FLUID MECHANICS (DYNAMICS)

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LECTURE 4

1- Dimensional Analysis of Pipe Flow

a. Major Losses.
b. Minor Losses.

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Dimensional Analysis of Pipe Flow

It is often necessary to determine the head loss, that occurs in a pipe flow so that the energy equation (1), can be used in the analysis of pipe flow problems.

$$\frac{p_{out}}{\gamma} + \frac{V_{out}^2}{2g} + z_{out} = \frac{p_{in}}{\gamma} + \frac{V_{in}^2}{2g} + z_{in} - h_L \quad \text{.....................(1)}$$

As shown in Fig.(1), a typical pipe system usually consists of various lengths of straight pipe interspersed with various types of components (valves, elbows, etc.). The overall head loss for the pipe system consists of the head loss due to viscous effects in the straight pipes, termed the ***major loss*** and denoted $h_{L\ major}$, and the head loss in the various pipe components, termed the ***minor loss*** and denoted $h_{L\ minor}$ That is,

$$h_L = h_{L\,\text{major}} + h_{L\,\text{minor}} \quad \text{....................(2)}$$

The head loss designations of “***major***” and “***minor***” do not necessarily reflect the relative importance of each type of loss.

For a pipe system that contains many components and a relatively short length of pipe, the minor loss may actually be larger than the major loss.

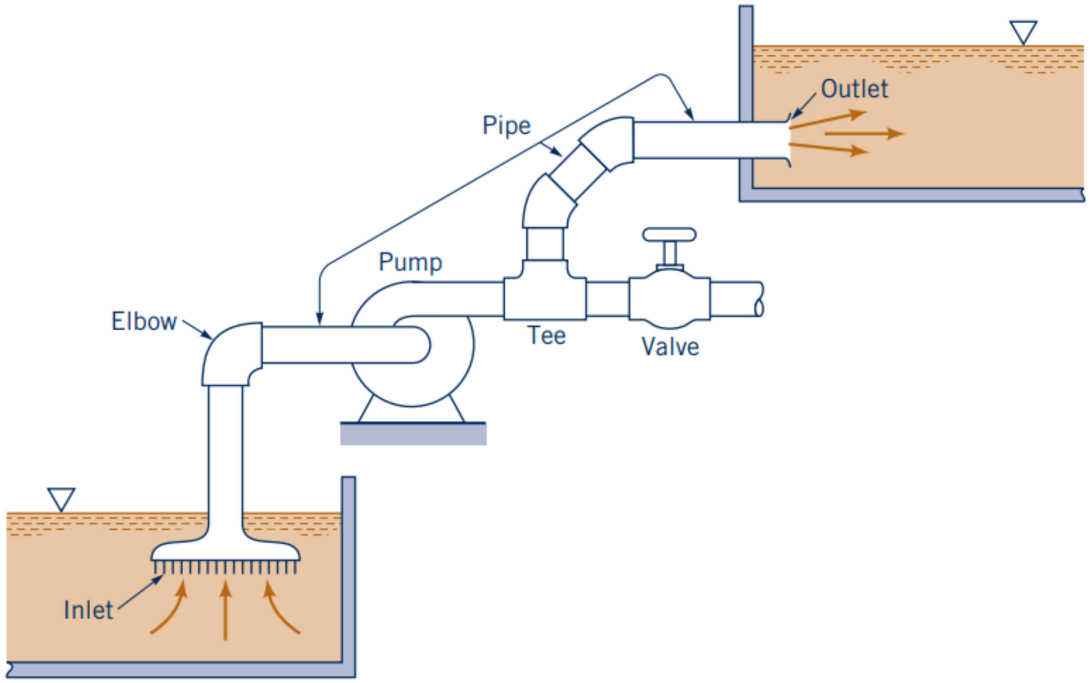
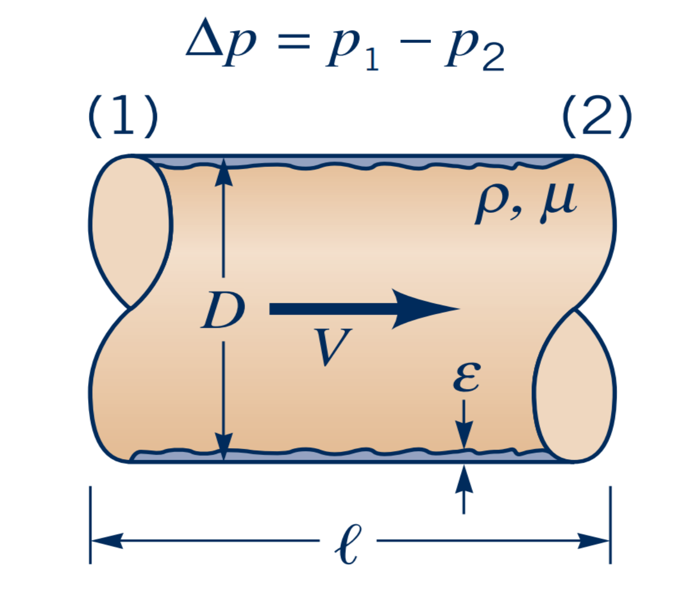


Figure 1 Typical pipe system components.

Major losses

A dimensional analysis treatment of pipe flow provides the most convenient base from which to consider turbulent, fully developed pipe flow.



The pressure drop and head loss in a pipe are dependent on the wall shear stress, τ_s between the fluid and pipe surface. A fundamental difference between laminar and turbulent flow is that the shear stress for turbulent flow is a function of the density of the fluid, **ρ**. For laminar flow, the shear stress is independent of the density, leaving the viscosity, **μ** as the only important fluid property.

Thus, as indicated by the figure in the margin, the pressure drop, for steady, incompressible turbulent flow in a horizontal round pipe of diameter D can be written in functional form as,

$$\Delta p = F\,(V, D, l, \varepsilon, \mu, \rho) \quad \text{..................(3)}$$

$$\Delta p = F(V, D, l, \varepsilon, \mu, \rho)$$

Where,

V- is the average velocity,

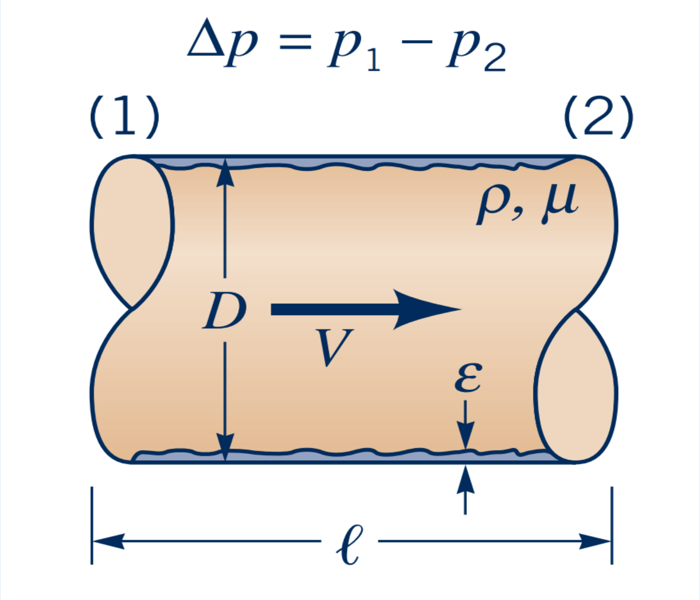
l- is the pipe length,

ε **-** is a measure of the roughness of the pipe wall.

D **–** Pipe diameter.

It is clear that ***Δp*** should be a function of ***V***, ***D***, and ***l*** .

The dependence of on the fluid properties, viscosity ***μ*** and density ***ρ*** is expected because of the dependence of on these parameters.



Although the pressure drop for laminar pipe flow is found to be independent of the roughness of the pipe, it is necessary to include this parameter when considering turbulent flow.

for turbulent flow there is a relatively thin viscous sublayer formed in the fluid near the pipe wall. In many instances this layer is very thin;

$\boldsymbol{\delta_s \ll 1}$, where $\boldsymbol{\delta_s}$ is the sublayer thickness.

If a typical wall roughness element protrudes sufficiently far into (or even through) this layer, the structure and properties of the viscous sublayer (along with Δp and τ_s) will be different than if the wall were smooth.
Thus, for turbulent flow the pressure drop is expected to be a function of the wall roughness. For laminar flow there is no thin viscous layer—viscous effects are important across the entire pipe. Thus, relatively small roughness elements have completely negligible effects on laminar pipe flow.

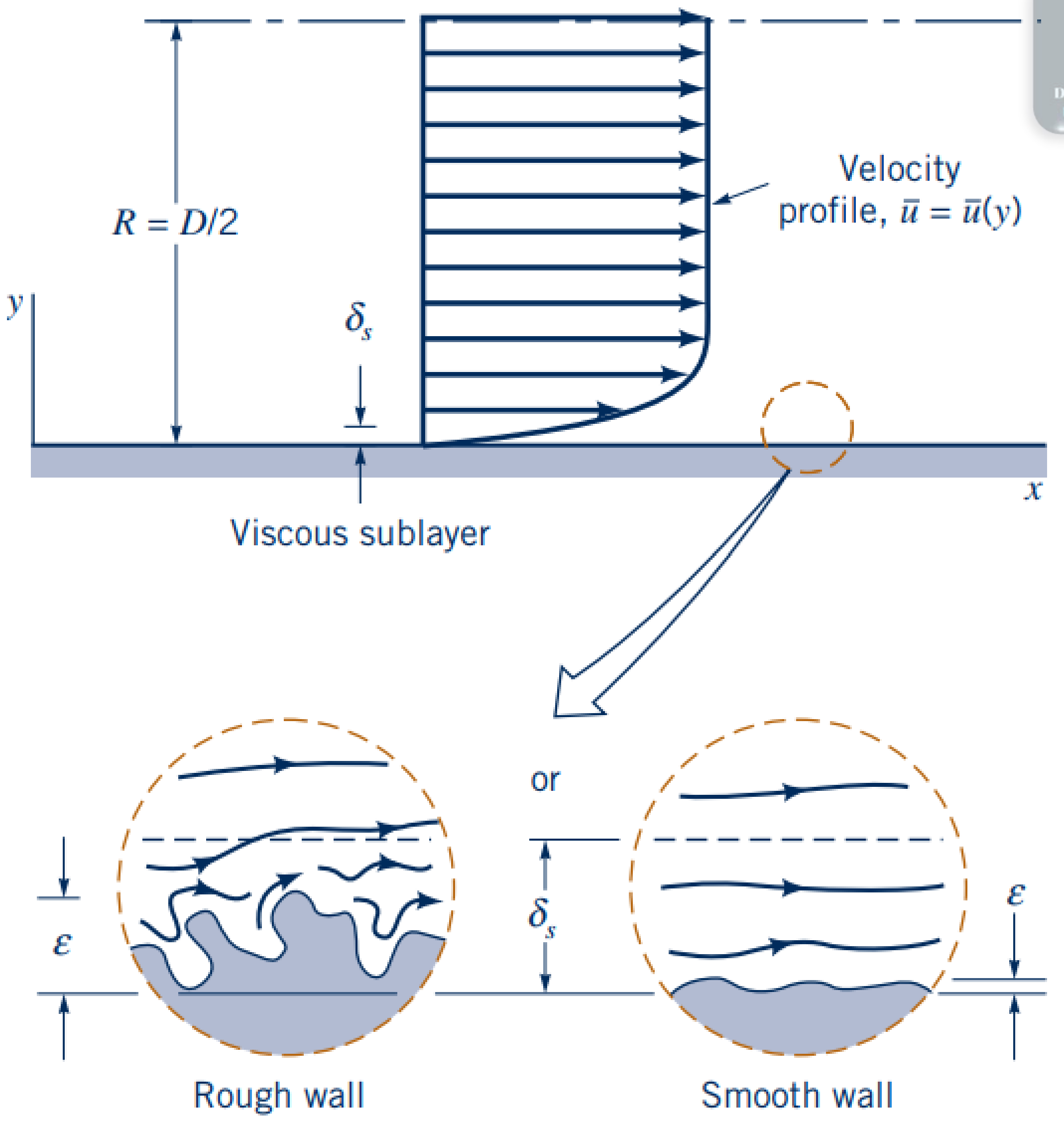


Figure 2 Flow in the viscous sublayer near rough and smooth walls.

For pipes with very large wall "roughness" (**ε/D** ≥ 0.1), such as that in corrugated pipes, the flowrate may be a function of the "roughness." We will consider only typical constant diameter pipes with relative roughness in the range ***(0 ≤ ε/D ≤ 0.05).*** Analysis of flow in corrugated pipes does not fit into the standard constant diameter pipe category, although experimental results for such pipes are available.

The list of parameters given in Eq. 3 is apparently a complete one. That is, experiments have shown that other parameters (such as surface tension, vapor pressure, etc.) do not affect the pressure drop for the conditions stated (steady, incompressible flow; round, horizontal pipe). Since there are seven variables ***(k=7)*** that can be written in terms of the three reference dimensions *MLT* ***(r = 3)*** Eq. 3 can be written in dimensionless form in terms of ***k – r = 4*** dimensionless groups. One such representation is,

$$\frac{\Delta p}{\frac{1}{2}\rho V^2} = \tilde{\phi}\left(\frac{\rho V D}{\mu}, \frac{\ell}{D}, \frac{\varepsilon}{D}\right) \quad \text{.....................(4)}$$

Re = $\frac{\rho\, V\, D}{\mu}$, Reynolds number

Relative roughness = $\frac{\varepsilon}{D}$

The functional representation can be simplified by imposing the reasonable assumption that the pressure drop should be proportional to the pipe length. Such a step is not within the realm of dimensional analysis. It is merely a logical assumption supported by experiments. The only way that this can be true is if the dependence is factored out as

$$\frac{\Delta p}{\frac{1}{2}\rho V^2} = \frac{\ell}{D}\phi\left(\mathrm{Re}, \frac{\varepsilon}{D}\right) \quad \text{..................(5)}$$

The quantity is $\Delta p D \big/ \left(\frac{l\rho V^2}{2}\right)$ termed the *friction factor*, *f*. Thus, for a horizontal pipe

$$f = \frac{\Delta p\, D}{\frac{1}{2}\, l\rho V^2} \qquad \Delta p = f\frac{\ell}{D}\frac{\rho V^2}{2} \quad \text{..................(6)}$$

Where,

$$f = \phi\left(\mathrm{Re}, \frac{\varepsilon}{D}\right) \quad \text{..................(7)}$$

For **laminar** fully developed flow, the value of ***f* = 64/Re**, is simply independent of ε/D.
For **turbulent** flow, the functional dependence of the **friction factor on the Reynolds number and the relative roughness**, is a rather complex one that cannot, as yet, be obtained from a theoretical analysis.
The results are obtained from an exhaustive set of experiments and usually resented in terms of a curve-fitting formula or the equivalent graphical form.
The energy equation for steady incompressible flow is,

$$\frac{p_1}{\gamma} + \frac{V_1^2}{2g} + z_1 = \frac{p_2}{\gamma} + \frac{V_2^2}{2g} + z_2 + h_L \quad \text{....................(8)}$$

where is the head loss between sections (1) and (2). With the **assumption** of a *constant diameter* (*$D_1 = D_2$ so that $V_1=V_2$*), so that *horizontal pipe* ($z_1=z_2$) with fully developed flow this becomes $\Delta p = p_1 - p_2 = \gamma h_L$ which can be combined with Eq. 6 to give ,

$$\Delta p = f\frac{\ell}{D}\frac{\rho V^2}{2}$$

$$\gamma = \rho g$$

$$h_{L\,\text{major}} = f\frac{\ell}{D}\frac{V^2}{2g} \quad \text{....................(9)}$$

Equation 9, called the ***Darcy–Weisbach equation***, is valid for any *fully developed, steady, incompressible pipe flow*—whether the pipe is horizontal or on a hill. On the other hand, Eq. 6
is valid only for horizontal pipes. In general, with the energy equation gives

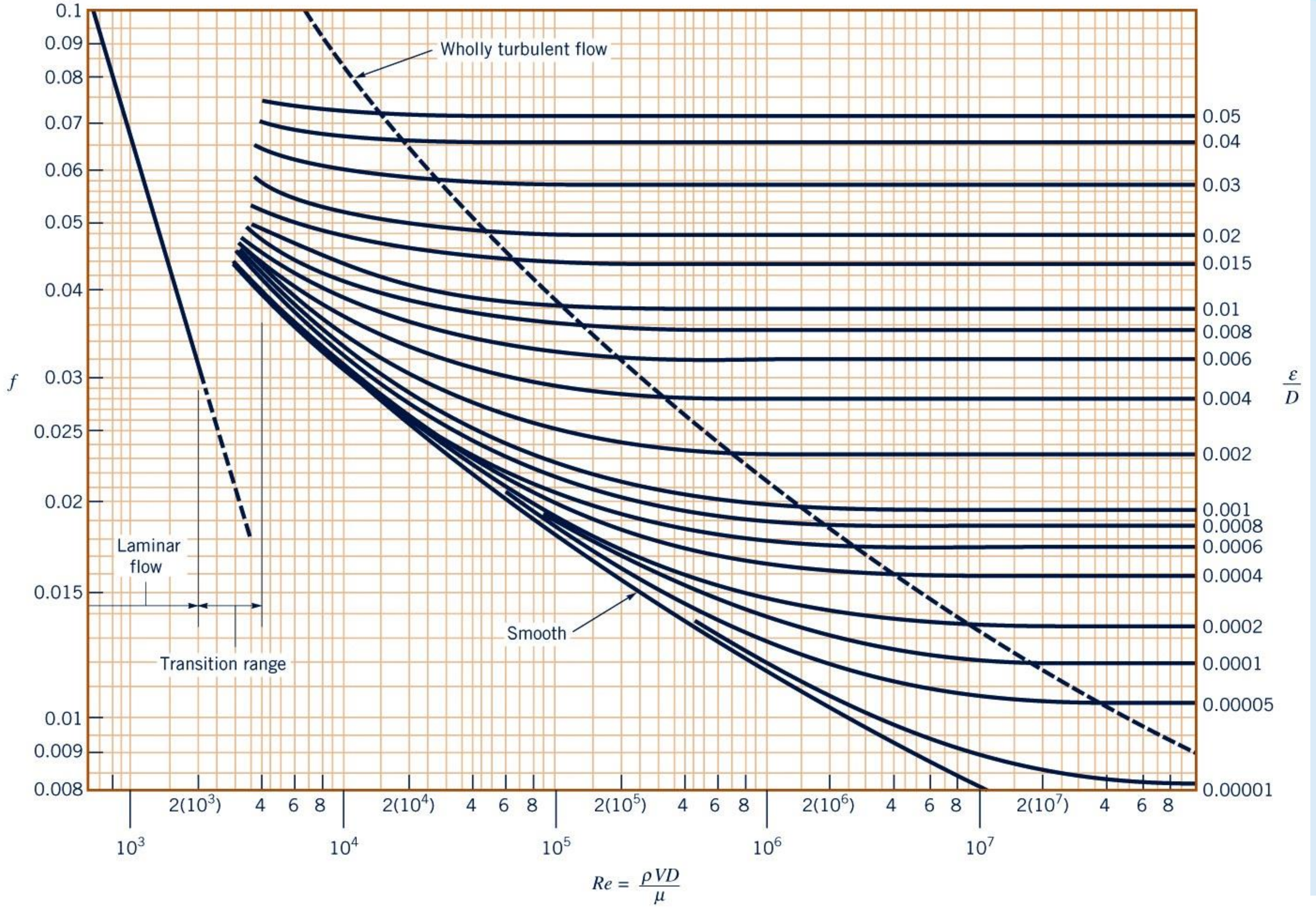
$$p_1 - p_2 = \gamma(z_2 - z_1) + \gamma h_L = \gamma(z_2 - z_1) + f\frac{\ell}{D}\frac{\rho V^2}{2}$$ ………….(10

Part of the pressure change is due to the elevation change and part is due to the head loss associated with frictional effects, which are given in terms of the friction factor, *f*.

Figure 3 shows the functional dependence of *f* on Re and is called the ***Moody chart.***
The Moody chart, is universally valid for all steady, fully developed, incompressible pipe flows.

The figure provide the correct correlation for $f = \emptyset\left(Re, \frac{\varepsilon}{D}\right)$.

Typical roughness values for various pipe surfaces are given in Table (1)



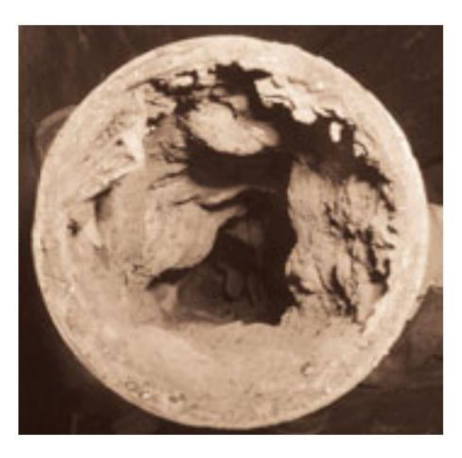
■ **Figure 3** **Friction factor as a function of Reynolds number and relative roughness for round pipes—the Moody chart.**

Table (1)

Equivalent Roughness for New Pipes [Adapted from Moody]

	Equivalent Roughness, ε	
Pipe	**Feet**	**Millimeters**
Riveted steel	0.003–0.03	0.9–9.0
Concrete	0.001–0.01	0.3–3.0
Wood stave	0.0006–0.003	0.18–0.9
Cast iron	0.00085	0.26
Galvanized iron	0.0005	0.15
Commercial steel or wrought iron	0.00015	0.045
Drawn tubing	0.000005	0.0015
Plastic, glass	0.0 (smooth)	0.0 (smooth)

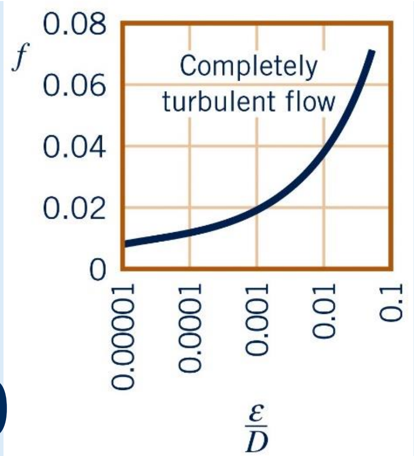
It is important to observe that the values of relative roughness given pertain to new, clean pipes. After considerable use, most pipes (because of a buildup of corrosion or scale) may have a relative roughness that is considerably larger (perhaps by an order of magnitude) than that given. As shown by the figure below, very old pipes may have enough scale buildup to not only alter the value of but also to change the



The figure provide the correct correlation for $f = \emptyset \left(Re, \frac{\varepsilon}{D}\right)$.

For laminar flow, f = 64/Re which is independent of relative roughness. For turbulent flows with very large Reynolds numbers $f = \emptyset \left(\frac{\varepsilon}{D}\right)$, which, as shown by the figure below, is independent of the Reynolds number.

For such flows, commonly termed ***completely turbulent flow*** (or wholly turbulent flow), the laminar sublayer is so thin (its thickness decreases with increasing Re) that the surface roughness completely dominates the character of the flow near the wall.



2100 < Re < 4000

Laminar , Transition , Turbulent

The following equation from **Colebrook** is valid for the entire nonlaminar range of the Moody chart

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\mathrm{Re}\sqrt{f}}\right)$$ ....(11.a)

The Moody chart is a graphical representation of this equation, which is an empirical fit of the pipe flow pressure drop data. Equation 11 is called the ***Colebrook formula*****.**

The **Haaland** equation, which is easier to use, is given by

$$\frac{1}{\sqrt{f}} = -1.8 \log\left[\left(\frac{\varepsilon/D}{3.7}\right)^{1.11} + \frac{6.9}{\mathrm{Re}}\right]$$ ....(11.b)

Table (2)

Approximate Physical Properties of Some Common Gases at Standard Atmospheric Pressure (BG Units)

Gas	Temperature (°F)	Density, ρ (slugs/ft^3)	Specific Weight, γ (lb/ft^3)	Dynamic Viscosity, μ (lb · s/ft^2)	Kinematic Viscosity, ν (ft^2/s)	Gas Constant,[a] R (ft · lb/slug · °R)	Specific Heat Ratio,[b] k
Air (standard)	59	2.38 E − 3	7.65 E − 2	3.74 E − 7	1.57 E − 4	1.716 E + 3	1.40
Carbon dioxide	68	3.55 E − 3	1.14 E − 1	3.07 E − 7	8.65 E − 5	1.130 E + 3	1.30
Helium	68	3.23 E − 4	1.04 E − 2	4.09 E − 7	1.27 E − 3	1.242 E + 4	1.66
Hydrogen	68	1.63 E − 4	5.25 E − 3	1.85 E − 7	1.13 E − 3	2.466 E + 4	1.41
Methane (natural gas)	68	1.29 E − 3	4.15 E − 2	2.29 E − 7	1.78 E − 4	3.099 E + 3	1.31
Nitrogen	68	2.26 E − 3	7.28 E − 2	3.68 E − 7	1.63 E − 4	1.775 E + 3	1.40
Oxygen	68	2.58 E − 3	8.31 E − 2	4.25 E − 7	1.65 E − 4	1.554 E + 3	1.40

[a]Values of the gas constant are independent of temperature.
[b]Values of the specific heat ratio depend only slightly on temperature.

Approximate Physical Properties of Some Common Gases at Standard Atmospheric Pressure (SI Units)

Gas	Temperature (°C)	Density, ρ (kg/m^3)	Specific Weight, γ (N/m^3)	Dynamic Viscosity, μ (N · s/m^2)	Kinematic Viscosity, ν (m^2/s)	Gas Constant,[a] R (J/kg · K)	Specific Heat Ratio,[b] k
Air (standard)	15	1.23 E + 0	1.20 E + 1	1.79 E − 5	1.46 E − 5	2.869 E + 2	1.40
Carbon dioxide	20	1.83 E + 0	1.80 E + 1	1.47 E − 5	8.03 E − 6	1.889 E + 2	1.30
Helium	20	1.66 E − 1	1.63 E + 0	1.94 E − 5	1.15 E − 4	2.077 E + 3	1.66
Hydrogen	20	8.38 E − 2	8.22 E − 1	8.84 E − 6	1.05 E − 4	4.124 E + 3	1.41
Methane (natural gas)	20	6.67 E − 1	6.54 E + 0	1.10 E − 5	1.65 E − 5	5.183 E + 2	1.31
Nitrogen	20	1.16 E + 0	1.14 E + 1	1.76 E − 5	1.52 E − 5	2.968 E + 2	1.40
Oxygen	20	1.33 E + 0	1.30 E + 1	2.04 E − 5	1.53 E − 5	2.598 E + 2	1.40

[a]Values of the gas constant are independent of temperature.
[b]Values of the specific heat ratio depend only slightly on temperature.

Table (3)

Approximate Physical Properties of Some Common Liquids (BG Units)

Liquid	Temperature (°F)	Density, ρ (slugs/ft^3)	Specific Weight, γ (lb/ft^3)	Dynamic Viscosity, μ (lb · s/ft^2)	Kinematic Viscosity, ν (ft^2/s)	Surface Tension,[a] σ (lb/ft)	Vapor Pressure, p_v [lb/in.^{2} (abs)]	Bulk Modulus,[b] E_v (lb/in.^{2})
Carbon tetrachloride	68	3.09	99.5	2.00 E − 5	6.47 E − 6	1.84 E − 3	1.9 E + 0	1.91 E + 5
Ethyl alcohol	68	1.53	49.3	2.49 E − 5	1.63 E − 5	1.56 E − 3	8.5 E − 1	1.54 E + 5
Gasoline[c]	60	1.32	42.5	6.5 E − 6	4.9 E − 6	1.5 E − 3	8.0 E + 0	1.9 E + 5
Glycerin	68	2.44	78.6	3.13 E − 2	1.28 E − 2	4.34 E − 3	2.0 E − 6	6.56 E + 5
Mercury	68	26.3	847	3.28 E − 5	1.25 E − 6	3.19 E − 2	2.3 E − 5	4.14 E + 6
SAE 30 oil[c]	60	1.77	57.0	8.0 E − 3	4.5 E − 3	2.5 E − 3	—	2.2 E + 5
Seawater	60	1.99	64.0	2.51 E − 5	1.26 E − 5	5.03 E − 3	2.56 E − 1	3.39 E + 5
Water	60	1.94	62.4	2.34 E − 5	1.21 E − 5	5.03 E − 3	2.56 E − 1	3.12 E + 5

[a]In contact with air.
[b]Isentropic bulk modulus calculated from speed of sound.
[c]Typical values. Properties of petroleum products vary.

Approximate Physical Properties of Some Common Liquids (SI Units)

Liquid	Temperature (°C)	Density, ρ (kg/m^3)	Specific Weight, γ (kN/m^3)	Dynamic Viscosity, μ (N · s/m^2)	Kinematic Viscosity, ν (m^2/s)	Surface Tension,[a] σ (N/m)	Vapor Pressure, p_v [N/m^2 (abs)]	Bulk Modulus,[b] E_v (N/m^2)
Carbon tetrachloride	20	1,590	15.6	9.58 E − 4	6.03 E − 7	2.69 E − 2	1.3 E + 4	1.31 E + 9
Ethyl alcohol	20	789	7.74	1.19 E − 3	1.51 E − 6	2.28 E − 2	5.9 E + 3	1.06 E + 9
Gasoline[c]	15.6	680	6.67	3.1 E − 4	4.6 E − 7	2.2 E − 2	5.5 E + 4	1.3 E + 9
Glycerin	20	1,260	12.4	1.50 E + 0	1.19 E − 3	6.33 E − 2	1.4 E − 2	4.52 E + 9
Mercury	20	13,600	133	1.57 E − 3	1.15 E − 7	4.66 E − 1	1.6 E − 1	2.85 E + 10
SAE 30 oil[c]	15.6	912	8.95	3.8 E − 1	4.2 E − 4	3.6 E − 2	—	1.5 E + 9
Seawater	15.6	1,030	10.1	1.20 E − 3	1.17 E − 6	7.34 E − 2	1.77 E + 3	2.34 E + 9
Water	15.6	999	9.80	1.12 E − 3	1.12 E − 6	7.34 E − 2	1.77 E + 3	2.15 E + 9

EXAMPLE Comparison of Laminar or Turbulent Pressure Drop

GIVEN Air under standard conditions flows through a 4.0-mm-diameter drawn tubing with an average velocity of $V = 50$ m/s. For such conditions the flow would normally be turbulent. However, if precautions are taken to eliminate disturbances to the flow (the entrance to the tube is very smooth, the air is dust free, the tube does not vibrate, etc.), it may be possible to maintain laminar flow.

FIND **(a)** Determine the pressure drop in a 0.1-m section of the tube if the flow is laminar.

(b) Repeat the calculations if the flow is turbulent.

SOLUTION

Under standard temperature and pressure conditions the density and viscosity are $\rho = 1.23 \text{ kg/m}^3$ and $\mu = 1.79 \times 10^{-5} \text{ N} \cdot \text{s/m}^2$. Thus, the Reynolds number is

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(1.23 \text{ kg/m}^3)(50 \text{ m/s})(0.004 \text{ m})}{1.79 \times 10^{-5} \text{ N} \cdot \text{s/m}^2} = 13{,}700$$

which would normally indicate turbulent flow.

(a) If the flow were laminar, then $f = 64/\text{Re} = 64/13{,}700 = 0.00467$, and the pressure drop in a 0.1-m-long horizontal section of the pipe would be

$$\Delta p = f \frac{\ell}{D} \frac{1}{2} \rho V^2$$

$$= (0.00467) \frac{(0.1 \text{ m})}{(0.004 \text{ m})} \frac{1}{2} (1.23 \text{ kg/m}^3)(50 \text{ m/s})^2$$

or

$$\Delta p = 0.179 \text{ kPa} \qquad \textbf{(Ans)}$$

(b) If the flow were turbulent, then $f = \phi(\text{Re}, \varepsilon/D)$, where from Table 1, $\varepsilon = 0.0015$ mm so that $\varepsilon/D = 0.0015 \text{ mm}/4.0 \text{ mm} = 0.000375$. From the Moody chart with $\text{Re} = 1.37 \times 10^4$ and $\varepsilon/D = 0.000375$ we obtain $f = 0.028$. Thus, the pressure drop in this case would be approximately

$$\Delta p = f \frac{\ell}{D} \frac{1}{2} \rho V^2 = (0.028) \frac{(0.1 \text{ m})}{(0.004 \text{ m})} \frac{1}{2} (1.23 \text{ kg/m}^3)(50 \text{ m/s})^2$$

or

$$\Delta p = 1.076 \text{ kPa} \qquad \textbf{(Ans)}$$

COMMENT A considerable savings in effort to force the fluid through the pipe could be realized (0.179 kPa rather than 1.076 kPa) if the flow could be maintained as laminar flow at this Reynolds number. In general this is very difficult to do, although laminar flow in pipes has been maintained up to Re ≈ 100,000 in rare instances.

An alternate method to determine the friction factor for the turbulent flow would be to use the Colebrook formula, Eq. 11.a Thus,

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\mathrm{Re}\sqrt{f}}\right) = -2.0 \log\left(\frac{0.000375}{3.7} + \frac{2.51}{1.37 \times 10^4\sqrt{f}}\right)$$

or

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(1.01 \times 10^{-4} + \frac{1.83 \times 10^{-4}}{\sqrt{f}}\right) \qquad \textbf{(1)}$$

By using a root-finding technique on a computer or calculator, the solution to Eq. 1 is determined to be $f = 0.0291$, in agreement (within the accuracy of reading the graph) with the Moody chart method of $f = 0.028$.

Equation 11.b provides an alternate form to the Colebrook formula that can be used to solve for the friction factor directly.

$$\frac{1}{\sqrt{f}} = -1.8 \log\left[\left(\frac{\varepsilon/D}{3.7}\right)^{1.11} + \frac{6.9}{\mathrm{Re}}\right] = -1.8 \log\left[\left(\frac{0.000375}{3.7}\right)^{1.11} + \frac{6.9}{1.37 \times 10^4}\right]$$

$$= 0.0289$$

This agrees with the Colebrook formula and Moody chart values obtained above.

Numerous other empirical formulas can be found in the literature (Ref. 5) for portions of the Moody chart. For example, an often-used equation, commonly referred to as the Blasius formula, for turbulent flow in smooth pipes ($\varepsilon/D = 0$) with $\mathrm{Re} < 10^5$ is

$$f = \frac{0.316}{\mathrm{Re}^{1/4}}$$

For our case this gives

$$f = 0.316(13{,}700)^{-0.25} = 0.0292$$

which is in agreement with the previous results. Note that the value of f is relatively insensitive to ε/D for this particular situation. Whether the tube was smooth glass ($\varepsilon/D = 0$) or the drawn tubing ($\varepsilon/D = 0.000375$) would not make much difference in the pressure drop. For this flow, an increase in relative roughness by a factor of 30 to $\varepsilon/D = 0.0113$ (equivalent to a commercial steel surface; see Table 8.1) would give $f = 0.043$. This would represent an increase in pressure drop and head loss by a factor of $0.043/0.0291 = 1.48$ compared with that for the original drawn tubing.

The pressure drop of 1.076 kPa in a length of 0.1 m of pipe corresponds to a change in absolute pressure [assuming $p =$ 101 kPa (abs) at $x = 0$] of approximately $1.076/101 = 0.0107$, or about 1%. Thus, the incompressible flow assumption on which the above calculations (and all of the formulas in this chapter) are based is reasonable. However, if the pipe were 2-m long the pressure drop would be 21.5 kPa, approximately 20% of the original pressure. In this case the density would not be approximately constant along the pipe, and a compressible flow analysis would be needed. Such considerations are discussed in Chapter 11.

Minor losses

As discussed in the previous section, the head loss in long, straight sections of pipe, the major losses, can be calculated by use of the friction factor obtained from either the Moody chart or the Colebrook equation.

Most pipe systems, however, consist of considerably more than straight pipes. These additional components (valves, bends, tees, and the like) add to the overall head loss of the system. Such losses are generally termed ***minor losses***, with the corresponding head loss denoted $h_{L_{minor}}$. In this section we indicate how to determine the various minor losses that commonly occur in pipe systems.

The head loss associated with flow through a valve is a common minor loss. The purpose of a valve is to provide a means to regulate the flowrate. This is accomplished by changing the geometry of the system (i.e., closing or opening the valve alters the flow pattern through the valve), which in turn alters the losses associated with the flow through the valve. The flow resistance or head loss through the valve may be a significant portion of the resistance in the system.

The flow pattern through a typical component such as a valve is shown in Fig. 4.

The most common method used to determine these head losses or pressure drops is to specify the ***loss coefficient*, K_L** which is defined as

$$h_{L\,\text{minor}} = K_L \frac{V^2}{2g} \quad \text{...........(12)}$$

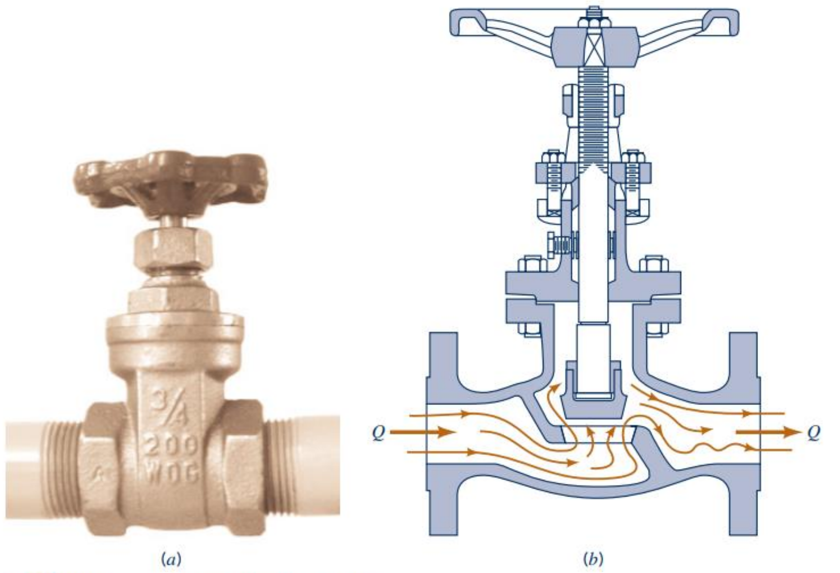


Figure 4 Flow through a valve.

The actual value of is strongly dependent on the geometry of the component considered. It may also be dependent on the fluid prop

$$K_L = \phi(\text{geometry}, \text{Re})$$

Minor losses are sometimes given in terms of an ***equivalent length*, l_{eq}** In this terminology, the head loss through a component is given in terms of the equivalent length of pipe that would produce the same head loss as the

$$h_{L\,\text{minor}} = K_L \frac{V^2}{2g} = f \frac{\ell_{eq}}{D} \frac{V^2}{2g}$$

$$\ell_{eq} = \frac{K_L D}{f}$$

where *D* and *f* are based on the pipe containing the component.

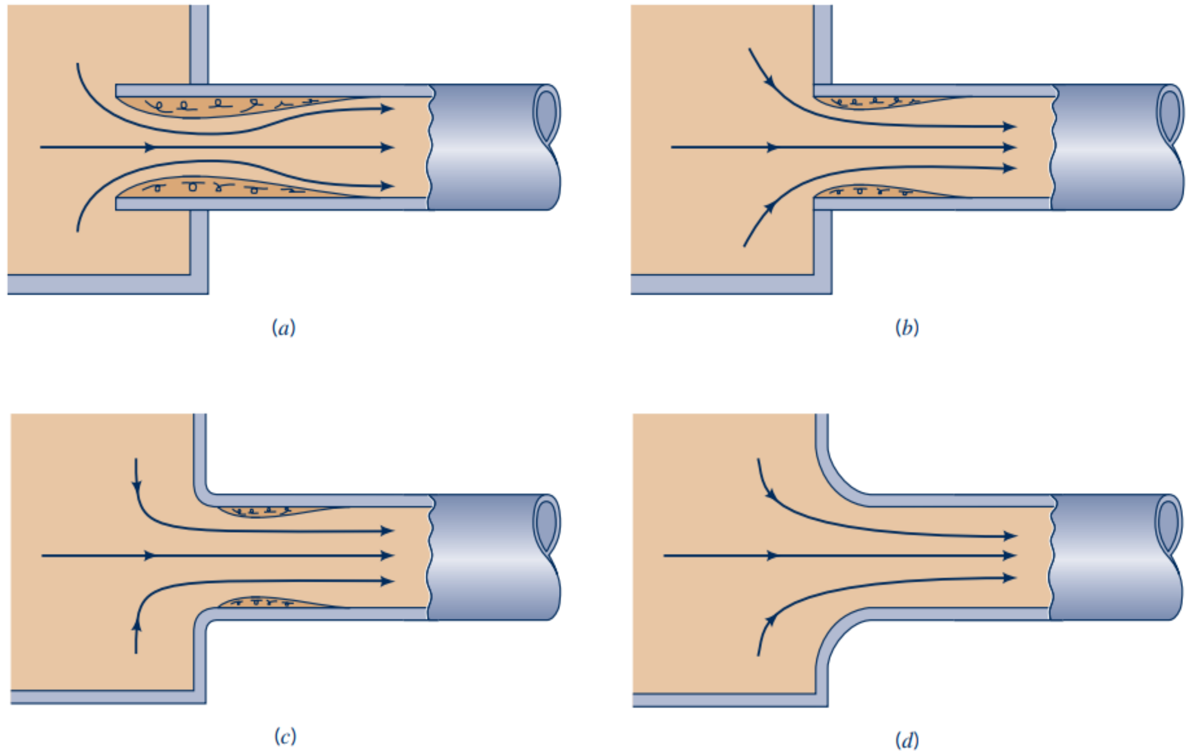


Figure 5 Entrance flow conditions and loss coefficient
(*a*) Reentrant, $K_L = 0.8$, (*b*) sharp-edged, $K_L = 0.5$, (*c*) slightly rounded, $K_L = 0.2$
(*d*) well-rounded, $K_L = 0.04$

The head loss of the pipe system is the same as that produced in a straight pipe whose length is equal to the pipes of the original system plus the sum of the additional equivalent lengths of all of the components of the system. Most pipe flow analyses, including those in this book, use the loss coefficient method rather than the equivalent length method to determine the minor losses.

Many pipe systems contain various transition sections in which the pipe diameter changes from one size to another. Such changes may occur abruptly or rather smoothly through some type of area change section. Any change in flow area contributes losses that are not accounted for in the fully developed head loss calculation (the friction factor). The extreme cases involve flow into a pipe from a reservoir (an entrance) or out of a pipe into a reservoir (an exit).

Energy Type		
Kinetic	Potential	Pressure
$\rho V^2/2$	γz	p

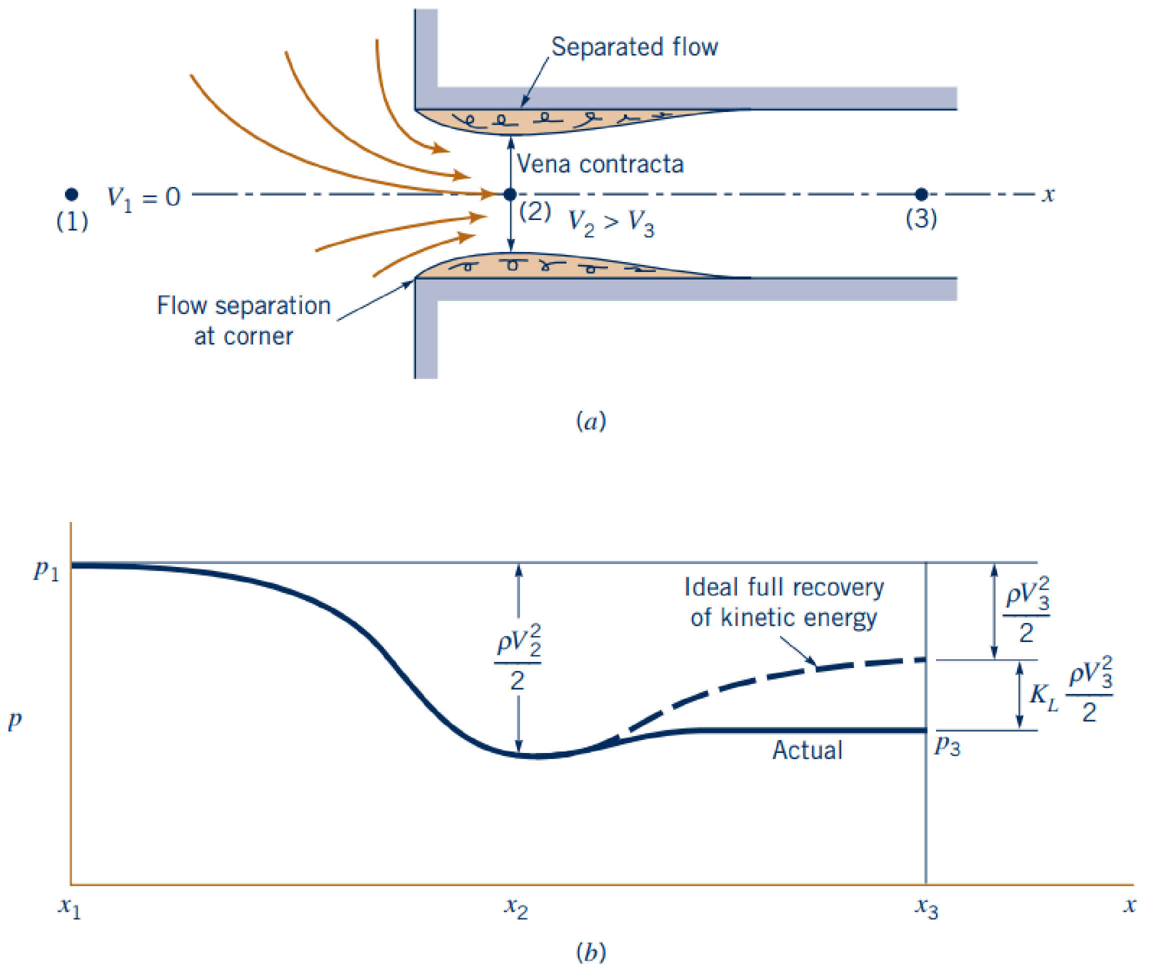


Figure 6 Flow pattern and pressure distribution for a sharp-edged entrance.

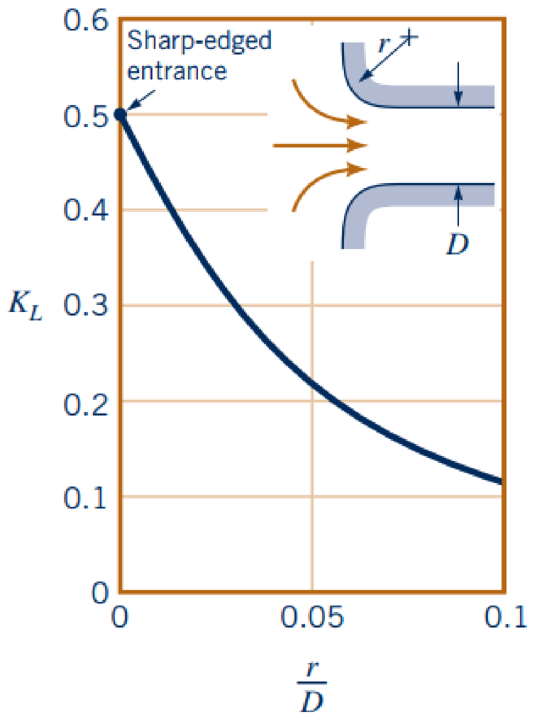
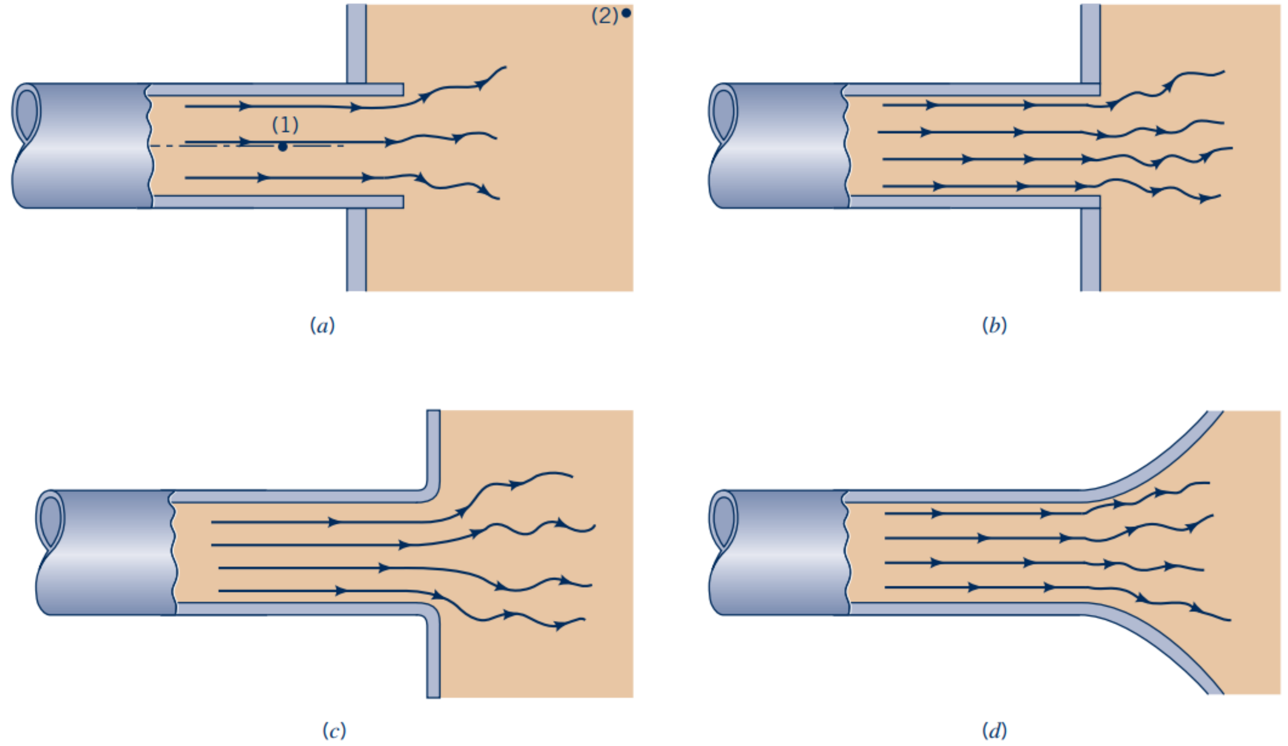


Figure 7 Entrance loss coefficient as a function of rounding of the inlet edge

A head loss (the exit loss) is also produced when a fluid flows from a pipe into a tank as is shown in Fig. 8

In these cases the entire kinetic energy of the exiting fluid (velocity V_1) is dissipated through viscous effects as the stream of fluid mixes with the fluid in the tank and eventually comes to rest ($V_2 = 0$). The exit loss from points (1) and (2) is therefore equivalent to one velocity head, $\boldsymbol{K_l=1}$.



■ Figure 8 Exit flow conditions and loss coefficient.
(*a*) Reentrant, $K_L = 1.0$, (*b*) sharp-edged, $K_L = 1.0$, (*c*) slightly rounded, $K_L = 1.0$, (*d*) well-rounded, $K_L = 1.0$.

Losses also occur because of a change in pipe diameter as is shown in Figs. 9 and 10.

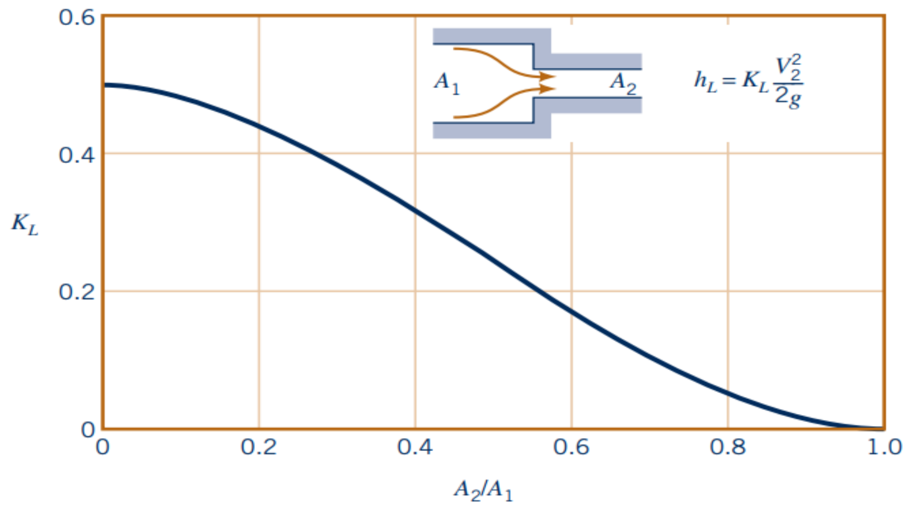


Figure 9 Loss coefficient for a sudden contraction

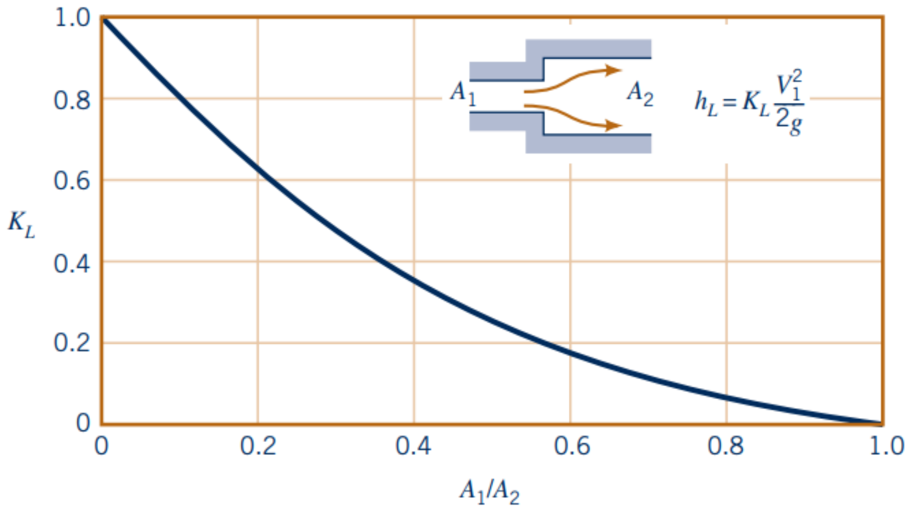


Figure 10 Loss coefficient for a sudden expansion

The sharp-edged entrance and exit flows discussed in the previous paragraphs are limiting cases of this type of flow with either ***A_1/A_2= ∞, or A_1/A_2 =0*** respectively. The loss coefficient for a sudden contraction, $K_L = h_L/(\frac{V_2^2}{2g})$ is a function of the area ratio, ***A_2/A_1*** as is shown in Fig. 9 . The value of ***K_L*** changes gradually from one extreme of a sharp-edged entrance (***A_2/A_1=0*** with ***K_L=0.5***) to the other extreme of no area change (***A_2/A_1=1*** with ***K_L=0***)

In many ways, the flow in a sudden expansion is similar to exit flow. As is indicated in Fig. 11, the fluid leaves the smaller pipe and initially forms a jet-type structure as it enters the larger pipe.

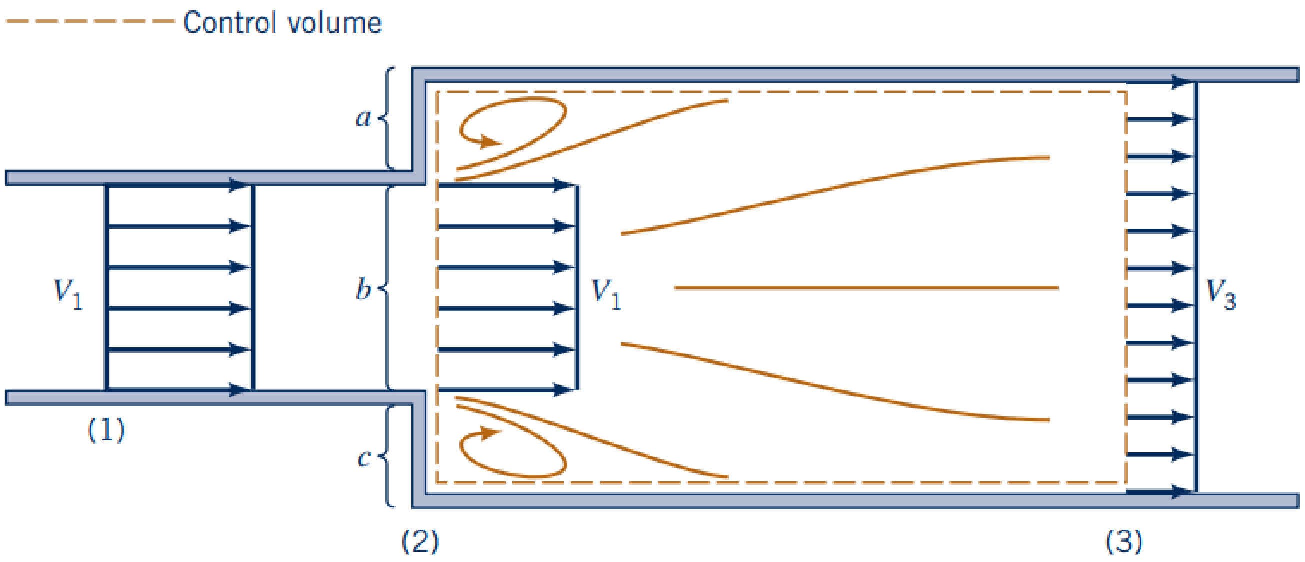


Figure 11 Control volume used to calculate the loss coefficient for a sudden expansion.

In this process [between sections (2) and (3)] a portion of the kinetic energy of the fluid is dissipated as a result of viscous effects. A square edged exit is the limiting case with $A_1/A_2=0$

From continuity and momentum equations for the control volume shown in Fig. 11 and the energy equation applied between (2) and (3). We assume that the flow is uniform at sections (1), (2), and (3) and the pressure is constant across the left-hand side of the control volume (***$p_a = p_b = p_c = p_1$***). The resulting three governing equations (mass, momentum, and energy) are

$$A_1 V_1 = A_3 V_3$$

$$p_1 A_3 - p_3 A_3 = \rho A_3 V_3 (V_3 - V_1)$$

And

$$\frac{p_1}{\gamma} + \frac{V_1^2}{2g} = \frac{p_3}{\gamma} + \frac{V_3^2}{2g} + h_L$$

These can be rearranged to give the loss coefficient,

as $K_L = h_L / (\frac{V_2^2}{2g})$ as:

$$K_L = \left(1 - \frac{A_1}{A_2}\right)^2$$

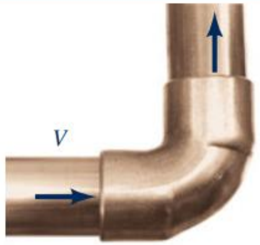
Where we have used the fact that ***$A_2 = A_3$***

As with so many minor loss situations, it is not the viscous effects directly (i.e., the wall shear stress) that cause the loss. Rather, it is the dissipation of kinetic energy (another type of viscous effect) as the fluid decelerates inefficiently.

Table (4)

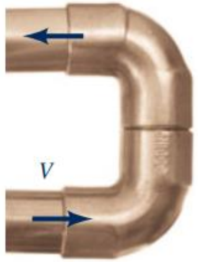
Loss Coefficients for Pipe Components $\left(h_L = K_L \frac{V^2}{2g}\right)$ **(Data from Refs. 5, 10, 27)**

Component	K_L
a. Elbows	
Regular 90°, flanged	0.3
Regular 90°, threaded	1.5
Long radius 90°, flanged	0.2
Long radius 90°, threaded	0.7
Long radius 45°, flanged	0.2
Regular 45°, threaded	0.4
b. 180° return bends	
180° return bend, flanged	0.2
180° return bend, threaded	1.5
c. Tees	
Line flow, flanged	0.2
Line flow, threaded	0.9
Branch flow, flanged	1.0
Branch flow, threaded	2.0
d. Union, threaded	0.08
***e. Valves**	
Globe, fully open	10
Angle, fully open	2
Gate, fully open	0.15
Gate, $\frac{1}{4}$ closed	0.26
Gate, $\frac{1}{2}$ closed	2.1
Gate, $\frac{3}{4}$ closed	17
Swing check, forward flow	2
Swing check, backward flow	∞
Ball valve, fully open	0.05
Ball valve, $\frac{1}{3}$ closed	5.5
Ball valve, $\frac{2}{3}$ closed	210



90° elbow

45° elbow



180° return bend

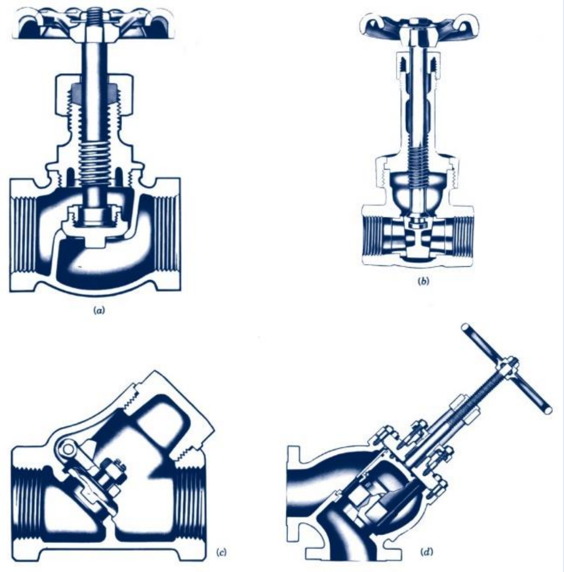


Figure 12 Internal structure of various valves: (*a*) globe valve, (*b*) gate valve, (*c*) swing check valve, (*d*) stop check valve. (Courtesy of Crane Co., Fluid Handling Division.)

Tee

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