	1622.2	19072.	223970.	108.434	0.7528
Double Extra Strong Pipe										
1/2	0.840	0.294	0.252	0.0210	0.0635	0.0160	0.004032	0.00102	0.050	0.00035
3/4	1.050	0.308	0.434	0.0362	0.1884	0.0817	0.03549	0.01540	0.148	0.00103
1	1.315	0.358	0.599	0.0499	0.3588	0.2149	0.1287	0.07711	0.282	0.00196
1 1/4	1.660	0.382	0.896	0.0747	0.8028	0.7193	0.6445	0.5775	0.630	0.00438
1 1/2	1.900	0.400	1.100	0.0917	1.210	1.331	1.4641	1.611	0.950	0.00660
2	2.375	0.436	1.503	0.1252	2.259	3.395	5.1031	7.670	1.774	0.01232
2 1/2	2.875	0.552	1.771	0.1476	3.136	5.554	9.8345	17.42	2.464	0.01710
3	3.500	0.600	2.300	0.1917	5.290	12.167	27.984	64.36	4.155	0.02885
3 1/2	4.000	0.636	2.728	0.2273	7.442	20.302	55.383	151.1	5.845	0.04059
4	4.500	0.674	3.152	0.2627	9.935	31.315	98.704	311.1	7.803	0.05419
5	5.563	0.750	4.063	0.3386	16.51	67.072	272.58	1107.	12.966	0.09006
6	6.625	0.864	4.897	0.4081	23.98	117.43	575.04	2816.	18.835	0.1308
8	8.625	0.875	6.875	0.5729	47.27	324.95	2234.4	15360.	37.122	0.2578

APPENDIX C

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