Moody Diagram and Head Losses

Learning Objectives:
- Explain what major and minor losses are and what causes them
- Evaluate major losses in pipe networks
- Evaluate minor losses in pipe networks

Motivational Question:
- How do we calculate $h_L$ if it is not given--how do we do design?
Turbulent Flow in Pipes:

Conservation of Mass (steady, in comp.)

\[ A_1 V_1 = A_2 V_2 \]

if \( A_1 = A_2 \) \( \rightarrow \) velocity profiles unchanged.

Does anything change?

YES: because of friction.

Friction erodes driving force

\[ \Rightarrow \Delta p = f(V, D, l, \varepsilon, \mu, \rho) \]

Dimensional Analysis:

\[ \frac{\Delta p}{\frac{1}{2} \rho V^2} = f\left( \frac{\varepsilon V D}{\mu}, \frac{l}{D}, \frac{\varepsilon}{D} \right) \]

\( \varepsilon \) ratio of driving force to kinetic energy.

Physics:

Re: tells us the intensity level of the turbulence.

\( \frac{\varepsilon}{D} \): tells us whether roughness pierces vsl and to what degree.

\( \frac{l}{D} \): for steady flow, reasonable that \( \Delta p \) linearly with \( l \)

\[ \frac{\Delta p}{\frac{1}{2} \rho V^2} = \frac{l}{D} f\left( Re, \frac{\varepsilon}{D} \right) \]
Friction Factor:

\[ f = \frac{\Delta p D}{l \frac{gV^2}{2}} \]

Thus:

\[ \Delta p = f \frac{l}{D} \frac{gV^2}{2} \]

\[ f = \phi(Re, \frac{e}{D}) \]

In terms of head loss:

\[ h_L = f \frac{l}{D} \frac{V^2}{2g} \]

* Darcy-Weisbach Eqn.

Moody Diagram:

\[ f = \phi(Re, \frac{e}{D}) \] is found by experiments.

Nikuradse: performed exp.
Moody & Colebrook developed chart.

Colebrook formula:

\[ \frac{1}{\sqrt{f}} = -2.0 \log \left( \frac{e/D}{3.7} + \frac{2.51}{Re \sqrt{f}} \right) \]

\[ \text{implicit in } f \Rightarrow \text{requires root finding.} \]
FIGURE 8.20  Friction factor as a function of Reynolds number and relative roughness for round pipes—the Moody chart. (Data from Ref. 7 with permission.)
Head Losses in Pipes:

Major losses: friction

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

Minor losses: local disruptions

\[ h_L = K \frac{v^2}{2g} ; \quad K = \Phi (Re, geometry) \]

- Entrance / Exit
- Contraction / Expansion
- Bends and fittings
- Valves

Examples.
wall shear stress within the entrance region. The net effect is that the loss coefficient for a square-edged entrance is approximately $K_L = 0.50$. One-half of a velocity head is lost as the fluid enters the pipe. If the pipe protrudes into the tank (a reentrant entrance) as is shown in Fig. 8.22a, the losses are even greater.

An obvious way to reduce the entrance loss is to round the entrance region as is shown in Fig. 8.22c, thereby reducing or eliminating the vena contracta effect. Typical values for the loss coefficient for entrances with various amounts of rounding of the lip are shown in Fig. 8.24. A significant reduction in $K_L$ can be obtained with only slight rounding.

A head loss (the exit loss) is also produced when a fluid flows from a pipe into a tank as is shown in Fig. 8.25. 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 ($V_3 = 0$). The exit loss from points (1) and (2) is therefore equivalent to one velocity head, or $K_L = 1$. 

![Figure 8.23](image1)  
**Figure 8.23** Flow pattern and pressure distribution for a sharp-edged entrance.

Pipe entrance losses can be reduced relatively easily by rounding the inlet.
Losses also occur because of a change in pipe diameter as is shown in Figs. 8.26 and 8.27. The sharp-edged entrance and exit flows discussed in the previous paragraphs are limiting cases of this type of flow with either $A_1/A_2 = \infty$, or $A_1/A_2 = 0$, respectively. The loss coefficient for a sudden contraction, $K_L = h_s/(V_s^2/2g)$, is a function of the area ratio, $A_2/A_1$, as is shown in Fig. 8.26. The value of $K_L$ changes gradually from one extreme of a sharp-edged entrance ($A_2/A_1 = 0$ with $K_L = 0.50$) 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. 8.28, the fluid leaves the smaller pipe and initially forms a jet-type structure as it enters the larger pipe. Within a few diameters downstream of the expansion, the jet becomes dispersed across the pipe, and fully developed flow becomes established again. 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 $A_1/A_2 = 0$. 

\[ K_L = \frac{h_s}{V_s^2/2g} \]
A sudden expansion is one of the few components (perhaps the only one) for which the loss coefficient can be obtained by means of a simple analysis. To do this we consider the continuity and momentum equations for the control volume shown in Fig. 8.28 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

\[
A_1 V_1 = A_3 V_3
\]

\[
p_1 A_1 - p_3 A_3 = \rho A_3 V_3 (V_3 - V_1)
\]

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, \( K_L = h_L / (V_3^2 / 2g) \), as

\[
K_L = \left( 1 - \frac{A_1}{A_3} \right)^2
\]

where we have used the fact that \( A_2 = A_3 \). This result, plotted in Fig. 8.27, is in good agreement with experimental data. 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.
The losses may be quite different if the contraction or expansion is gradual. Typical results for a conical diffuser with a given area ratio, $A_2/A_1$, are shown in Fig. 8.29. (A diffuser is a device shaped to decelerate a fluid.) Clearly the included angle of the diffuser, $\theta$, is a very important parameter. For very small angles, the diffuser is excessively long and most of the head loss is due to the wall shear stress as in fully developed flow. For moderate or large angles, the flow separates from the walls and the losses are due mainly to a dissipation of the kinetic energy of the jet leaving the smaller diameter pipe. In fact, for moderate or large values of $\theta$ (i.e., $\theta > 35^\circ$ for the case shown in Fig. 8.29), the conical diffuser is, perhaps unexpectedly, less efficient than a sharp-edged expansion which has $K_L = (1 - A_1/A_2)^2$. There is an optimum angle ($\theta \approx 8^\circ$ for the case illustrated) for which the loss coefficient is a minimum. The relatively small value of $\theta$ for the minimum $K_L$ results in a long diffuser and is an indication of the fact that it is difficult to efficiently decelerate a fluid.

It must be noted that the conditions indicated in Fig. 8.29 represent typical results only. Flow through a diffuser is very complicated and may be strongly dependent on the area ratio $A_2/A_1$, specific details of the geometry, and the Reynolds number. The data are often presented in terms of a pressure recovery coefficient, $C_p = (p_2 - p_1)/(\rho V_1^2/2)$, which is the ratio of the static pressure rise across the diffuser to the inlet dynamic pressure. Considerable effort has gone into understanding this important topic (Refs. 11, 12).

Flow in a conical contraction (a nozzle; reverse the flow direction shown in Fig. 8.29) is less complex than that in a conical expansion. Typical loss coefficients based on the downstream (high-speed) velocity can be quite small, ranging from $K_L = 0.02$ for $\theta = 30^\circ$, to $K_L = 0.07$ for $\theta = 60^\circ$, for example. It is relatively easy to accelerate a fluid efficiently.

Bends in pipes produce a greater head loss than if the pipe were straight. The losses are due to the separated region of flow near the inside of the bend (especially if the bend is sharp) and the swirling secondary flow that occurs because of the imbalance of centripetal forces as a result of the curvature of the pipe centerline. These effects and the associated values of $K_L$ for large Reynolds number flows through a 90° bend are shown in Fig. 8.30. The friction loss due to the axial length of the pipe bend must be calculated and added to that given by the loss coefficient of Fig. 8.30.

For situations in which space is limited, a flow direction change is often accomplished by use of miter bends, as is shown in Fig. 8.31, rather than smooth bends. The considerable losses in such bends can be reduced by the use of carefully designed guide vanes that help direct the flow with less unwanted swirl and disturbances.

![Figure 8.29](image-url)  
**Figure 8.29** Loss coefficient for a typical conical diffuser (Ref. 5).
Another important category of pipe system components is that of commercially available pipe fittings such as elbows, tees, reducers, valves, and filters. The values of $K_L$ for such components depend strongly on the shape of the component and only very weakly on the Reynolds number for typical large Re flows. Thus, the loss coefficient for a 90° elbow depends on whether the pipe joints are threaded or flanged but is, within the accuracy of the data, fairly independent of the pipe diameter, flow rate, or fluid properties (the Reynolds number effect). Typical values of $K_L$ for such components are given in Table 8.2. These typical components are designed more for ease of manufacturing and costs than for reduction of the head losses that they produce. The flowrate from a faucet in a typical house is sufficient whether the value of $K_L$ for an elbow is the typical $K_L = 1.5$, or it is reduced to $K_L = 0.2$ by use of a more expensive long-radius, gradual bend (Fig. 8.30).

Valves control the flowrate by providing a means to adjust the overall system loss coefficient to the desired value. When the valve is closed, the value of $K_L$ is infinite and no fluid flows. Opening of the valve reduces $K_L$, producing the desired flowrate. Typical cross sections of various types of valves are shown in Fig. 8.32. Some valves (such as the
or

\[ h_{L,\text{minor}} = K_L \frac{V^2}{2g} \]  \hspace{1cm} (8.36)

The pressure drop across a component that has a loss coefficient of \( K_L = 1 \) is equal to the dynamic pressure, \( \rho V^2/2 \).

The actual value of \( K_L \) is strongly dependent on the geometry of the component considered. It may also be dependent on the fluid properties. That is,

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

where \( \text{Re} = \rho V D/\mu \) is the pipe Reynolds number. For many practical applications the Reynolds number is large enough so that the flow through the component is dominated by inertia effects, with viscous effects being of secondary importance. This is true because of the relatively large accelerations and decelerations experienced by the fluid as it flows along a rather curved, variable area (perhaps even torturous) path through the component (see Fig. 8.21). In a flow that is dominated by inertia effects rather than viscous effects, it is usually found that pressure drops and head losses correlate directly with the dynamic pressure. This is the reason why the friction factor for very large Reynolds number, fully developed pipe flow is independent of the Reynolds number. The same condition is found to be true for flow through pipe components. Thus, in most cases of practical interest the loss coefficients for components are a function of geometry only, \( K_L = \phi(\text{geometry}) \).

Minor losses are sometimes given in terms of an equivalent length, \( \ell_{eq} \). 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 component. That is,

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

or

\[ \ell_{eq} = \frac{K_L D}{f} \]
**Figure 8.32** Internal structure of various valves: (a) globe valve, (b) gate valve, (c) swing check valve, (d) stop check valve. (Courtesy of Crane Co., Valve Division.)

**Figure 8.33** Head loss in a valve is due to dissipation of the kinetic energy of the large-velocity fluid near the valve seat.
conventional globe valve) are designed for general use, providing convenient control between the extremes of fully closed and fully open. Others (such as a needle valve) are designed to provide very fine control of the flowrate. The check valve provides a diode type operation that allows fluid to flow in one direction only.

Loss coefficients for typical valves are given in Table 8.2. As with many system components, the head loss in valves is mainly a result of the dissipation of kinetic energy of a high-speed portion of the flow. This is illustrated in Fig. 8.33.

## Table 8.2
Loss Coefficients for Pipe Components \( h_L = K_L \frac{v^2}{2g} \) (Data from Refs. 5, 10, 27)

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

*See Fig. 8.32 for typical valve geometry.*
that used for circular pipes, Eq. 8.6) is a partial differential equation rather than an ordinary
differential equation. Although the equation is linear (for fully developed flow the convective
acceleration is zero), its solution is not as straightforward as for round pipes. Typically the
velocity profile is given in terms of an infinite series representation (Ref. 17).

Practical, easy-to-use results can be obtained as follows. Regardless of the cross-sectional
shape, there are no inertia effects in fully developed laminar pipe flow. Thus, the friction fac-
tor can be written as \( f = C/Re_h \), where the constant \( C \) depends on the particular shape of
the duct, and \( Re_h \) is the Reynolds number, \( Re_h = \rho V D_h / \mu \), based on the hydraulic diameter. The
**hydraulic diameter** defined as \( D_h = 4A / P \) is four times the ratio of the cross-sectional flow
area divided by the wetted perimeter, \( P \), of the pipe as is illustrated in Fig. 8.34. It represents
a characteristic length that defines the size of a cross section of a specified shape. The factor
of 4 is included in the definition of \( D_h \) so that for round pipes the diameter and hydraulic di-
neter are equal \([ D_h = 4A / P = 4(\pi D^2 / 4) / (\pi D) = D ] \). The hydraulic diameter is also used in
the definition of the friction factor, \( h_L = f (\ell / D_h) V^2 / 2g \), and the relative roughness, \( \varepsilon / D_h \).

The values of \( C = f Re_h \) for laminar flow have been obtained from theory and/or ex-
periment for various shapes. Typical values are given in Table 8.3 along with the hydraulic

![Diagram](image1)

![Diagram](image2)

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**TABLE 8.3**

<table>
<thead>
<tr>
<th>Shape</th>
<th>Parameter</th>
<th>( C = f Re_h )</th>
</tr>
</thead>
<tbody>
<tr>
<td>I. Concentric Annulus</td>
<td>( D_1 / D_2 )</td>
<td>71.8</td>
</tr>
<tr>
<td>( D_h = D_2 - D_1 )</td>
<td>0.0001</td>
<td>80.1</td>
</tr>
<tr>
<td></td>
<td>0.01</td>
<td>89.4</td>
</tr>
<tr>
<td></td>
<td>0.1</td>
<td>95.6</td>
</tr>
<tr>
<td></td>
<td>0.6</td>
<td>96.0</td>
</tr>
<tr>
<td></td>
<td>1.00</td>
<td>96.0</td>
</tr>
<tr>
<td>II. Rectangle</td>
<td>( a/b )</td>
<td>96.0</td>
</tr>
<tr>
<td>( D_h = \frac{2ab}{a+b} )</td>
<td>0</td>
<td>96.0</td>
</tr>
<tr>
<td></td>
<td>0.05</td>
<td>89.9</td>
</tr>
<tr>
<td></td>
<td>0.10</td>
<td>84.7</td>
</tr>
<tr>
<td></td>
<td>0.25</td>
<td>72.9</td>
</tr>
<tr>
<td></td>
<td>0.50</td>
<td>62.2</td>
</tr>
<tr>
<td></td>
<td>0.75</td>
<td>57.9</td>
</tr>
<tr>
<td></td>
<td>1.00</td>
<td>56.9</td>
</tr>
</tbody>
</table>
8.77 The pressure at section (2) shown in Fig. P8.77 is not to fall below 60 psi when the flowrate from the tank varies from 0 to 1.0 cfs and the branch line is shut off. Determine the minimum height, \( h \), of the water tank under the assumption that (a) minor losses are negligible, (b) minor losses are not negligible.

\[
\frac{Q^2}{2g} + \frac{V^2}{2g} + Z_1 = \frac{Q^2}{2g} + \frac{V^2}{2g} + Z_2 + \left( \frac{fL}{D} + \sum K_L \right) \frac{V^2}{2g}, \text{ where } p_1 = 0, V_1 = 0, Z_1 = 16 \text{ ft} + h, \text{ and } Z_2 = 0.
\]

Thus, with \( V = V_2 \)

\[16 + h = \frac{Q^2}{2g} + \frac{V^2}{2g} + \left( \frac{fL}{D} + \sum K_L \right) \frac{V^2}{2g}.\]

Note: \( h \) must be no less than that with

\[p_{min} = 60 \text{ psi and } p_{max} = 1 \text{ cfs}, \text{ or }\]

\[V_2 = V = \frac{Q}{A} = \frac{1}{4} \left( \frac{6 \text{ ft}}{4 \text{ ft}} \right) = 5.09 \text{ ft}^2 \]

Hence,

\[h = -16 + \frac{(60 \text{ psi})(224 \text{ in}^2)}{62.4 + \frac{6}{A}} + \left[ 1 + \left( \frac{f(L + 600 + 900)}{D} \right) + \sum K_L \right] \frac{5.09 \text{ ft}^2}{2(32.2 \text{ ft}^2)}
\]

or

\[h = 122.5 + \left[ 1 + \left( \frac{1506 + h}{0.5} \right) + \sum K_L \right] (0.402) \text{ ft, where } h \approx \text{ ft}
\]

With \( \frac{c}{D} = 0 \) and \( Re = \frac{VD}{\nu} = \frac{(5.09 \text{ ft}^2)(0.5 \text{ ft})}{1.21 \times 10^{-5} \text{ ft}^2} = 2.10 \times 10^{-5} \) we obtain

\( f = 0.0155 \) (see Fig. 8.20)

a) Neglect minor losses (\( \sum K_L = 0 \)):

From Eq. (1)

\[h = 122.5 + \left[ 1 + (0.0155) \left( \frac{1506 + h}{0.5} \right) \right] (0.402)
\]

or

\[h = 143 \text{ ft}
\]

b) Include minor losses:

\[\sum K_L = K_{\text{entrance}} + 15 K_{\text{elbow}} + K_{\text{tee}} = 0.5 + 15 (0.3) + 0.2 = 5.2\]

(see Table 8.2, assume flanged fittings)

Thus, from Eq. (1)

\[h = 122.5 + \left[ 1 + (0.0155) \left( \frac{1506 + h}{0.5} \right) + 5.2 \right] (0.402)
\]

or

\[h = 146 \text{ ft}
\]

Note: For this case minor losses are not very important.
Repeat Problem 8.77 with the assumption that the branch line is open so that half of the flow from the tank goes into the branch, and half continues in the main line.

For the flow from (1) to (2):
\[ \frac{Q_1^2}{2g} + \frac{V_1^2}{2g} + z_1 = \frac{Q_a^2}{2g} + \frac{V_a^2}{2g} + z_2 + \left( f_a \frac{L_a}{D_a} + \sum K_{L_a} \right) \frac{V_a^2}{2g} + \left( f_b \frac{L_b}{D_b} + \sum K_{L_b} \right) \frac{V_b^2}{2g} \]

(1)

where \((a)\) and \((b)\) denote pipes “a” and “b” as indicated in the figure.

Thus, with \(a_i = 0\), \(V_i = 0\), \(z_i = 16 + h\), \(z_2 = 0\), and \(p_2 = 60 \text{ psi}\). Also,

\[ V_a = \frac{Q_a}{A_a} = \frac{1 \text{ ft}^3}{\text{sec} \cdot \frac{1}{4} \text{ ft}^2} = 5.09 \text{ ft/s}, \quad V_b = \frac{Q_b}{A_b} = \frac{0.5 \text{ ft}^3}{\text{sec} \cdot \frac{1}{4} \text{ ft}^2} = 2.55 \text{ ft/s} \]

Eq. (1) becomes

\[ 16 + h = \frac{(60 \text{ ft}^3)}{62.4 \text{ lb} \cdot \text{ft}} + \left( 1 + f_a \left( \frac{h + 6 + 600}{6} \right) \right) \frac{5.09^2 \text{ ft}^2}{2(32.2 \text{ ft}^2)} + \left( f_b \left( \frac{900}{25} \right) \right) \frac{2.55^2 \text{ ft}^2}{2(32.2 \text{ ft}^2)} \]

or

\[ h = 122.5 + \left( 1 + f_a \left( \frac{606 + h}{0.5} \right) + \sum K_{L_a}(0.402) + (1800 f_b + \sum K_{L_b})(0.101) \right) \]

(2)

With \(f_b = 0\), \(Re_a = \frac{V_a D_a}{\nu} = \frac{(5.09 \text{ ft/s})(\frac{1}{4} \text{ ft})}{1.2 \times 10^{-5} \frac{\text{ ft}^2}{\text{ s}}} = 2.10 \times 10^5\), and

\[ Re_b = \frac{V_b D_b}{\nu} = \frac{1}{2} Re_a = 1.05 \times 10^5 \]

we obtain \(f_a = 0.0155\) and \(f_b = 0.0175\) (Fig. 8.20)

a) Neglect minor losses (\(\sum K_{L_a} = \sum K_{L_b} = 0\));

From Eq. (2)

\[ h = 122.5 + \left( 1 + (0.0155) \left( \frac{606 + h}{0.5} \right) \right)(0.402) + (1800(0.0175)(0.101) \]

or

\[ h = 135 \text{ ft} \]

b) Include minor losses:

\[ \sum K_{L_a} = K_{L\text{entrance}} + 15 K_{L\text{ elbow}} = 0.5 + 15(0.3) = 5.0 \] (see Table 8.2; assume flanged fittings)

and

\[ \sum K_{L_b} = K_{L\text{tee}} = 0.2 \]

From Eq. (2)

\[ h = 122.5 + \left( 1 + (0.0155) \left( \frac{606 + h}{0.5} \right) \right)(5.0)(0.402) + (1800(0.0175) + 0.2)(0.101) \]

or

\[ h = 137 \text{ ft} \]

Note: For this case minor losses are not very important.