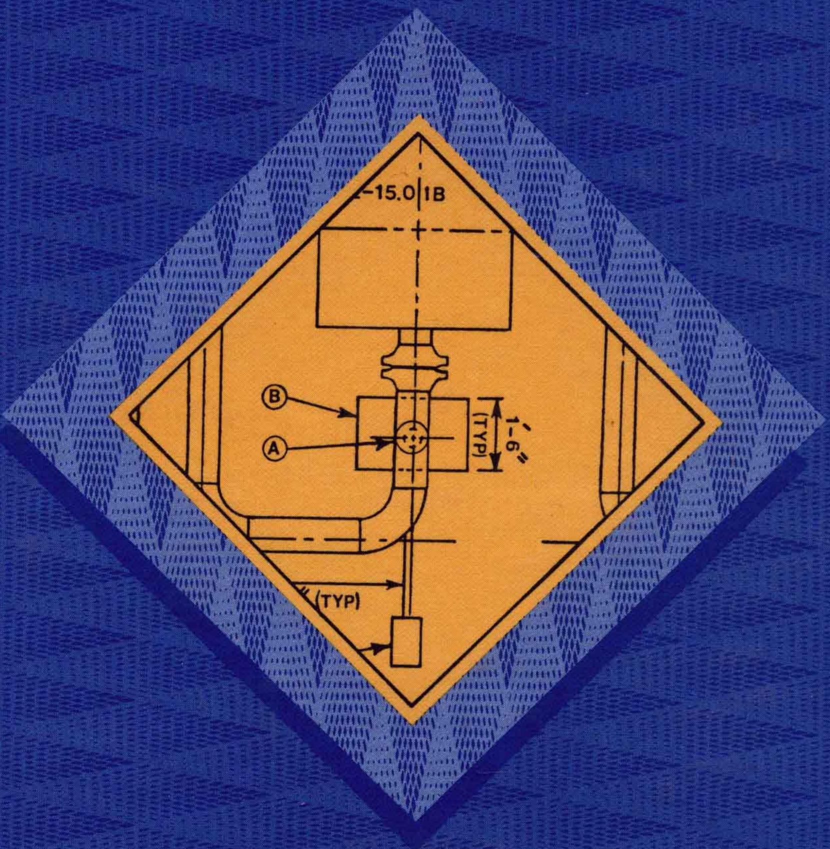
2nd Edition

Mechanical Design of Process Systems

Volume I

Piping and Pressure Vessels



A. Keith Escoe

Helps you design process equipment/components using the latest techniques in heat transfer, hydraulics, and static and dynamic analyses

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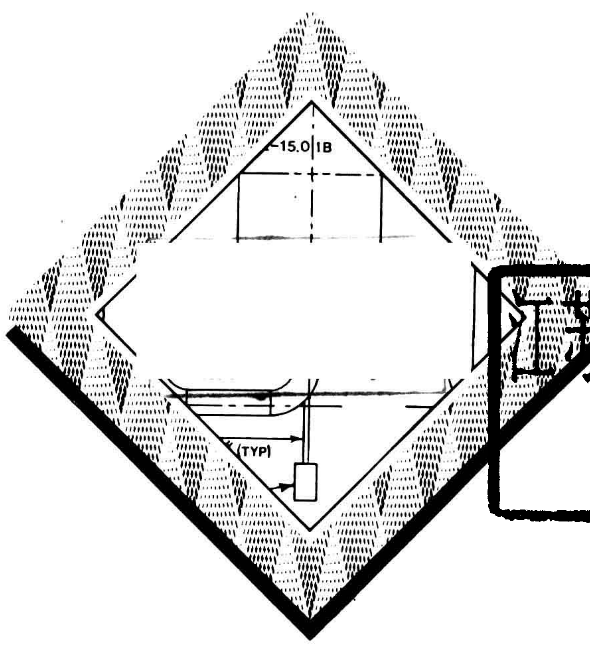

A. Keith Escoe

To the memory of my beloved parents, Aubrey H. Escoe and Odessa Davies Escoe; and to the dedicated engineer, Dr. Judith Arlene Resnik, U.S. astronaut aboard the ill-fated space shuttle *Challenger* (Flight 51-L).

Mechanical Design of Process Systems
Volume 1, Second Edition
Piping and Pressure Vessels

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Foreword

The engineer who understands the impact of process design decisions on mechanical design details is in a position to save his client or his company a lot of money.

That is because the test of any process design is in how cost-effectively it yields the desired product, and how "cost" generally translates to "equipment": How much will the process require? How long will it last? How much energy will it consume per unit of product?

In this two-volume work on *Mechanical Design of Process Systems,* A. K. Escoe has performed a monumental service for mechanical design engineers and chemical process engineers alike. It is presented in such a manner that even the neophyte engineer can grasp its full value. He has produced an in-depth review of the way in which process design specifications are interpreted into precise equipment designs. Perhaps most valuable of all are the extensive *worked examples* throughout the text, of actual designs that have been successfully executed in the field.

The piping system is the central nervous system of a fluid flow process, and the author has treated this with proper respect in two excellent chapters on fluid mechanics and the engineering mechanics of piping.

The chapter on heat transfer in vessels and piping illustrates lucidly the interrelationship between process and mechanical design. Every engineer working with industrial process systems will benefit from reading this chapter.

Although the author has made a herculean effort in covering the mechanical design of pressure vessels, heat exchangers, rotating equipment, and bins, silos and stacks, it is true that there are omissions. It is hoped that, as the author hints in his preface, a future volume might be added covering multiphase flow, specific cogeneration processes, turbines, and detailed piping dynamics.

Still, at this writing these two volumes comprise an outstanding practical reference for chemical and mechanical engineers and a detailed instructional manual for students.

I recommend these volumes highly for each design engineer's professional library.

John J. McKetta, Ph.D., P.E.
Joe C. Walter Professor of Chemical Engineering
University of Texas, Austin

Preface to the Second Edition

This book's purpose is to show how to apply mechanical engineering concepts to process system design. Process systems are common to a wide variety of industries including petrochemical processing, food and pharmaceutical manufacturing, power generation (including cogeneration), ship building, and even the aerospace industry. The book is based on years of proven, successful practice, and almost all of the examples described are from process systems now in operation.

This second edition comes during the development of much new technology, and reflects the wide use of the finite element method (FEM) in problem solving. This powerful numerical technique has been one of the greatest innovations of the twentieth century. This computerized method has opened possibilities that never before existed. Problems solutions have been verified and enhanced with FEM in portions of this new edition. Also included are new and reworked examples that reflect the latest codes and standards that are clear and easy to understand; a new appendix with computer algorithms; expanded discussions of heat transfer in piping and equipment; and new developments and techniques in flow-induced vibration.

While practicality is probably its key asset, this first volume contains a unique collection of valuable information, such as velocity head data; comparison of the flexibility and stiffness methods of pipe stress analyses; analysis of heat transfer through pipe supports and vessel skirts; a comprehensive method on the design of horizontal vessel saddles as well as a method to determine when wear plates are required; detailed static and dynamic methods of tower design considering wind gusts, vortex-induced vibration and seismic analysis of towers; and a comparative synopsis of the various national wind codes.

Topics included in the text are considered to be those typically encountered in engineering practice. Therefore, because most mechanical systems involve single-phase flow, two-phase flow is not covered. Because of its ubiquitous coverage in the literature, flange design is also excluded in this presentation. Since all of the major pressure vessel codes thoroughly discuss and illustrate the phenomenon of external pressure, this subject is only mentioned briefly.

This book is not intended to be a substitute or a replacement of any accepted code or standard. The reader is strongly encouraged to consult and be knowledgeable of any accepted standard or code that may govern. It is felt that this book is a valuable supplement to any standard or code used.

The book is slanted toward the practices of the ASME vessel and piping codes. In one area of vessel design the British Standard is favored because it provides excellent technical information on Zick rings. The book is written to be useful regardless of which code or standard is used. The intent is not to be heavily prejudiced toward any standard, but to discuss the issue—engineering. If one feels that a certain standard or code should be mentioned, please keep in mind that there are others who may be using different standards and it is impossible to discuss all of them.

The reader's academic level is assumed to be a bachelor of science degree in mechanical engineering, but engineers with bachelor of science degrees in civil, chemical, electrical, or other engineering disciplines should have little difficulty with the book, provided, of course, that they have received adequate academic training or experience.

Junior or senior undergraduate engineering students should find the book a useful introduction to the application of mechanical engineering to process systems.

Professors should find the book a helpful reference (and a source for potential exam problems), as well as a practical textbook for junior-, senior-, or graduate-level courses in the mechanical, civil, or chemical engineering fields. The book can also be used to supplement an introductory level textbook.

The French philosopher Voltaire once said, "Common sense is not very common," and unfortunately, this is sometimes the case in engineering. Common sense is often the by-product of experience, and while both are essential to sound engineering practice, neither can be learned from books alone. It is one of this book's goals to unite these three elements of "book learning," common sense, and experience to give the novice a better grasp of engineering principles and procedures, and serve as a practical design reference for the veteran engineer.

The contents of this book do not necessarily reflect the practices of the Exxon Chemical Company. I do wish to thank the company for its support in this endeavor, particularly Jim Wykowski. My friends and colleagues, Wolf Schmidt of Germany, and Matt Findlay of the United Kingdom, have provided helpful comments and many words of encouragement.

Finally I wish to thank Professor John J. McKetta of the University of Texas at Austin for his valuable and continued support. I wish to express gratitude to my friend D. H. Rawal of Bombay, India for his helpful comments and continuous support and enthusiasm for this project. I also wish to thank William J. Lowe and Timothy W. Calk of Gulf Publishing Company for their continued support and hard work. Last, and certainly not least, I wish to thank my wife, Emma, for her lasting patience.

A. Keith Escoe, P.E.

Contents

1

Piping Fluid Mechanics

The study of fluid energy in piping systems is a comprehensive subject that could in itself fill countless volumes. This chapter is primarily concerned with fluid energy dissipated as friction resulting in a head loss. Although this topic is popularly known in industry as "hydraulics," the term "piping fluid mechanics" is used here to avoid confusion.

BASIC EQUATIONS

The basic equation of fluid mechanics, originally derived by Daniel Bernoulli in 1738, evolved from the principle of conservation of energy:

$$-\frac{dP}{\rho} = \frac{d\,(V^2)}{2g_c} + \frac{g}{g_c}\,dY + dF + dH_A + dH_E \qquad (1\text{-}1)$$

where
- ρ = density, lb_m/ft^3 or g_m/cm^3
- P = pressure, lb_f/ft^2 or kg_f/cm^2
- V = velocity, ft/sec or cm/sec
- g_c = conversion constant, 32.17 ($ft\text{-}lb_m/sec^2lb_f$)
- g = gravitational acceleration = 32.2 ft/sec², cm/sec²; g/g_c = 1.0
- Y = height above datum, ft, cm
- dY = differential between height above datum and reference point, ft, cm
- F = head loss, friction loss, or frictional pressure drop, $ft\text{-}lb_f/lb_m$, $cm\text{-}kg_f/g_m$
- H_A = energy added by mechanical devices, e.g. pumps, $ft\text{-}lb_f/lb_m$, $cm\text{-}kg_f/g_m$
- H_E = energy extracted by mechanical devices, e.g. turbines, $ft\text{-}lb_f/lb_m$, $cm\text{-}kg_f/g_m$

Rewriting Equation 1-1 along a fluid streamline between points 1 and 2 with steady, incompressible flow and no mechanical energy added or extracted results in

$$\frac{P_1 - P_2}{\rho} = \frac{V_1^2 - V_2^2}{2g_c} + (Y_1 - Y_2)\frac{g}{g_c} + F \qquad (1\text{-}2)$$

where subscripts 1 and 2 refer to flow upstream (after the flow process) and downstream (before the flow process), respectively, and

$$\frac{P_1 - P_2}{\rho} = \text{change in pressure head}$$

$$\frac{V_1 - V_2}{2g_c} = \text{change in velocity head (kinetic energy)}$$

$$= dZ$$

$$(Y_1 - Y_2)\frac{g}{g_c} = \text{change in static head (potential energy)}$$

$$F = \text{friction loss in } \frac{ft\,(lb_f)}{lb_m},\ \frac{cm\,(kg_f)}{g_m}$$

The following are expressions of the Bernoulli equation when applied to various incompressible and compressible flow conditions:

Incompressible flow—

$$\frac{P_1 - P_2}{\rho} = \frac{V_1^2 - V_2^2}{2g_c} + (Z_1 - Z_2)\frac{g}{g_c} + F + H_A + H_E$$

Compressible-isothermal flow—

$$\left(\frac{P_1}{\rho_1}\right)\ln\left(\frac{P_1}{P_2}\right) = \frac{V_2^2}{2g}\left[1 - \left(\frac{A_2}{A_1}\right)^2\left(\frac{P_2}{P_1}\right)^2\right] + (Z_2 - Z_1)$$
$$+ F + H_A + H_E$$

Compressible-adiabatic flow—

$$\left(\frac{k}{k-1}\right)\left(\frac{P_1}{\rho_1}\right)\left[1-\left(\frac{P_2}{P_1}\right)^{(k-1)/k}\right] = \frac{V_2^2}{2g}\left[1-\left(\frac{P_2}{P_1}\right)^{2/k}\left(\frac{A_2}{A_1}\right)^2\right] + (Z_2 - Z_1) + F + H_A + H_E$$

where $\left(\frac{\rho}{\rho_1}\right)^k = \left(\frac{P}{P_1}\right)$ = general gas law

k = specific heat ratio (adiabatic coefficient), C_p/C_v
C_p = specific heat at constant pressure, $Btu/lb_m\text{-}°F$
C_v = specific heat at constant volume, $Btu/lb_m\text{-}°F$

Equation 1-2 is the analytical expression that states a pressure loss is caused by a change in velocity head, static head, and friction head. The most common units are "feet of head." lb_m and lb_f do not cancel out and the expression is exactly "energy ($ft\text{-}lb_f$) per pound of mass."

In most industrial fluid problems, Equation 1-2 is cumbersome to use, because the friction loss is the parameter most often desired. The friction loss is the work done by the fluid in overcoming viscous resistance. This friction loss can only rarely be analytically derived and is determined by empirical data developed through experimental testing.

Forcing a fluid through a pipe component requires energy. This energy is expended by shear forces that develop between the pipe wall and the fluid, and to a lesser extent among the fluid elements themselves. These shear forces are opposed to fluid flow and require excess energy to overcome. Figure 1-1 shows a simple version of this phenomenon and illustrates how shear stresses increase in the radial direction away from the pipe center line and are maximum within the boundary layer next to the wall. Friction energy loss is a result of these shear stresses next to the pipe wall. Excess loss in energy occurs because of local turbulence and changes in the direction and speed of flow. As a fluid changes direction, energy is expended because of a change in momentum. The methods used to determine energy loss caused by wall friction are essentially the same, where the pipe component is treated as a straight piece of pipe. However, the methods used to determine energy loss caused by change in momentum differ, and a couple are described as follows.

Equivalent Length

In this approach to determining energy loss caused by a change in fluid momentum, a piping component is extended a theoretical length that would yield the same energy loss as the actual component. This length is called the "equivalent length" because it is that length required to obtain the same amount of friction pressure drop as the piping component alone. The major problem with

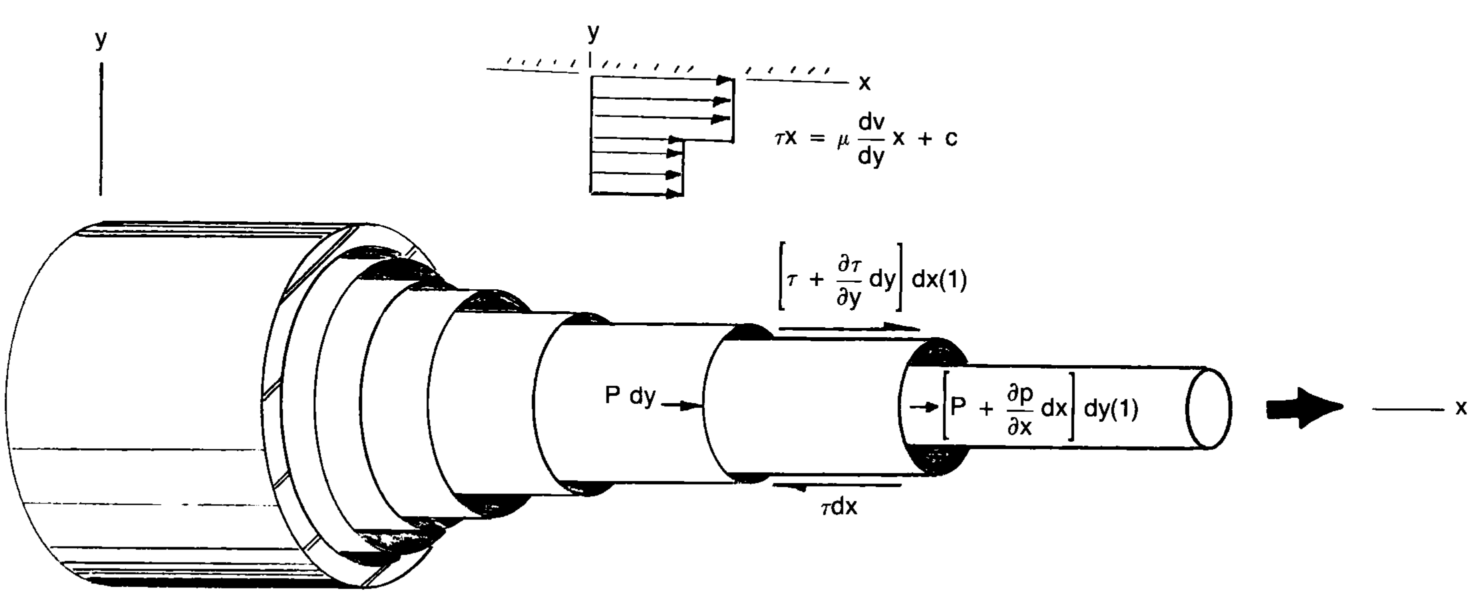

Figure 1-1. Shear stresses in fully developed flow. Shown here are imaginary fluid elements "slipping" over one another.

this method is that the equivalent length for a pipe component varies with the Reynolds number, roughness, size, and geometry of the pipe. All these parameters must be analyzed in using this method.

Velocity Head Method

Since the excess head loss is mostly attributed to fluid turbulence, the velocity head method is widely accepted and is replacing the equivalent length method in fluid calculations. Throughout this book, the velocity head approach will be used.

The velocity head is the amount of kinetic energy in a fluid, $V^2/2g_c$. This quantity may be represented by the amount of potential energy required to accelerate a fluid to a given velocity. Consider a tank holding a fluid with a pipe entrance shown in Figure 1-2. We draw a streamline from point 1 of the fluid surface to point 2 at the pipe entrance. Applying Equation 1-2 at point 1 we obtain the following:

$$\frac{P_1}{\rho} = Y_1 \frac{g}{g_c}$$

And applying Equation 1-2 at point 2 we have

$$\frac{P_1 - P_2}{\rho} = \frac{P_1}{\rho} - \frac{V_2^2}{2g_c}$$

in which the change in fluid pressure between points 1 and 2 is $V_2^2/2g_c$ or one velocity head. A pressure gauge mounted on the pipe entrance would record the difference of pressure of one velocity head. This term is accounted for in Equation 1-2 by $V_1^2 - V_2^2/2g_c$.

Analyzing a simple conversion from potential to kinetic energy is an elementary procedure, as demonstrated. After the fluid passes through the pipe entrance

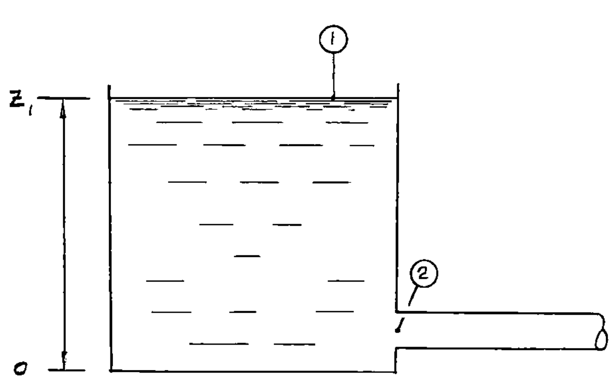

Figure 1-2. Storage tank.

into the piping system, the factor F in Equation 1-2 becomes the desired parameter. This friction loss is the work done by the fluid in overcoming viscous resistance and loss attributed to turbulence. The parameter F is composed of two components, pipe wall friction and losses for the various pipe fittings, pipe entrances, pipe exits, and fluid obstructions that contribute to a loss in fluid energy. These latter losses are described in terms of velocity heads, K_i. In solving for F in Equation 1-2, we first obtain pressure loss attributed to pipe wall friction, represented by

$$-\Delta P_F = F = \frac{\rho V^2}{2g_c}\left(\frac{fL}{d}\right) \tag{1-3}$$

By adding values of velocity head losses to Equation 1-3, we obtain the following for any piping system:

$$-\Delta P_f = \left(\frac{fL}{d} + \sum K_i\right)\frac{\rho V^2}{2g_c} \tag{1-4}$$

where fL/d is the dependent pipe friction of the pipe of diameter d over the length L, and ΣK_i the summation of velocity head losses. Equation 1-4 provides the friction pressure drop in a pipe for a steady-state incompressible fluid of fully developed flow with a flat velocity profile. Examples of this equation are given after the terms in Equation 1-4 are further explained.

The term (fL/d) $(\rho V^2/2g_c)$ expresses the amount of energy loss attributed to shear forces at the pipe wall and is based on experimental evidence. It is a function of the pipe component length and diameter and the velocity of the fluid. Writing the relationship for friction pressure drop as a result of pipe wall friction results in

$$F_{P_f} = \frac{fL}{144d}\frac{\rho V^2}{2g_c} \tag{1-5}$$

where F_{P_f} = friction loss, psi
L = length of pipe, in.
d = corroded inside diameter, in.

The other terms are explained with Equation 1-1. Equation 1-5 may be expressed in various forms. To express flow rate in gpm (w) and d in inches use

$$F_{P_f} = 0.000217\ fLW/d^5 \tag{1-5a}$$

Equation 1-5 is the most commonly used relationship and is known as the Fanning equation. Dividing the equation by $\rho/144$ yields feet of friction loss rather than psi.

The reader is cautioned in applying the friction factor f, because it is not always defined as above and some authors use $4f_l$ in place of f. If such factors are used, particular attention should be paid to the specific friction factor chart used.

The friction factor f is dependent upon the dimensionless term expressing the roughness of the pipe (E/D, where E is the depth of the pipe) and the dimensionless Reynolds number $N_{Re} = d\rho V/M$, where μ is the absolute viscosity of the fluid, in-lb_f-sec/ft^2. The Reynolds number is the single most important parameter in fluid mechanics because it establishes flow regimes and dynamic similarity. The relationship between the friction factor f, the pipe roughness, and the Reynolds number is shown in the classic relationship given by Moody in Figure 1-3.

Figure 1-3 may be presented in a more convenient form as shown in Figure 1-4, where the relative roughness of the pipe is based on a single value of roughness. This value of roughness must be an average value estimated to simplify the problem. The figures presented herein are the best available until more reliable friction factor data can be obtained and better understood through use of new methods for measuring roughness.

Figure 1-3 is broken into three flow regimes—laminar, transition from laminar to turbulent, and turbulent. The Reynolds numbers establishing these zones are 2,100 for laminar, 2,100 to 3,000 for transition zone, and 3,000 or more for turbulent.

The basis for Figure 1-3 is the classic Colebrook equation

$$\frac{1}{(f)^{0.5}} = -2 \log_{10}\left(\frac{\epsilon/d}{3.7} + \frac{2.51}{N_{Re}(f)^{0.5}}\right) \tag{1-6a}$$

$$\text{for } (3{,}000 \text{ to } 4{,}000) \leqslant N_{Re} \leqslant 10^8$$

For laminar flow the friction factor is determined by the simple expression

$$f = \frac{64}{N_{Re}} \tag{1-6b}$$

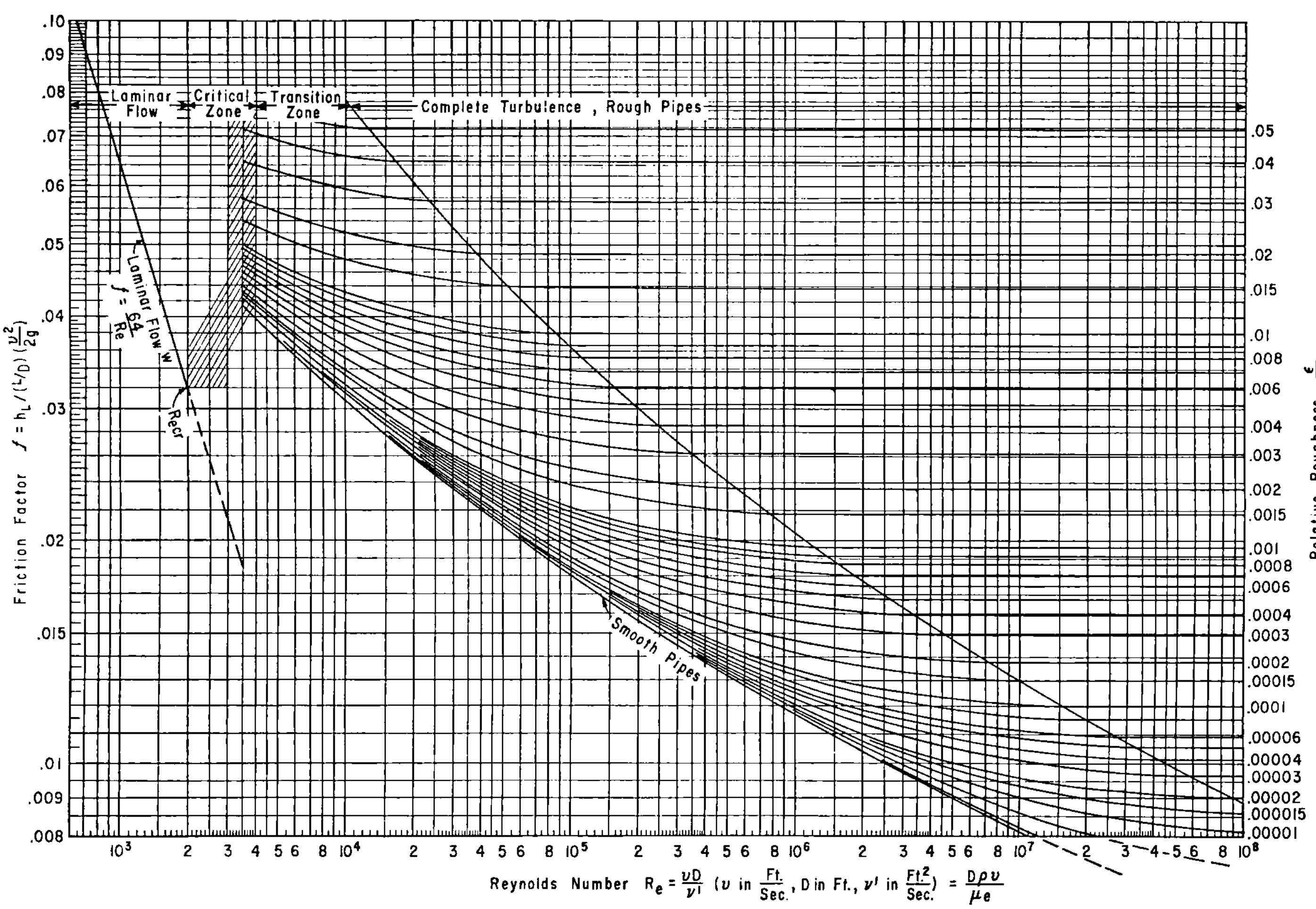

Figure 1-3. Moody friction factors. (Reprinted from *Pipe Friction Manual,* © 1954 by Hydraulic Institute. Data from L. F. Moody, *Friction Factors for Pipe Flow,* permission of ASME.)

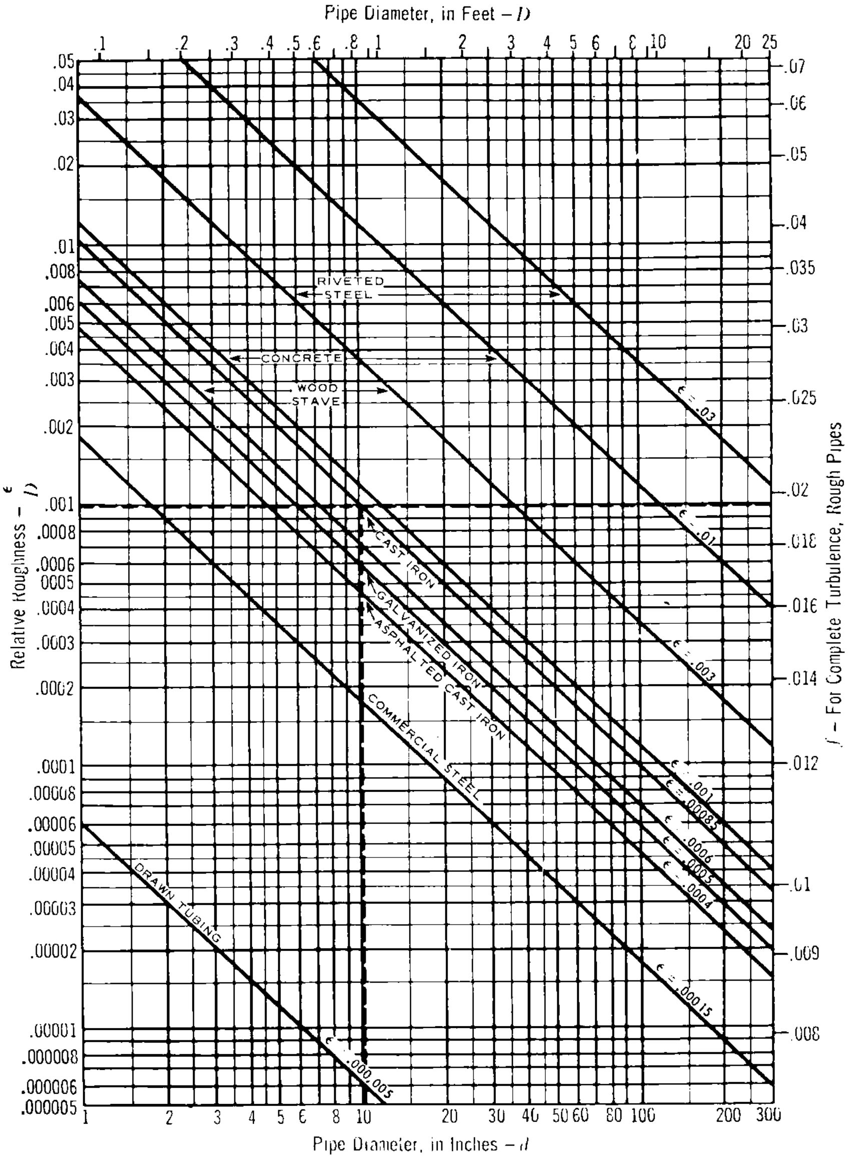

Figure 1-4. Relative roughness of pipe materials and friction factors for complete turbulence. (Courtesy of Crane Company [5]. Data from L. F. Moody, *Friction Factors for Pipe Flow,* permission of ASME.)

Equation 1-6a, which describes the friction factor for turbulent flow in pipe of any roughness, is a simple addition of the Prandtl solution for smooth pipe and the von Karman solution for rough pipe. The relationship holds for the transition between rough and smooth pipe.

To solve Equation 1-6a for the friction factor f an iterative analysis is required because the function is nonhomogeneous and inseparable. There are several empirical relations of f expressed as an independent separate function of f (ϵ/d, N_{Re}), but with today's micro-computers Equation 1-6 can be solved more accurately and expediently with iteration.

Dimensional forms of Equation 1-4 are presented in Table 1-1 [1], where the equation is conveniently shown in various units that are used to solve fluid pressure loss problems.

NON-NEWTONIAN FLUIDS

The Colebrook equation holds for fluids whose flow properties are dependent on the fluid viscosity. These fluids consist of all gases, liquids, and solutions of low molecular weight and are known as Newtonian fluids. In

Table 1-1
Dimensional Forms Used With Equation 1-4 [1]

Pressure Drop, Reynolds Number, and Velocity					Friction Factor
Flow rate	$-\Delta P_f$ or ρH_f	N_{Re}	V	H_V	If $N_{Re} < 2{,}000$, $f = 64/N_{Re}$
W	$\left(\frac{bfL}{D} + \Sigma K_i\right)\frac{aW^2}{\rho D^4}$	$\frac{cW}{\mu D}$	$\frac{dW}{\rho D^2}$	$\frac{\rho V^2}{e}$	If $N_{Re} \geqslant 2{,}000$, $f = [2 \log_{10}(0.27\ \epsilon/D + (7/N_{Re}^{0.9}))]^{-2*}$
Q	$\left(\frac{bfL}{D} + \Sigma K_i\right)\frac{a\rho Q^2}{D^4}$	$\frac{cQ\rho}{\mu D}$	$\frac{dQ}{D^2}$	$\frac{\rho V^2}{e}$	

Units and constants

	Conventional units				[Gas]	Metric units			
$-\Delta P_f(H_f)$	psi	psi	(ft)	(ft)	inH_2O[60°F]	bar	bar	Pa	(m)
W(Q)	lb/h	(gpm)	lb/h	(gpm)	(acfm)	kg/s	(L/s)	kg/s	(m^3/s)
D	in.	in.	in.	in.	in.	mm	mm	m	m
ϵ	in.	in.	in.	in.	in.	mm	mm	m	m
L	ft	ft	ft	ft	ft	m	m	m	m
ρ	lb/ft^3	lb/ft^3	lb/ft^3	lb/ft^3	lb/ft^3	kg/m^3	kg/m^3	kg/m^3	kg/m^3
μ	cp	cp	cp	cp	cp	mPa-s(cp)	mPa-s(cp)	Pa-s	Pa-s
V	ft/s	ft/s	ft/s	ft/s	ft/min	m/s	m/s	m/s	m/s
H_V	psi	psi	ft	ft	in. H_2O	bar	bar	Pa	m
a	2.799×10^{-7}	1.801×10^{-5}	4.031×10^{-5}	2.593×10^{-3}	0.02792	8.106×10^{6}	8.106	0.8106	0.08265
b	12	12	12	12	12	1,000	1,000	1	1
c	6.316	50.66	6.316	50.66	379.0	1.273×10^{6}	1.273	1.273	1.273
d	0.05093	0.4085	0.05093	0.4085	183.3	1.273×10^{6}	1.273	1.273	1.273
e	9,266	9,266	$64.35\times\rho$	$64.35\times\rho$	1.204×10^{6}	2×10^{5}	2×10^{5}	2	$19.61\times\rho$

a,b,c,d,e = constants
D = pipe diameter
f = Weisbach friction factor
H_f = frictional head loss
K = number of velocity heads
L = pipe length
$-\Delta P_f$ = frictional pressure drop
Q = volumetric flowrate
N_{Re} = Reynolds number
V = velocity
H_V = velocity head
W = mass flow rate
ϵ = pipe roughness
μ = fluid viscosity
ρ = fluid density

Newtonian fluids the viscosity alone defines the rheological behavior.

Non-Newtonian fluids are those in which the viscosity alone does not define their rheological behavior. Such fluids are solutions composed of solid particles that expand. Clay and very dense slurries are examples of non-Newtonian fluids. The flow properties of such fluids are a function of the particle characteristics, e.g., size and flexibility and thermal expansion.

Purely viscous non-Newtonian fluids are classified into three categories: time-dependent and time-independent and viscoelastic. A time-dependent fluid displays slow changes in rheological properties, such as thixotropic fluids that exhibit reversible structural changes. Several types of crude oil fit into this category. Another type of time-dependent non-Newtonian fluid is rheopectic fluids. Under constant sustained shear, these fluids' rate of structural deformation exceeds the rate of structural decay. One such category of fluids is polyester. Rheopectic fluids are less common than thixotropic fluids.

Time-independent fluids that are purely viscous are classified as pseudoplastic, dilatant, Bingham, and yield-pseudoplastic fluids. In *pseudoplastic fluids* an infinitesimal shear stress will initiate motion and the ratio of shear stress with velocity decreases with increasing velocity gradient. This type of fluid is encountered in solutions or suspensions of fine particles that form loosely bounded aggregates that can break down or reform with an increase or decrease in shear rate. Such solutions are aqueous dispersions of polyvinyl acetate and of an acrylic copolymer; aqueous solutions of sodium carboxymethyl cellulose, and of ammonium polymethacrylate; and an aqueous suspension of limestone.

In dilatant fluids an infinitesimal shear stress will start motion and the ratio of shear stress to velocity increases as the velocity is increased. A dilatant fluid is characterized by an increase in volume of a fixed amount of dispersion, such as wet sand, when subjected to a deformation that alters the interparticle distances of its constituents from their minimum-size configuration. Such fluids are titanium dioxide particles in water or sucrose solution. Dilatant fluids are much rarer than pseudoplastic fluids.

In *Bingham fluids* a finite shearing stress is required to initiate motion and there is a linear relationship between the shearing stress—after motion impends—and the velocity gradient. Such fluids include thickened hydrocarbon greases, certain asphalts, water suspensions of clay, fly ash, finely divided minerals, quartz, sewage sludge, and point systems.

Yield-pseudoplastic fluids are similar to Bingham fluids, but the relationship between the excess shearing stress after motion impends and velocity gradient is nonlinear. Fluids in this category are defined by their rheograms, where relationships between the shear stress and rate of shear exhibit a geometric convexity to the shear stress axis. Such fluids are many clay-water and similar suspensions and aqueous solutions of carboxypolymethylene (carbopol).

Viscoelastic fluids make up the last category of non-Newtonian fluids. The term "viscoelastic fluid" is applied to the most general of fluids—those that exhibit the characteristic of partial elastic recovery of the fluid structure. Whenever a viscoelastic fluid is subjected to a rapid change in deformation, elastic recoil or stress relaxation occurs. Many solutions exhibit viscoelastic properties under appropriate conditions—molten polymers, which are highly elastic; and solutions of long-charged molecules, such as polyethylene oxide and polyacrylamides. Processes such as coagulation, oil-well fracturing, and high-capacity pipelines rely on polymeric additives to cause pressure drops. Viscoelastic fluids exhibit the "Weissenberg effect," which is caused by normal stresses and produces unusual phenomena, such as the tendency of the fluid to climb up a shaft rotating in the fluid.

For any time-independent non-Newtonian fluid, Metzer and Reed [2] have developed the following generalized Reynolds number fraction:

$$N_{Re} = \frac{D^n U^{2-n} \rho}{\gamma} \tag{1-7}$$

where D = pipe ID, ft
U = average bulk velocity, ft/sec
ρ = density, lb_m/ft^3
γ = generalized viscosity coefficient, lb_m/ft sec = $g_c\, c\, 8^{n-1}$ (see Table 1-1)
C = experimentally determined flow constant, μ/g_c for a Newtonian fluid
n = empirical constant that is a function of non-Newtonian behavior (flow behavior index), 1.0 for Newtonian fluids

For $n = 1.0$ and $C = \mu/g_c$, Equation 1-7 reduces to $N_{Re} = Du\,\rho/\mu$ for Newtonian fluids. For $2{,}100 \leqslant N_{Re} \leqslant 100{,}000$ the following empirical relations can be used for determining the friction factor:

$$f = \frac{a_n}{(N'_{Re})\, b_n}$$

where $b_n = 0.0019498\,(n)^{-4.511}$
$n = (7.8958 \times 10^{-7})\,(a_n)\ 182.1321$

Typical values for γ and n are given in Table 1-2 [3]. Values for γ and n not available in literature must be determined by viscosimeter measurements.

Figure 1-5 shows the rheological classification of non-Newtonian mixtures that behave as single-phase flow. The reader is urged to refer to Govier [4] for further information on non-Newtonian fluid or other complex mixtures. Usually, the mechanical design of process systems does not involve non-Newtonian fluids, but knowledge of them and their peculiarities is a must if the need arises.

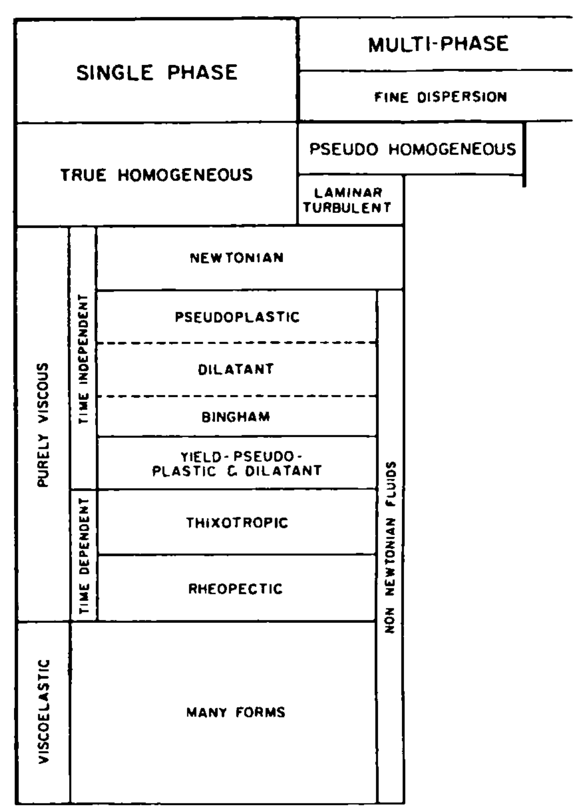

Figure 1-5. Rheological classification of complex mixtures that behave as single phase fluids [4].

Table 1-2
Rheological Constants for Some Typical Non-Newtonian Fluids* [3]

Composition of Fluid	Rheological Constants n	γ	Composition of Fluid	Rheological Constants n	γ
23.3% Illinois yellow clay in water	0.229	0.863	18.6% solids, Mississippi clay in water	0.022	0.105
0.67% carboxy-methyl-cellulose (CMC) in water	0.716	0.121	14.3% clay in water	0.350	0.0344
1.5% CMC in water	0.554	0.920	21.2% clay in water	0.335	0.0855
3.0% CMC in water	0.566	2.80	25.0% clay in water	0.185	0.204
33% lime water	0.171	0.983	31.9% clay in water	0.251	0.414
10% napalm in kerosene	0.520	1.18	36.8% clay in water	0.176	1.07
4% paper pulp in water	0.575	6.13	40.4% clay in water	0.132	2.30
54.3% cement rock in water	0.153	0.331	23% lime in water	0.178	1.04

* Reproduced by permission: A. B. Metzner and J. C. Reed, *AICHE Journal*, 1, 434 (1955).

VELOCITY HEADS

Returning to Equation 1-4, let's focus on the term ΣK_i. This term represents the *excess* velocity heads lost in fluid motion due to fluid turbulence caused by local turbulence at the pipe wall and change in flow direction. The latter is the greatest contributor to the ΣK_i term. When a fluid strikes a surface and changes flow direction, it loses momentum and, therefore, energy. Considering the 90° elbow in Figure 1-6, we see that the fluid changes direction from the x to the y direction and imparts reactions F_x and F_y, each a function of the pressure and velocity of the fluid. End conditions of the elbow determine some of the velocity head loss, that is, where the fitting is a "smooth elbow" or a "screwed elbow." A smooth elbow is one that is either flanged or welded to the pipe such that a smooth internal surface is encountered by the flow. In a screwed elbow there are abrupt changes in the wall causing local turbulence and hence increased velocity head loss.

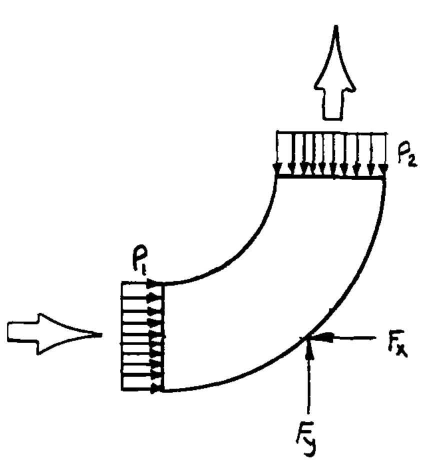

Figure 1-6. Reactions on an elbow induced by a change of flow.

Analytical determination of velocity heads can only be accomplished in a few simplified cases. The values for velocity heads must be determined and verified empirically. Comprehensive listings of such velocity head (K) values are given in Figures 1-7 [5], 1-8 [5], 1-9 [6], and 1-10. Using these values in Equation 1-4, you can analyze most cases of friction pressure drop for pipe under 24 inches in diameter. For pipe with diameter greater than 24 inches, an additional analysis must be made in solving for the velocity head term. This method, presented by Hooper [7] is called the "two-K method."

TWO-K METHOD

As explained previously, the value of K does not depend on the roughness of the fitting or the fitting size, but rather on the Reynolds number and the geometry of the fitting. The published data for single K values apply to fully-developed turbulent flow, and K is independent of N_{Re} when N_{Re} is well into the turbulent zone. As N_{Re} approaches 1,000, the value of K increases. When $N_{Re} < 1{,}000$, the value of K becomes inversely proportional to N_{Re}. In large diameter pipe (> 24 in.) the value of N_{Re} must be carefully considered if values of 1,000 or less are encountered. The two-K method accounts for this dependency in the following equation:

$$K = K_1/N_{Re} + K_\infty (1 + 1/d) \tag{1-8}$$

where K_1 = K for the fitting of $N_{Re} = 1$
K_∞ = K for a large fitting of $N_{Re} = \infty$
d = internal diameter of attached pipe, in.

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