Chapter

6

Oil Systems Piping

Introduction

Oil systems piping includes those pipelines that transport oil and petroleum products from refineries and tank farms to storage facilities and end-user locations. We will discuss calculations that are required for sizing crude oil and petroleum products (diesel, gasoline, etc.) pipelines. Since oil is generally considered incompressible and therefore its volume does not change appreciably with pressure, its analysis is similar to that of other incompressible fluids such as water. We will begin our discussion with an exploration of the properties of crude oil and petroleum products and how they affect pipeline transportation. We will also cover pumping requirements such as the type of equipment and horsepower needed to transport these products from the various sources to their destinations. We will discuss short piping systems such as oil gathering lines as well as long-distance trunk lines. Throughout this chapter we will use the term *petroleum products* to refer to crude oil as well as refined petroleum products such as gasoline, kerosene, and diesel fuels.

6.1 Density, Specific Weight, and Specific Gravity

The *density* of a liquid is defined as its mass per unit volume. The *specific weight* is defined as weight per unit volume. Sometimes these two terms are used interchangeably. Density is expressed as slug/ft^3 and specific weight as lb/ft^3 in English, or U.S. Customary (USCS), units. For example, a typical crude oil may have a density of 1.65 slug/ft^3 and a specific weight of 53.0 lb/ft^3. In comparison water has a density of 1.94 slug/ft^3 and a specific weight of 62.4 lb/ft^3. Both the density and specific weight of petroleum products change with temperature. These two properties decrease as the temperature is increased, and vice versa.

The *volume* of a petroleum product is measured in gallons or barrels in USCS units and in cubic meters (m^3) or liters (L) in Système International (SI) units. One barrel of a petroleum product is equal to 42 U.S. gallons. Volume flow rates in oil pipelines are generally reported in gal/min, barrels per hour (bbl/h), or bbl/day in USCS units and in m^3/h or L/s in SI units. As indicated before, since liquids are incompressible, pressure has little effect on their volume or density.

Specific gravity is a measure of how heavy a liquid is compared to water at a particular temperature. Thus considering some standard temperature such as 60°F, if the density of petroleum product is 6 lb/gal and that of water is 8.33 lb/gal, we can say that the specific gravity Sg of the petroleum product is

$$\text{Sg} = \frac{6}{8.33} = 0.72$$

Note that this comparison must use densities measured at the same temperature; otherwise it is meaningless. In USCS units, the standard temperature and pressure are taken as 60°F and 14.7 psi. In SI units the corresponding values are 15°C and 1 bar or 101 kPa. Typical specific gravities of common crude oils, diesel, gasoline, etc., are listed in Table 6.1.

In the petroleum industry a commonly used term is the *API gravity*, named after the American Petroleum Institute (API). The API gravity of a petroleum product is measured in the laboratory using the ASTM D1298 method. It is a measure of how heavy a liquid is compared to water and therefore has a correlation with specific gravity. However, the API scale of gravity is based on a temperature of 60°F and an API gravity of 10 for water. Liquids lighter than water have an API gravity greater than 10. Those liquids that are heavier than water will have

TABLE 6.1 Specific Gravities of Petroleum Products

Liquid	Specific Gravity at 60°F	API Gravity at 60°F
Propane	0.5118	N/A
Butane	0.5908	N/A
Gasoline	0.7272	63.0
Kerosene	0.7796	50.0
Diesel	0.8398	37.0
Light crude	0.8348	38.0
Heavy crude	0.8927	27.0
Very heavy crude	0.9218	22.0
Water	1.0000	10.0

N/A = not applicable.

an API gravity of less than 10. In comparison the specific gravity of a liquid lighter than water may be 0.85 compared to water with a specific gravity of 1.0. Similarly, brine, a heavier liquid, has a specific gravity of 1.26. It can thus be seen that the API gravity numbers increase as the product gets lighter than water whereas specific gravity numbers decrease. The API gravity is always measured at 60°F. It is incorrect to state that the API of a liquid is 37°API at 70°F. The phrase "37°API" automatically implies the temperature of measurement is 60°F.

The specific gravity of a liquid and its API gravity are related by the following two equations:

$$\mathrm{Sg} = \frac{141.5}{131.5 + \mathrm{API}} \tag{6.1}$$

$$\mathrm{API} = \frac{141.5}{\mathrm{Sg}} - 131.5 \tag{6.2}$$

Again, it must be remembered that in both Eqs. (6.1) and (6.2) the specific gravity Sg is the value at 60°F since by definition the API is always at 60°F. Thus, given the value of API gravity of a petroleum product we can easily calculate the corresponding specific gravity at 60°F using these equations.

Example 6.1

(a) A sample of crude oil when tested in a lab showed an API gravity of 35. What is the specific gravity of this crude oil?

(b) Calculate the API gravity of gasoline, if its specific gravity is 0.736 at 60°F.

Solution

(a) Using Eq. (6.1),

$$\mathrm{Sg} = \frac{141.5}{131.5 + 35} = 0.8498 \text{ at } 60^\circ\mathrm{F}$$

(b) Using Eq. (6.2),

$$\mathrm{API} = \frac{141.5}{0.736} - 131.5 = 60.76$$

It is understood that the above API value is at 60°F.

The specific gravity of a petroleum product decreases with an increase in temperature. Therefore, if the specific gravity of crude oil is 0.895 at 60°F, when the oil is heated to 100°F, the specific gravity will drop to some lower value, such as 0.825. The API gravity, on the other hand, still remains at the same value as before, since it is always referred to at 60°F.

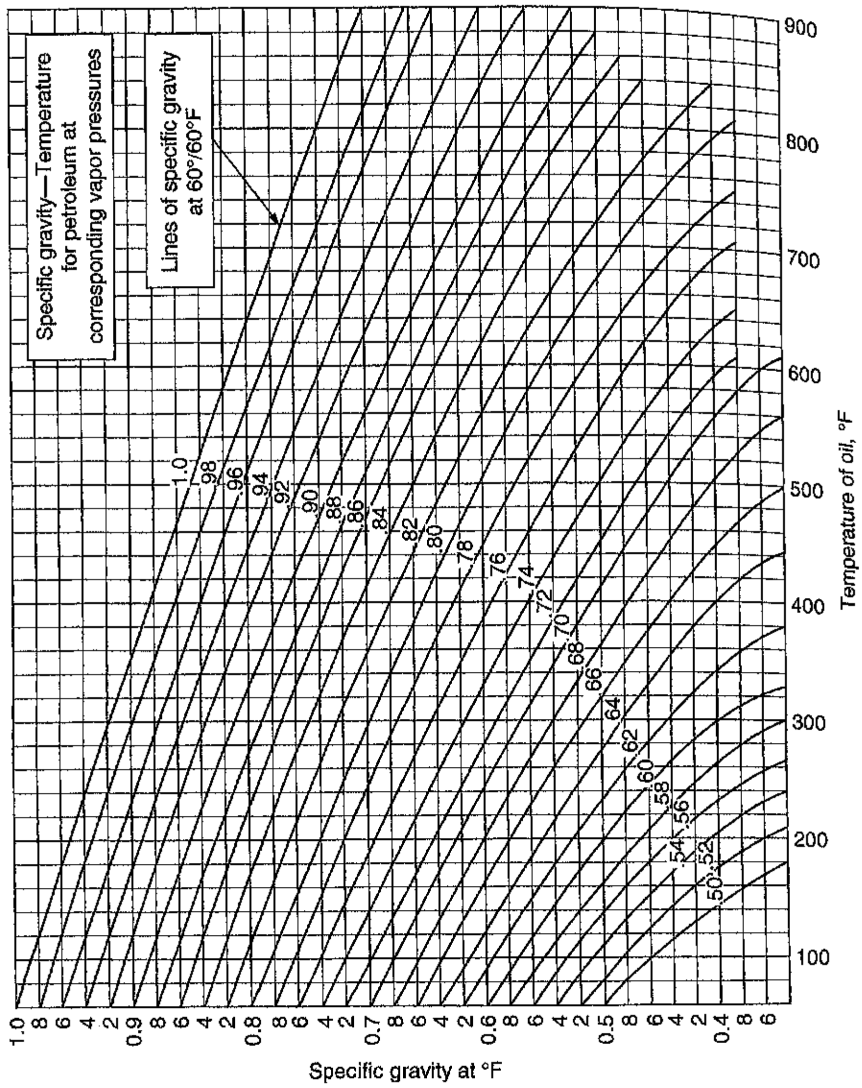


Figure 6.1 Variation of specific gravity with temperature for various petroleum liquids.

Let Sg_1 and Sg_2 represent the specific gravity at two different temperatures T_1 and T_2. We find that an approximately linear relationship exists between specific gravity and temperature within the normal range of temperatures encountered in oil pipelines. Thus a probable relationship between the specific gravity and temperature may be expressed as

$$Sg_1 - Sg_2 = a(T_2 - T_1) + b \tag{6.3}$$

where a and b are constants.

It is more common to calculate the specific gravity of a petroleum product at any temperature from the specific gravity at the standard temperature of 60°F. We can then rewrite Eq. (6.3) in terms of the unknown value of specific gravity Sg_t at some given temperature T as follows:

$$Sg_t = Sg_{60} + a \times (60 - T) \tag{6.4}$$

The constant a in Eq. (6.4) depends on the particular liquid and represents the slope of the specific gravity versus temperature line for that product. Figure 6.1 shows the variation of specific gravity with temperature for various petroleum liquids.

Example 6.2 The specific gravity of kerosene at 60°F is 0.815. Calculate its specific gravity at 75°F, given that the constant a in Eq. (6.4) is 0.0001.

Solution Using Eq. (6.4) we calculate

$$Sg = 0.815 + 0.0001 \times (60 - 75) = 0.8135$$

Therefore, the specific gravity of kerosene at 75°F is 0.8135.

6.2 Specific Gravity of Blended Products

The specific gravity of a mixture of two or more petroleum products can be calculated fairly easily using the weighted-average method. Since weight is the product of volume and specific weight and the total weight of the mixture is equal to the sum of the component weights, we can write the following equation for the specific gravity of a blend of two or more products, assuming a homogenous mixture.

$$Sg_{blend} = \frac{(Sg_1 \times pct_1) + (Sg_2 \times pct_2) + \cdots}{100} \tag{6.5}$$

where Sg_1 and Sg_2 are the specific gravities, respectively, of the liquids with percentage volumes of pct_1 and pct_2 and Sg_{blend} is the specific gravity of the mixture.

Example 6.3 A mixture consists of 20 percent of light crude of 35 API gravity and 80 percent of heavy crude of 25 API gravity. Calculate the specific gravity and API gravity of the mixture.

Solution To use the specific gravity blending Eq. (6.5) we must convert API gravity to specific gravity,

$$\text{Specific gravity of light crude oil } Sg_1 = \frac{141.5}{131.5 + 35} = 0.8498$$

$$\text{Specific gravity of heavy crude oil } Sg_2 = \frac{141.5}{131.5 + 25} = 0.9042$$

Using Eq. (6.5), the specific gravity of the mixture is calculated as follows:

$$Sg_{\text{blend}} = \frac{(0.8498 \times 20) + (0.9042 \times 80)}{100} = 0.8933$$

The corresponding API gravity of the mixture, using Eq. (6.2), is

$$API_{\text{blend}} = \frac{141.5}{0.8933} - 131.5 = 26.9$$

6.3 Viscosity

Viscosity is a measure of a liquid's resistance to flow. Consider petroleum product flowing through a pipeline. Each layer of liquid flowing through the pipe exerts a certain amount of frictional resistance to the adjacent layer. This is illustrated in Fig. 6.2, where a velocity gradient is shown to exist across the pipe diameter.

According to Newton, the frictional shear stress between adjacent layers of the liquid is related to the flowing velocity across a section of the pipe as

$$\text{Shear stress} = \mu \times \text{velocity gradient}$$

or

$$\tau = \mu \frac{dv}{dy}$$

The *velocity gradient* is defined as the rate of change of liquid velocity along a pipe diameter. The proportionality constant μ in the preceding equation is referred to as the absolute, or dynamic viscosity. In SI units μ is expressed in poise [(dynes · s)/cm^2 or g/(cm · s)] or centipoise (cP). In USCS units absolute viscosity is expressed as (lb · s)/ft^2 or slug/(ft · s). However, centipoise is also used in calculations involving USCS units.

The viscosity of petroleum product, like the specific gravity, decreases with an increase in temperature, and vice versa. Typical viscosities of common petroleum products are listed in Table 6.2.

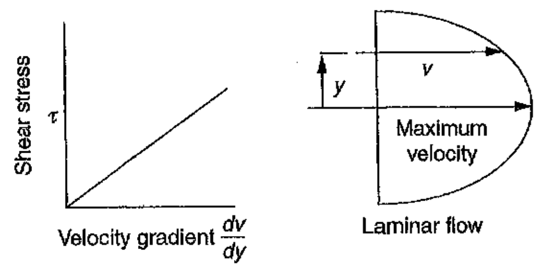


Figure 6.2 Viscosity and Newton's law.

TABLE 6.2 Viscosities of Petroleum Products

Product	Viscosity, cSt at 60°F
Regular gasoline	
Summer grade	0.70
Interseasonal grade	0.70
Winter grade	0.70
Premium gasoline	
Summer grade	0.70
Interseasonal grade	0.70
Winter grade	0.70
No. 1 fuel oil	2.57
No. 2 fuel oil	3.90
Kerosene	2.17
Jet fuel JP-4	1.40
Jet fuel JP-5	2.17

The absolute viscosity μ was defined earlier. Another term known as the *kinematic viscosity* of a liquid is defined as the absolute viscosity divided by the density. It is generally represented by the symbol ν. Therefore,

$$\text{Kinematic viscosity } \nu = \frac{\text{absolute viscosity } \mu}{\text{density } \rho}$$

In USCS units kinematic viscosity is measured in ft^2/s. In SI units, kinematic viscosity is expressed as m^2/s, stokes, or centistokes (cSt). However, centistoke units are also used in calculations involving USCS units. One stoke equals 1 cm^2/s. In SI units, absolute viscosity and kinematic viscosity are related simply by specific gravity as follows:

$$\text{Kinematic viscosity (cSt)} = \frac{\text{absolute viscosity (cP)}}{\text{specific gravity}}$$

In the petroleum industry kinematic viscosity is also expressed in terms of seconds Saybolt Universal (SSU) or seconds Saybolt Furol (SSF). These do not actually represent the physical concept of viscosity but rather a relative measure of how difficult or how easily the liquid flows. In fact both SSU and SSF represent the time taken for a fixed volume [usually 60 milliliters (mL)] of liquid to flow through a specified orifice as measured in a lab. Thus the viscosity of Alaskan North Slope (ANS) crude may be reported as 200 SSU at 60°F. This simply means that in a laboratory a 60-mL sample of ANS crude at 60°F took 200 seconds (s) to flow through a specified orifice. In comparison lighter crude may take only 80 seconds to flow through the same orifice at the same temperature. Therefore the lighter crude has a viscosity of 80 SSU.

The kinematic viscosity of a liquid may thus be expressed in cSt, SSU, or SSF. The equations to convert between these units are given here.

To convert viscosity from SSU to centistokes:

$$\text{Centistokes} = \begin{cases} 0.226 \times \text{SSU} - \dfrac{195}{\text{SSU}} & \text{for } 32 \leq \text{SSU} \leq 100 \quad (6.6) \\ 0.220 \times \text{SSU} - \dfrac{135}{\text{SSU}} & \text{for SSU} > 100 \quad (6.7) \end{cases}$$

To convert viscosity from SSF to centistokes:

$$\text{Centistokes} = \begin{cases} 2.24 \times \text{SSF} - \dfrac{184}{\text{SSF}} & \text{for } 25 \leq \text{SSF} \leq 40 \quad (6.8) \\ 2.16 \times \text{SSF} - \dfrac{60}{\text{SSF}} & \text{for SSF} > 40 \quad (6.9) \end{cases}$$

To convert viscosity from centistokes to SSU, we have to solve for SSU from Eqs. (6.6) or (6.7). It can be seen that this is not very straightforward. We have to solve a quadratic equation in the unknown quantity SSU, as follows:

$$0.226(\text{SSU})^2 - c(\text{SSU}) - 195 = 0 \qquad \text{for } 32 \leq \text{SSU} \leq 100 \quad (6.10)$$

$$0.220(\text{SSU})^2 - c(\text{SSU}) - 135 = 0 \qquad \text{for SSU} > 100 \quad (6.11)$$

In both Eqs. (6.10) and (6.11) the viscosity in centistokes is represented by the variable c.

For example, if the value of viscosity is 10 cSt and we want to convert it to SSU, we need to first guess the answer so we can choose which one of Eqs. (6.10) and (6.11) we should use. The SSU value is generally about 5 times the cSt value. So a viscosity of 10 cSt will be approximately 50 SSU. Therefore we must use Eq. (6.10) since that is for SSU values between 32 and 100. So the solution for the conversion of 10 cSt to SSU will be found from

$$0.226(\text{SSU})^2 - 10(\text{SSU}) - 195 = 0$$

An example will illustrate the method.

Example 6.4

(a) The kinematic viscosity of Alaskan North Slope (ANS) crude oil at 60°F is 200 SSU. Express this viscosity in cSt. The specific gravity of ANS at 60°F is 0.895.

(b) If a light crude oil has a kinematic viscosity of 5.9 cSt, what is this viscosity in SSU?

(c) A heavy fuel oil has a viscosity of 350 SSF. Convert this viscosity to kinematic viscosity in centistokes. If the specific gravity of the fuel oil is 0.95, what is the absolute viscosity in cP?

Solution

(a) From Eq. (6.7) we convert SSU to cSt,

$$\text{Centistokes} = 0.220 \times 200 - \frac{135}{200} = 43.33 \text{ cSt}$$

(b) First we guess the SSU as 5×cSt = 30 SSU. Then using Eq. (6.6) we get

$$5.9 = 0.226(\text{SSU}) - \frac{195}{\text{SSU}}$$

Simplifying,

$$0.226(\text{SSU})^2 - 5.9(\text{SSU}) - 195 = 0$$

Solving the quadratic equation for SSU, we get

$$\text{SSU} = \frac{5.9 \pm \sqrt{(5.9)^2 + 4 \times 195 \times 0.226}}{2 \times 0.226} = \frac{5.9 \pm 14.53}{0.452}$$

or, taking the positive value of the solution,

$$\text{SSU} = 45.20$$

(c) Using Eq. (6.9) to convert SSF to centistokes,

$$\text{Centistokes} = 2.16(350) - \frac{60}{350} = 756 \text{ cSt}$$

The viscosity of a liquid decreases as the temperature increases, similar to the specific gravity. However, even in the normal range of temperature, unlike specific gravity, the viscosity variation with temperature is nonlinear. Several correlations have been proposed to calculate viscosity variation with temperature. The ASTM D341 method uses a log-log correlation that can be used to plot the viscosity versus temperature on a special graph paper. The temperatures and viscosities are plotted on a graph paper with logarithmic scales on each axis.

Sometimes, the viscosity ν in centistokes of a petroleum product and its absolute temperature T may be represented by the following equation:

$$\log_e \nu = A - B(T) \tag{6.12}$$

where A and B are constants that depend on the petroleum product and T is the absolute temperature in °R (°F + 460) or K (°C + 273).

Based on relationship (6.12), a graph of $\log_e \nu$ plotted against temperature T will be a straight line. The slope of the line will be represented by the constant B, and the intercept on the vertical axis would be the constant A. In fact, A would represent the log (viscosity) at the temperature $T = 0$.

If we are given two sets of viscosity values corresponding to two different temperatures, from lab data we could substitute those values in Eq. (6.12) and find the constants A and B for the particular petroleum product. Having calculated A and B, we will then be able to calculate the viscosity of the product at any other temperature using Eq. (6.12). We will explain this method using an example.

Example 6.5 A petroleum oil has the following viscosities at the two temperatures:

$$\text{Viscosity at } 60°\text{F} = 43 \text{ cSt}$$

$$\text{Viscosity at } 100°\text{F} = 10 \text{ cSt}$$

We are required to find the viscosity versus temperature correlation and calculate the viscosity of this oil at 80°F.

Solution Using Eq. (6.12), substituting the given pairs of temperature-viscosity data, we get two equations to solve for A and B as follows:

$$A - B(60 + 460) = \log_e 43$$

$$A - B(100 + 460) = \log_e 10$$

Solving these equations, we get the following values for the constants A and B:

$$A = 22.72 \qquad B = 0.0365$$

We can now calculate the viscosity of this liquid at any temperature from Eq. (6.12). To calculate the viscosity at 80°F, substitute the temperature in the equation as follows:

$$\log_e \nu = 22.72 - 0.0365(80 + 460)$$

Solving for viscosity, we get

$$\text{Viscosity at } 80°\text{F} = 20.35 \text{ cSt}$$

In addition to the simple logarithmic relationship previously described for viscosity versus temperature, other empirical correlations have been put forth by several researchers. One of the more popular formulas is the ASTM method of calculating the viscosities of petroleum products. Using this approach, also known as the ASTM D341 method, a graph paper with logarithmic scales is used to plot the temperature versus viscosity of a liquid at two known temperatures. From two pairs of data plotted on the log-log paper, a straight line is drawn connecting them. The viscosity at any intermediate temperature can then be interpolated. Sometimes, viscosity may also be extrapolated from this chart, beyond

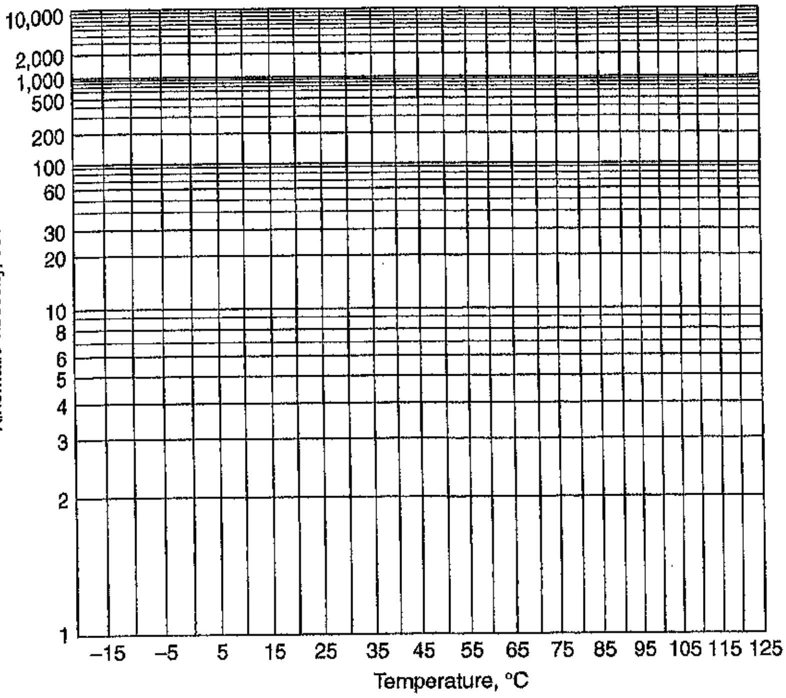


Figure 6.3 ASTM D341—Viscosity temperature chart.

the temperature range used. The ASTM viscosity versus temperature chart is shown in Fig. 6.3.

For viscosity variations with temperature, using the ASTM method, the following analytical method may be used. Here the relationship between viscosity and temperature is given by a log log equation as follows:

$$\log\log Z = A - B \log T \tag{6.13}$$

where log is the logarithm to base 10 and Z is a parameter that depends on the kinematic viscosity of the liquid ν in centistokes and T is the absolute temperature in °R or K. As before, the constants A and B depend on the specific petroleum product.

The parameter Z depends on the liquid viscosity as follows:

$$Z = \nu + 0.7 + C - D \tag{6.14}$$

where C and D are further parameters that depend on the viscosity as follows:

$$C = \exp(-1.14883 - 2.65868\nu) \tag{6.15}$$

$$D = \exp(-0.0038138 - 12.5645\nu) \tag{6.16}$$

where exp(x) represents the value of e^x where e is the base of natural logarithms and numerically $e = 2.71828$.

If we are given two sets of temperature-viscosity data, we can substitute those values in Eqs. (6.14) to (6.16) and calculate the pair of values for the parameters C, D, and Z. Next we can substitute the two sets of temperature and Z values in Eq. (6.13) to calculate the values of the constants A and B. Once we know A and B we can calculate the viscosity at any other temperature using Eq. (6.13). We will illustrate this method using an example.

Example 6.6 A certain petroleum product has temperature versus viscosity data obtained from a lab as follows:

Temperature, °F	60	180
Viscosity, cSt	750	25

(a) Determine the viscosity versus temperature relationship for this product based on the ASTM equations (6.14) to (6.16).

(b) Calculate the viscosity of this liquid at 110°F.

Solution

(a) First calculate the values of C, D, and Z at 60°F using Eqs. (6.14) through (6.16):

$$C_1 = \exp(-1.14883 - 2.65868 \times 750) = 0$$

$$D_1 = \exp(-0.0038138 - 12.5645 \times 750) = 0$$

$$Z_1 = 750 + 0.7 = 750.7$$

Next we repeat these calculations using the 180°F data. The values of C, D, and Z at 180°F are

$$C_2 = \exp(-1.14883 - 2.65868 \times 25) = 0$$

$$D_2 = \exp(-0.0038138 - 12.5645 \times 25) = 0$$

$$Z_2 = 25 + 0.7 = 25.7$$

Next, use the two sets of Z values at the two temperatures in Eq. (6.13) to produce two equations in A and B as follows:

$$\log\log 750.7 = A - B\ \log(60 + 460)$$

$$\log\log 25.7 = A - B\ \log(180 + 460)$$

Simplifying, these equations become,

$$0.4587 = A - 2.716B$$

and

$$0.1492 = A - 2.8062B$$

The values of A and B can now be found by solving the preceding two simultaneous equations, to yield

$$A = 9.78 \qquad B = 3.43$$

Therefore, the viscosity versus temperature relationship for this product is

$$\log\log Z = A - B\ \log T$$

where Z is a parameter that depends on viscosity in cSt, T is the absolute temperature in °F, and the logarithms are to base 10.

(b) At a temperature of 110°F using the equation generated in part (a), we get

$$\log\log Z = A - B\ \log(110 + 460)$$

Substituting the values of A and B, we have

$$\log\log Z = 9.78 - 3.43 \times 2.7559 = 0.3273$$

Solving for Z we get

$$Z = 133.26$$

The viscosity at 110°F is then found from Eq. (6.14) as

$$\text{Viscosity} = 133.26 - 0.7 = 132.56\,\text{cSt}$$

Example 6.7 A crude oil has a dynamic viscosity of 30 cP at 20°C. Calculate its kinematic viscosity in SI units. The density is 0.85 gram per cubic centimeter (g/cm^3).

Solution Since the density in g/cm^3 is numerically the same as specific gravity,

$$\begin{aligned}\text{Kinematic viscosity (cSt)} &= \frac{\text{absolute viscosity (cP)}}{\text{specific gravity}}\\ &= \frac{30.0}{0.85}\\ &= 35.29\,\text{cSt}\end{aligned}$$

Example 6.8 The viscosity of a typical crude oil was measured at two different temperatures as follows:

Temperature, °F	60	100
Viscosity, cSt	35	15

Using the ASTM method of correlation and the log log equations (6.14) to (6.16), calculate the viscosity of this oil at 75°F.

Solution First calculate the values of C, D, and Z at 60°F using Eqs. (6.14) through (6.16):

$$C_1 = \exp(-1.14883 - 2.65868 \times 35) = 0$$

$$D_1 = \exp(-0.0038138 - 12.5645 \times 35) = 0$$

$$Z_1 = 35 + 0.7 = 35.7$$

Next we repeat these calculations using the 100°F data. The values of C, D, and Z at 100°F are

$$C_2 = \exp(-1.14883 - 2.65868 \times 15) = 0$$

$$D_2 = \exp(-0.0038138 - 12.5645 \times 15) = 0$$

$$Z_2 = 15 + 0.7 = 15.7$$

Next, use the two sets of Z values at the two temperatures in Eq. (6.13) to produce two equations in A and B as follows:

$$\log\log 35.7 = A - B\ \log(60 + 460)$$

$$\log\log 15.7 = A - B\ \log(100 + 460)$$

Solving for A and B we get

$$A = 9.7561 \quad \text{and} \quad B = 3.5217$$

The viscosity of the oil at 75°F using Eq. (6.13) is

$$\log\log Z = 9.7561 - 3.5217 \times \log(75 + 460)$$

Solving for Z we get

$$Z = 25.406$$

Therefore the viscosity at 75°F using Eq. (6.14) is

$$\text{Viscosity} = Z - 0.7 = 24.71\,\text{cSt}$$

6.4 Viscosity of Blended Products

The viscosity of a mixture of two or more petroleum products can be calculated using one of two methods. Viscosity, unlike specific gravity, is a nonlinear property. Therefore we cannot use a weighted-average method to calculate the viscosity of a mixture of two or more liquids. For example, 20 percent of a liquid with 10 cSt viscosity when blended with 80 percent of a liquid of 20 cSt viscosity will not result in the following weight-averaged viscosity:

$$\text{Viscosity} = \frac{(10 \times 20) + (20 \times 80)}{100}$$

$$= 18\ \text{cSt}$$

This viscosity of mixture is incorrect. We will now show how to calculate the viscosity of the blend of two or more liquids using an empirical method. The viscosity of a mixture of petroleum products can be calculated using the following formula:

$$\sqrt{V_b} = \frac{Q_1 + Q_2 + \cdots}{(Q_1/\sqrt{V_1}) + (Q_2/\sqrt{V_2}) + \cdots} \tag{6.17}$$

where V_b = viscosity of blend, SSU
Q_1, Q_2, etc. = volumes of each liquid component
V_1, V_2, etc. = viscosity of each liquid component, SSU

Note that in Eq. (6.17) for calculating the viscosity of a mixture or a blend of multiple liquids, all viscosities must be in SSU. If the viscosities of the liquids are given in cSt, we must first convert the viscosities from cSt to SSU before using the equation to calculate the blended viscosity. Also the minimum viscosity that can be used is 32 SSU, equivalent to 1.0 cSt which happens to be the viscosity of water.

Another method for calculating the viscosity of a mixture of products is using the so-called *blending index*. It has been used in the petroleum pipeline industry for many years. Using this method involves calculating a parameter called the blending index for each liquid based on its viscosity. Next, from the component blending index, the blending index of the mixture is calculated using the weighted average of the composition of the mixture. Finally, the viscosity of the mixture is calculated from the blending index of the mixture. The calculation method is as follows:

$$H = 40.073 - 46.414 \log\log(\nu + A) \tag{6.18}$$

$$A = \begin{cases} 0.931(1.72)^{\nu} & \text{for } 0.2 < \nu < 1.5 \quad (6.19) \\ 0.6 & \text{for } \nu \geq 1.5 \quad (6.20) \end{cases}$$

$$H_m = \frac{H_1(\text{pct}_1) + H_2(\text{pct}_2) + \cdots}{100} \tag{6.21}$$

where H, H_1, H_2, etc. = blending index of the liquids
H_m = blending index of the mixture
A = constant in blending index equation
ν = viscosity, cSt
pct_1, pct_2, etc.... = percentage of liquids 1,2, etc., in the mixture
log = logarithm to base 10

Another method to calculate the blended viscosities of two or more petroleum products is the ASTM D341-77 method which employs a graphical approach. Two products at a time are considered and can be extended to more products, taking the blended properties of the first two products and combining with the third, etc. In this method, a special logarithmic graph paper with viscosity scales on the left and right sides of the paper and the percentage of the two products listed on the horizontal axis is used. This is shown in Fig. 6.4. This chart is also available in many handbooks such as the Hydraulic Institute's *Engineering Data Book*. Using this method requires that the viscosities of all products be in SSU and at the same temperature.

For more than two liquids, the blended viscosity of two product at a time is calculated and the process is then repeated for additional

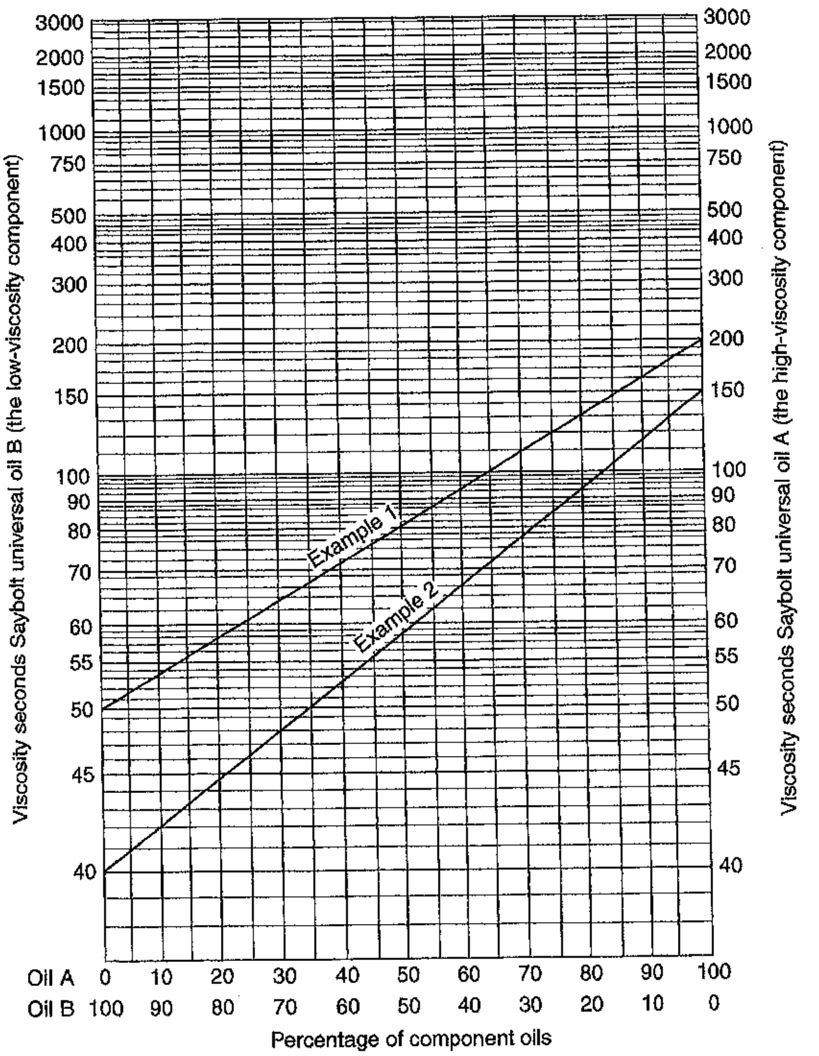


Figure 6.4 Viscosity blending chart.

products, combining the third product with the mixture of the first two products, and so on. Therefore if three products are to be blended in the ratios of 10, 30, and 60 percent, we would first calculate the viscosity of the blend of the first two liquids considering 10 parts of liquid A blended with 30 parts of liquid B. Therefore we would calculate the blend viscosity based on one-fourth of liquid A and three-fourths of liquid B. Next, we would calculate the blend of this mixture combined with liquid C in the proportions of 40 and 60 percent, respectively.

Example 6.9 Calculate the blended viscosity of a liquid consisting of a mixture of 15 percent of liquid A with 85 percent of liquid B. The liquids A and B have a viscosity of 12 and 23 cSt, respectively, at 60°F.

Solution For liquid A, the viscosity of 12 cSt is converted to SSU as follows. Since 12 cSt is estimated to be approximately 12 × 5 = 60 SSU, we use Eq. (6.6):

$$\text{Centistokes} = 0.226 \times \text{SSU} - \frac{195}{\text{SSU}} \qquad \text{for } 32 \le \text{SSU} \le 100$$

Substituting the 12 cSt in the preceding equation and rearranging, we get

$$\nu_A{}^2 - \frac{12}{0.226}\nu_A - \frac{195}{0.226} = 0$$

Solving this quadratic equation;

$$\nu_A = 66.14 \text{ SSU}$$

Next the viscosity of liquid B (23 cSt) is converted to SSU using Eq. (6.7) as follows:

$$\nu_B{}^2 - \frac{23}{0.22}\nu_B - \frac{135}{0.22} = 0$$

Solving we get

$$\nu_B = 110.12 \text{ SSU}$$

To calculate the blended viscosity we use Eq. (6.17):

$$\sqrt{\nu_{\text{blend}}} = \frac{15 + 85}{(15/\sqrt{66.14}) + (85/\sqrt{110.12})} = 10.06$$

Therefore the viscosity of the mixture is

$$\nu_{\text{blend}} = 101.12 \text{ SSU}$$

Converting this viscosity to cSt using Eq. (6.7),

$$\text{Centistokes} = 0.220 \times \text{SSU} - \frac{135}{\text{SSU}} \qquad \text{for SSU} > 100$$

$$= 0.22 \times 101.12 - \frac{135}{101.12} = 20.91$$

Thus the viscosity of the mixture is 20.91 cSt.

6.5 Bulk Modulus

The bulk modulus of a liquid indicates the compressibility of the liquid. Even though most petroleum liquids are incompressible for all practical purposes, this property becomes significant in some instances of liquid flow through pipelines. *Bulk modulus* is generally defined as the pressure required to produce a unit change in volume. If the volume is V and a pressure of ΔP causes a volume change of ΔV, the bulk modulus becomes

$$K = \frac{V \Delta P}{\Delta V} \tag{6.22}$$

where the ratio $\Delta V/V$ represents the change in volume divided by the original volume. In other words, it is the fractional change in volume generated by the pressure change ΔP. If the ratio $\Delta V/V$ becomes equal to 1.0, then numerically, the bulk modulus equals the value of ΔP from Eq. (6.22). For most petroleum products the bulk modulus K is in the range of 200,000 to 400,000 psi (29 to 58 GPa in SI units). There are two distinct values of bulk modulus defined in practice. The isothermal bulk modulus is measured at a constant temperature, while the adiabatic bulk modulus is based on adiabatic conditions (no heat transfer).

The bulk modulus is used in flow measurements of petroleum products and in line pack calculations of long-distance pipelines. The following equations are used to calculate the bulk modulus of a petroleum product, based on the API gravity, pressure, and temperature. Adiabatic bulk modulus K_a is calculated from

$$K_a = A + BP - C(T)^{1/2} - D(\text{API}) - E(\text{API})^2 + FT(\text{API}) \tag{6.23}$$

where $A = 1.286 \times 10^6$
$B = 13.55$
$C = 4.122 \times 10^4$
$D = 4.53 \times 10^3$
$E = 10.59$
$F = 3.228$
P = pressure, psig
T = temperature, °R
API = API gravity of liquid

The isothermal bulk modulus K_i is calculated from

$$K_i = A + BP - C(T)^{1/2} + D(T)^{3/2} - E(\text{API})^{3/2} \tag{6.24}$$

where $A = 2.619 \times 10^6$
$B = 9.203$
$C = 1.417 \times 10^5$

$D = 73.05$
$E = 341.0$
$P = $ pressure, psig
$T = $ temperature, °R
$\text{API} = $ API gravity of liquid

Example 6.10 A typical crude oil has an API gravity of 35°. If the pressure is 1200 psig and the temperature of the crude is 75°F, calculate the bulk modulus.

Solution From Eq. (6.23), the adiabatic bulk modulus is

$$K_a = A + B(P) - C(T)^{1/2} - D(\text{API}) - E(\text{API})^2 + F(T)(\text{API})$$

Therefore,

$$K_a = 1.286 \times 10^6 + 13.55 \times 1200 - 4.122 \times 10^4 \times (75 + 460)^{1/2} - 4.53 \times 10^3 \times 35 - 10.59 \times (35)^2 + 3.228 \times (75 + 460)(35)$$

or

$$K_a = 237{,}760 \text{ psi}$$

From Eq. (6.24), the isothermal bulk modulus is

$$K_i = A + B(P) - C(T)^{1/2} + D(T)^{3/2} - E(\text{API})^{3/2}$$

Therefore,

$$K_i = 2.619 \times 10^6 + 9.203 \times (1200) - 1.417 \times 10^5 \times (75 + 460)^{1/2} + 73.05 \times (75 + 460)^{3/2} - 341.0 \times (35)^{3/2}$$

or

$$K_i = 186{,}868$$

In summary,

$$\text{Adiabatic bulk modulus} = 237{,}760 \text{ psi}$$
$$\text{Isothermal bulk modulus} = 186{,}868 \text{ psi}$$

6.6 Vapor Pressure

Vapor pressure is an important property of petroleum liquids when dealing with storage tanks and centrifugal pumps. Depending upon the location of petroleum product storage tanks, local air quality regulations require certain types of seals around floating roof tanks. These seal designs depend upon the vapor pressure of the liquid in the storage tank. Also, careful analysis of centrifugal pump suction piping used for higher vapor pressure liquids is required in order to prevent cavitation damage to pump impellers at low suction pressures.

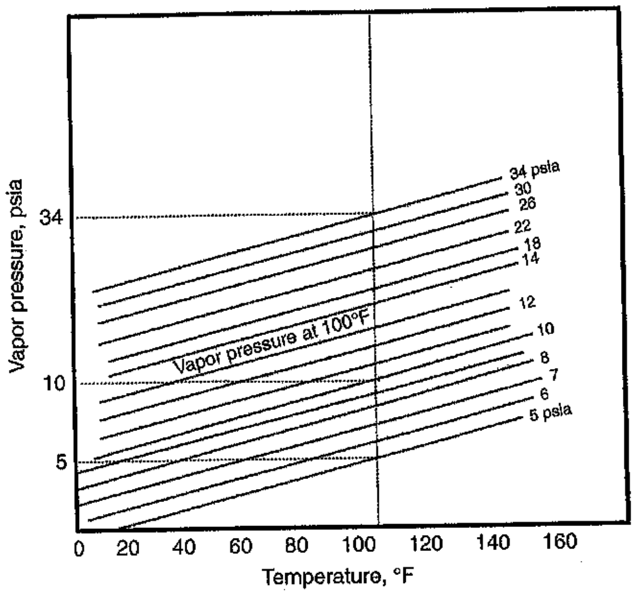


Figure 6.5 Vapor pressure chart for various petroleum products.

The *vapor pressure* may be defined as the pressure at a particular temperature when the liquid and its vapors are in equilibrium, under boiling conditions. When pumping petroleum products through a pipeline, the pressure at any point along the pipeline must be maintained above the vapor pressure of the liquid at the pumping temperature. This will ensure that the petroleum product will remain in the liquid phase throughout. Otherwise liquid may vaporize at some points and two-phase flow may occur that will cause damage to pumping equipment.

Vapor pressure is measured in the laboratory at a standard temperature of 100°F and is referred to as the Reid vapor pressure. ASTM specifications outline the laboratory method of determining this value. Once we know the Reid vapor pressure, we can calculate the vapor pressure at the operating temperature, such as 60°F or 70°F . Charts are available to determine the actual vapor pressure of a petroleum product at storage temperature from a given value of Reid vapor pressure. Figure 6.5 shows a sample vapor pressure chart for various petroleum products.

6.7 Pressure

Pressure within a body of fluid is defined as the force per unit area. In USCS units, pressure is measured in lb/in^2 (psi) and in SI units it is measured in N/m^2 or pascals (Pa). Other units for pressure include lb/ft^2, kPa, MPa, GPa, kg/cm^2, and bar.

The pressure at any point within a liquid is the same in all directions. The actual value of pressure at a point changes with the location of the point within the liquid. Consider a storage tank with the liquid surface exposed to the atmosphere. At all points along the surface of the liquid the pressure is equal to the atmospheric pressure (usually 14.7 psi at sea level or 1 bar in SI units). As we move vertically down through the liquid, the pressure at any point within the liquid is equal to the atmospheric pressure plus the intensity of pressure due to the depth below the free surface. This is defined as the *absolute pressure* since it includes the atmospheric pressure. If we neglect the atmospheric pressure, the pressure within the liquid is termed the *gauge pressure*. Since the atmospheric pressure is present everywhere, it is customary to ignore this and to refer to pressure in gauge pressure.

Returning to the example of the pressure within a storage tank, if the location is at a depth H below the free surface of the liquid, the pressure is equal to the column of liquid of height h acting over a unit cross-sectional area. If the specific weight of the liquid is γ lb/ft^3 and if we consider a cylindrical volume of cross-sectional area A ft^2 and height h ft the pressure at a depth of h is calculated as follows:

$$\text{Pressure } P = \frac{h \times A \times \gamma}{A} = \gamma H \quad \text{lb/ft}^2$$

Converting to the USCS unit of psi,

$$P = \frac{\gamma h}{144} \text{ psi}$$

This is the gauge pressure. The absolute pressure would be $(\gamma h/144) + P_{\text{atm}}$ where P_{atm} is the atmospheric pressure.

More generally we can state that the absolute pressure is

$$P_{\text{abs}} = P_{\text{gauge}} + P_{\text{atm}}$$

The unit for absolute pressure is designated as psia, and the unit for gauge pressure is psig. Since the pressure for most petroleum product applications is measured by gauges, this unit is assumed. Unless otherwise specified, psi means gauge pressure.

Consider a numerical example based on the preceding. At a depth of 50 ft below the free surface of a petroleum (specific gravity = 0.85) storage tank the pressure in the liquid is calculated as follows:

$$\begin{aligned}\text{Pressure} &= \text{weight of 50-ft column of liquid acting on an area 1 in}^2\\ &= 50 \times \left(0.85 \times \frac{62.4}{144}\right) = 18.4 \text{ psig}\end{aligned}$$

we have assumed 62.4 lb/ft^3 as the specific weight of water.

Liquid pressure may also be expressed as head pressure, in which case it is expressed in feet of liquid head (or meters in SI units). Therefore, a pressure of 1000 psi in crude oil of specific gravity 0.895 is said to be equivalent to a pressure head of

$$h = \frac{1000 \times 144}{62.4 \times 0.895} = 2578.4 \text{ ft}$$

In a more general form, the pressure P in psi and liquid head h in feet for a specific gravity of Sg are related by

$$P = \frac{h \times \text{Sg}}{2.31} \tag{6.25}$$

In SI units, pressure P in kPa and head h in meters are related by the following equation:

$$P = \frac{h \times \text{Sg}}{0.102} \tag{6.26}$$

Example 6.11 Calculate the pressure in psi at a depth of 40 ft in a crude oil tank assuming 56.0 lb/ft^3 for the specific weight of crude oil. What is the equivalent pressure in kPa? If the atmospheric pressure is 14.7 psi, calculate the absolute pressure at that depth.

Solution Using Eq. (6.25),

$$\text{Pressure} = \frac{56.0/62.4 \times 40}{2.31} = 15.54 \text{ psig}$$

Thus,

$$\text{Pressure at depth 40 ft} = 15.54 \text{ psig}$$

$$\text{Absolute pressure} = 15.54 + 14.7 = 30.24 \text{ psia}$$

In SI units we can calculate the pressures as follows. Since 1 kPa = 0.145 psi,

$$\text{Pressure at depth 40 ft} = \frac{15.54 \text{ psig}}{0.145 \text{ psi/kPa}} = 107.2 \text{ Pa (gauge)}$$

6.8 Velocity

The speed at which a petroleum product flows through a pipeline, also referred to as velocity, is an important parameter in pipeline pressure drop calculations. The velocity of flow depends on the pipe diameter and flow rate. If the flow rate is constant throughout the pipeline (steady flow) and the pipe diameter is uniform, the velocity at every cross section along the pipe will be a constant value. However, there is a variation in velocity along the pipe cross section. The velocity at the pipe wall will be zero, increasing to a maximum at the centerline of the pipe. This is illustrated in Fig. 6.6.

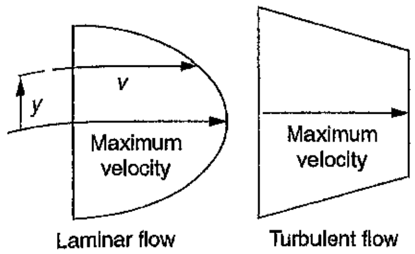


Figure 6.6 Velocity variation—laminar and turbulent.

We can define an average velocity of flow at any cross section of the pipe as follows:

$$\text{Velocity} = \frac{\text{flow rate}}{\text{area of flow}}$$

If the flow rate is in ft^3/s and the pipe cross-sectional area is in ft^2, the velocity from the preceding equation is in ft/s.

Consider liquid flowing through a circular pipe of internal diameter D at a flow rate of Q. Then the average flow velocity is

$$v = \frac{Q}{\pi D^2/4} \tag{6.27}$$

Employing commonly used units of flow rate Q in ft^3/s and pipe diameter in inches, the velocity in ft/s is as follows:

$$v = \frac{144Q}{\pi D^2/4}$$

Simplifying to

$$v = 183.3461\,\frac{Q}{D^2} \tag{6.28}$$

where the flow rate Q is in ft^3/s and the pipe inside diameter is in inches.

In petroleum transportation, flow rates are usually expressed in bbl/h, bbl/day, or gal/min. Therefore Eq. (6.28) for velocity can be modified in terms of more conventional pipeline units as follows. For flow rate in bbl/h:

$$v = 0.2859\,\frac{Q}{D^2} \tag{6.29}$$

where v = velocity, ft/s
Q = flow rate, bbl/h
D = pipe inside diameter, in

For flow rate in bbl/day:

$$v = 0.0119 \frac{Q}{D^2} \tag{6.30}$$

where v = velocity, ft/s
Q = flow rate, bbl/day
D = pipe inside diameter, in

For flow rate in gal/min:

$$v = 0.4085 \frac{Q}{D^2} \tag{6.31}$$

where v = velocity, ft/s
Q = flow rate, gal/min
D = pipe inside diameter, in

In SI units, the velocity equation is as follows:

$$v = 353.6777 \frac{Q}{D^2} \tag{6.32}$$

where v = velocity, m/s
Q = flow rate, m^3/h
D = pipe inside diameter, mm

Example 6.12 Diesel flows through an NPS 16 (15.5-in inside diameter) pipeline at the rate of 4000 gal/min. Calculate the average velocity for steady-state flow. (*Note*: The designation NPS 16 means nominal pipe size of 16 in.)

Solution From Eq. (6.31) the average flow velocity is

$$v = 0.4085 \frac{4000}{15.5^2} = 6.80 \text{ ft/s}$$

Example 6.13 Gasoline flows through a DN 400 outside diameter (10-mm wall thickness) pipeline at 200 L/s. Calculate the average velocity for steady flow.

Solution The designation DN 400 in SI units corresponds to NPS 16 in USCS units. DN 400 means metric pipe size of 400-mm outside diameter. First convert the flow rate in L/s to m^3/h.

$$\text{Flow rate} = 200 \text{ L/s} = 200 \times 60 \times 60 \times 10^{-3} \text{ m}^3\text{/h} = 720 \text{ m}^3\text{/h}$$

From Eq. (6.32) the average flow velocity is

$$v = 353.6777 \frac{720}{380^2} = 1.764 \text{ m/s}$$

The variation of flow velocity along the cross section of a pipe as depicted in Fig. 6.6 depends on the type of flow. In laminar flow, the velocity variation is parabolic. As the flow rate becomes turbulent, the velocity profile approximates a more trapezoidal shape as shown. Laminar and turbulent flows are discussed after we introduce the concept of the Reynolds number.

6.9 Reynolds Number

The Reynolds number of flow is a dimensionless parameter that depends on the pipe diameter liquid flow rate, liquid viscosity, and density. It is defined as follows:

$$R = \frac{vD\rho}{\mu} \tag{6.33}$$

or

$$R = \frac{vD}{\nu} \tag{6.34}$$

where R = Reynolds number, dimensionless
v = average flow velocity, ft/s
D = inside diameter of pipe, ft
ρ = mass density of liquid, slug/ft^3
μ = dynamic viscosity, slug/(ft · s)
ν = kinematic viscosity, ft^2/s

In terms of more commonly used units in the oil industry, we have the following versions of the Reynolds number equation:

$$R = 3162.5 \frac{Q}{D\nu} \tag{6.35}$$

where R = Reynolds number, dimensionless
Q = flow rate, gal/min
D = inside diameter of pipe, in
ν = kinematic viscosity, cSt

In petroleum transportation units, the Reynolds number is calculated using the following equations:

$$R = 2213.76 \frac{Q}{D\nu} \tag{6.36}$$

$$R = 92.24 \frac{\text{BPD}}{D\nu} \tag{6.37}$$

where R = Reynolds number, dimensionless
Q = flow rate, bbl/h
BPD = flow rate, bbl/day
D = inside diameter of pipe, in
ν = kinematic viscosity, cSt

In SI units, the Reynolds number is expressed as follows

$$R = 353{,}678 \frac{Q}{\nu D} \tag{6.38}$$

where R = Reynolds number, dimensionless
Q = flow rate, m^3/h
D = inside diameter of pipe, mm
ν = kinematic viscosity, cSt

Example 6.14 A crude oil of specific gravity 0.85 and viscosity 10 cSt flows through an NPS 20 (0.375-in wall thickness) pipeline at 5000 gal/min. Calculate the average velocity and the Reynolds number of flow.

Solution The NPS 20 (0.375-in wall thickness) pipe has an inside diameter = $20.0 - 2 \times 0.375 = 19.25$ in. From Eq. (6.31) the average velocity is calculated first:

$$v = 0.4085 \frac{5000}{19.25^2} = 5.51 \text{ ft/s}$$

From Eq. (6.35) the Reynolds number is therefore

$$R = 3162.5 \frac{5000}{19.25 \times 10.0} = 82{,}143$$

Example 6.15 A petroleum product with a specific gravity of 0.815 and viscosity of 15 cSt flows through a DN 400 (10-mm wall thickness) pipeline at 800 m^3/h. Calculate the average flow velocity and the Reynolds number of flow.

Solution The DN 400 (10-mm wall thickness) pipe has an inside diameter = $400 - 2 \times 10 = 380$ mm. From Eq. (6.32) the average velocity is therefore

$$v = 353.6777 \frac{800}{380^2} = 1.96 \text{ m/s}$$

Next, from Eq. (6.38) the Reynolds number is

$$R = 353{,}678 \frac{800}{380 \times 15.0} = 49{,}639$$

6.10 Types of Flow

Flow through a pipeline is classified as laminar flow, turbulent flow, or critical flow depending on the magnitude of the Reynolds number of flow.

If the Reynolds number is less than 2100, the flow is said to be *laminar*. When the Reynolds number is greater than 4000, the flow is considered to be *turbulent*. *Critical flow* occurs when the Reynolds number is in the range of 2100 to 4000. Laminar flow is characterized by smooth flow in which no eddies or turbulence are visible. The flow is also said to occur in laminations. If dye was injected into a transparent pipeline, laminar flow would be manifested in the form of smooth streamlines of dye. Turbulent flow occurs at higher velocities and is accompanied by eddies and other disturbances in the liquid. More energy is lost in friction in the critical flow and turbulent flow regions as compared to the laminar flow region.

The three flow regimes characterized by the Reynolds number of flow are

$$\begin{aligned} &\text{Laminar flow}: && R \leq 2100 \\ &\text{Critical flow}: && 2100 < R \leq 4000 \\ &\text{Turbulent flow}: && R > 4000 \end{aligned}$$

In the critical flow regime, where the Reynolds number is between 2100 and 4000, the flow is undefined and unstable, as far as pressure drop calculations are concerned. In the absence of better data, it is customary to use the turbulent flow equation to calculate pressure drop in the critical flow regime as well.

6.11 Pressure Drop Due to Friction

As a liquid flows through a pipeline, energy is lost due to resistance between the flowing liquid layers as well as due to the friction between the liquid and the pipe wall. One of the objectives of pipeline calculation is to determine the amount of energy and hence the pressure lost due to friction as the liquid flows from the source to the destination. First we will introduce the equation for conservation of energy in liquid flow in a pipeline. After that we will cover the approach to calculating the frictional pressure drop or head loss calculations. We will begin by discussing Bernoulli's equation for the various forms of liquid energy in a flowing pipeline.

6.11.1 Bernoulli's equation

Bernoulli's equation is another way of stating the principle of conservation of energy applied to liquid flow through a pipeline. At each point along the pipeline the total energy of the liquid is computed by taking into consideration the liquid energy due to pressure, velocity, and elevation combined with any energy input, energy output, and energy losses. The total energy of the liquid contained in the pipeline at any

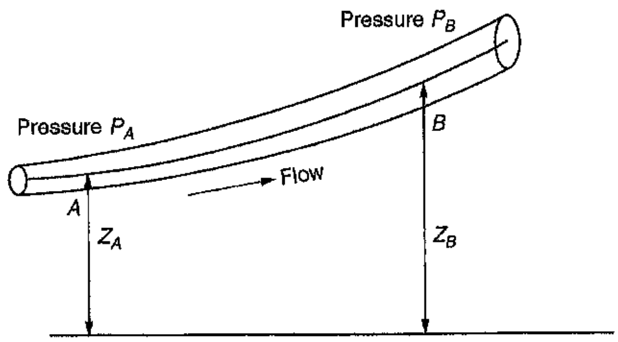


Figure 6.7 Total energy of liquid in pipe flow.

point is a constant. This is also known as the principle of conservation of energy.

Consider a liquid flow through a pipeline from point A to point B as shown in Fig. 6.7. The elevation of point A is Z_A and the elevation at B is Z_B above some common datum, such as mean sea level. The pressure at point A is P_A and that at B is P_B. It is assumed that the pipe diameter at A and B are different, and hence the flow velocity at A and B will be represented by V_A and V_B, respectively. A particle of the liquid of unit weight at point A in the pipeline possesses a total energy E which consists of three components:

$$\text{Potential energy} = Z_A$$

$$\text{Pressure energy} = \frac{P_A}{\gamma}$$

$$\text{Kinetic energy} = \frac{{v_A}^2}{2g}$$

where γ is the specific weight of liquid.

Therefore the total energy E is

$$E = Z_A + \frac{P_A}{\gamma} + \frac{{v_A}^2}{2g} \tag{6.39}$$

Since each term in Eq. (6.39) has dimensions of length, we refer to the total energy at point A as H_A in feet of liquid head. Therefore, rewriting the total energy in feet of liquid head at point A, we obtain

$$H_A = Z_A + \frac{P_A}{\gamma} + \frac{{v_A}^2}{2g} \tag{6.40}$$

Similarly, the same unit weight of liquid at point B has a total energy per unit weight equal to H_B given by

$$H_B = Z_B + \frac{P_B}{\gamma} + \frac{{v_B}^2}{2g} \tag{6.41}$$

By the principle of conservation of energy

$$H_A = H_B \tag{6.42}$$

Therefore,

$$Z_A + \frac{P_A}{\gamma} + \frac{{v_A}^2}{2g} = Z_B + \frac{P_B}{\gamma} + \frac{{v_B}^2}{2g} \tag{6.43}$$

In Eq. (6.43), referred to as Bernoulli's equation, we have not considered any energy added to the liquid, energy taken out of the liquid, or energy losses due to friction. Therefore, modifying Eq. (6.43) to take into account the addition of energy (such as from a pump at A) and accounting for frictional head losses h_f, we get the more common form of Bernoulli's equation as follows:

$$Z_A + \frac{P_A}{\gamma} + \frac{{v_A}^2}{2g} + H_p = Z_B + \frac{P_B}{\gamma} + \frac{{v_B}^2}{2g} + h_f \tag{6.44}$$

where H_P is the equivalent head added to the liquid by the pump at A and h_f represents the total frictional head losses between points A and B.

We will next discuss how the head loss due to friction h_f in Bernoulli's equation is calculated for various conditions of flow of petroleum products of water flow in pipelines. We begin with the classical pressure drop equation known as the Darcy-Weisbach equation, or simply the Darcy equation.

6.11.2 Darcy equation

As a petroleum product flows through a pipeline from point A to point B the pressure decreases due to frictional loss between the flowing liquid and the pipe. The extent of pressure loss due to friction, designated in feet of liquid, depends on various factors. These factors include the liquid flow rate, liquid specific gravity and viscosity, pipe inside diameter, pipe length, and internal condition of the pipe (rough, smooth, etc.). The Darcy equation may be used to calculate the pressure drop in a pipeline as follows:

$$h = f \frac{L}{D} \frac{v^2}{2g} \tag{6.45}$$

where h = frictional pressure loss, ft of liquid head
f = Darcy friction factor, dimensionless
L = pipe length, ft
D = inside pipe diameter, ft
v = average flow velocity, ft/s
g = acceleration due to gravity, ft/s^2

Note that the Darcy equation gives the frictional pressure loss in feet of liquid head, which must be converted to pressure loss in psi using Eq. (6.25). The term $v^2/2g$ in the Darcy equation is called the velocity head, and it represents the kinetic energy of the liquid. The term *velocity head* will be used in subsequent sections of this chapter when discussing frictional head loss through pipe fittings and valves.

The friction factor f in the Darcy equation is the only unknown on the right-hand side of Eq. (6.45). This friction factor is a nondimensional number between 0.0 and 0.1 that depends on the internal roughness of the pipe, the pipe diameter, and the Reynolds number of flow.

In laminar flow, the friction factor f depends only on the Reynolds number and is calculated from

$$f = \frac{64}{R} \tag{6.46}$$

where f is the friction factor for laminar flow and R is the Reynolds number for laminar flow ($R < 2100$) (dimensionless).

Therefore, if a particular flow has a Reynolds number of 1780 we can conclude that in this laminar flow condition the friction factor f to be used in the Darcy equation is

$$f = \frac{64}{1780} = 0.036$$

Some pipeline hydraulics texts may refer to another friction factor called the Fanning friction factor. This is numerically equal to one-fourth the Darcy friction factor. In this example the Fanning friction factor can be calculated as

$$\frac{0.036}{4} = 0.009$$

To avoid any confusion, throughout this chapter we will use only the Darcy friction factor as defined in Eq. (6.45).

In practical situations involving petroleum product pipelines it is inconvenient to use the Darcy equation in the form described in Eq. (6.45). We must convert the equation in terms of commonly used petroleum pipeline units. One form of the Darcy equation in pipeline units is as follows:

$$h = 0.1863 \frac{fLv^2}{D} \tag{6.47}$$

where h = frictional pressure loss, ft of liquid head
f = Darcy friction factor, dimensionless
L = pipe length, ft
D = pipe inside diameter, in
v = average flow velocity, ft/s

Another form of the Darcy equation with frictional pressure drop expressed in psi/mi and using a flow rate instead of velocity is as follows:

$$P_m = \text{const} \frac{fQ^2\text{Sg}}{D^5} \tag{6.48}$$

where P_m = frictional pressure loss, psi/mi
f = Darcy friction factor, dimensionless
Q = flow rate, bbl/h
D = pipe inside diameter, in
Sg = liquid specific gravity
const = factor that depends on flow units

$$= \begin{cases} 34.87 & \text{for } Q \text{ in bbl/h} \\ 0.0605 & \text{for } Q \text{ in bbl/day} \\ 71.16 & \text{for } Q \text{ in gal/min} \end{cases}$$

In SI units, the Darcy equation may be written as

$$h = 50.94 \frac{fLv^2}{D} \tag{6.49}$$

where h = frictional pressure loss, m of liquid head
f = Darcy friction factor, dimensionless
L = pipe length, m
D = pipe inside diameter, mm
v = average flow velocity, m/s

Another version of the Darcy equation in SI units is as follows:

$$P_{km} = (6.2475 \times 10^{10}) \left(\frac{fQ^2\text{Sg}}{D^5}\right) \tag{6.50}$$

where P_{km} = pressure drop due to friction, kPa/km
Q = liquid flow rate, m^3/h
f = Darcy friction factor, dimensionless
Sg = liquid specific gravity
D = pipe inside diameter, mm

6.11.3 Colebrook-White equation

We have seen that in laminar flow the friction factor f is easily calculated from the Reynolds number as shown in Eq. (6.46). In turbulent flow, the calculation of friction factor f is more complex. It depends on the pipe inside diameter, the pipe roughness, and the Reynolds number. Based on work by Moody, Colebrook and White, and others, the following empirical equation, known as the Colebrook-White equation, has been proposed for calculating the friction factor in turbulent flow:

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{e}{3.7D} + \frac{2.51}{R\sqrt{f}}\right) \tag{6.51}$$

where f = Darcy friction factor, dimensionless
D = pipe inside diameter, in
e = absolute pipe roughness, in
R = Reynolds number, dimensionless

The absolute pipe roughness, also known as internal pipe roughness, may range from 0.0 to 0.01 depending on the internal condition of the pipe. It is listed for common piping systems in Table 6.3. The ratio e/D is termed the relative roughness and is dimensionless. Equation (6.51) is also sometimes called simply the Colebrook equation.

In SI units, we can use the same form of the Colebrook equation. The absolute pipe roughness e and the pipe diameter D are both expressed in millimeters. All other terms in the equation are dimensionless.

TABLE 6.3 Pipe Internal Roughness

Pipe material	Roughness in	Roughness mm
Riveted steel	0.035–0.35	0.9–9.0
Commercial steel/welded steel	0.0018	0.045
Cast iron	0.010	0.26
Galvanized iron	0.006	0.15
Asphalted cast iron	0.0047	0.12
Wrought iron	0.0018	0.045
PVC, drawn tubing, glass	0.000059	0.0015
Concrete	0.0118–0.118	0.3–3.0

It can be seen from the Colebrook-White equation that the calculation of the friction factor f is not straightforward since it appears on both sides of the equation. This is known as an implicit equation in f, compared to an explicit equation. An explicit equation in f will have the unknown quantity f on one side of the equation. In the present case, a trial-and-error approach is used to solve for the friction factor. First an initial value for f is assumed (for example, $f = 0.01$) and substituted in the right-hand side of the Colebrook equation. This will result in a new calculated value of f, which is used as the next approximation and f recalculated based on this second approximation. The process is continued until successive values of f calculated by such iterations is within a small value such as 0.001. Usually three or four iterations will yield a satisfactory solution. There are other explicit equations for the friction factor proposed by many researchers, such as Churchill and Swamee-Jain that are easier to use than the Colebrook equation.

6.11.4 Moody diagram

A graphical method of determining the friction factor for turbulent flow is available using the Moody diagram shown in Fig. 6.8. First the Reynolds number is calculated based upon liquid properties, flow rate, and pipe diameter. This Reynolds number is used to locate the ordinate on the horizontal axis of the Moody diagram. A vertical line is drawn up to the curve representing the relative roughness e/D of the pipe. The friction factor is then read off of the vertical axis to the left. From the Moody diagram it is seen that the turbulent region is further divided into two regions: the "transition" zone and the "complete turbulence in rough pipes" zone. The lower boundary is designated as "smooth pipes." The transition zone extends up to the dashed line, beyond which is known as the zone of complete turbulence in rough pipes. In this zone, the friction factor depends very little on the Reynolds number and more on the relative roughness.

The *transmission factor* is a term that is used in conjunction with pressure drop and flow rate in pipelines. The transmission factor, a dimensionless number, is proportional to the flow rate, whereas the friction factor is inversely proportional to the flow rate. With a higher transmission factor, the flow rate is increased, whereas with a higher friction factor, flow rate decreases. The transmission factor F is inversely related to the Darcy friction factor f as follows:

$$F = \frac{2}{\sqrt{f}} \tag{6.52}$$

Examining the Moody diagram we see that the friction factor f ranges from 0.008 to 0.10. Therefore, from Eq. (6.52) we can conclude that

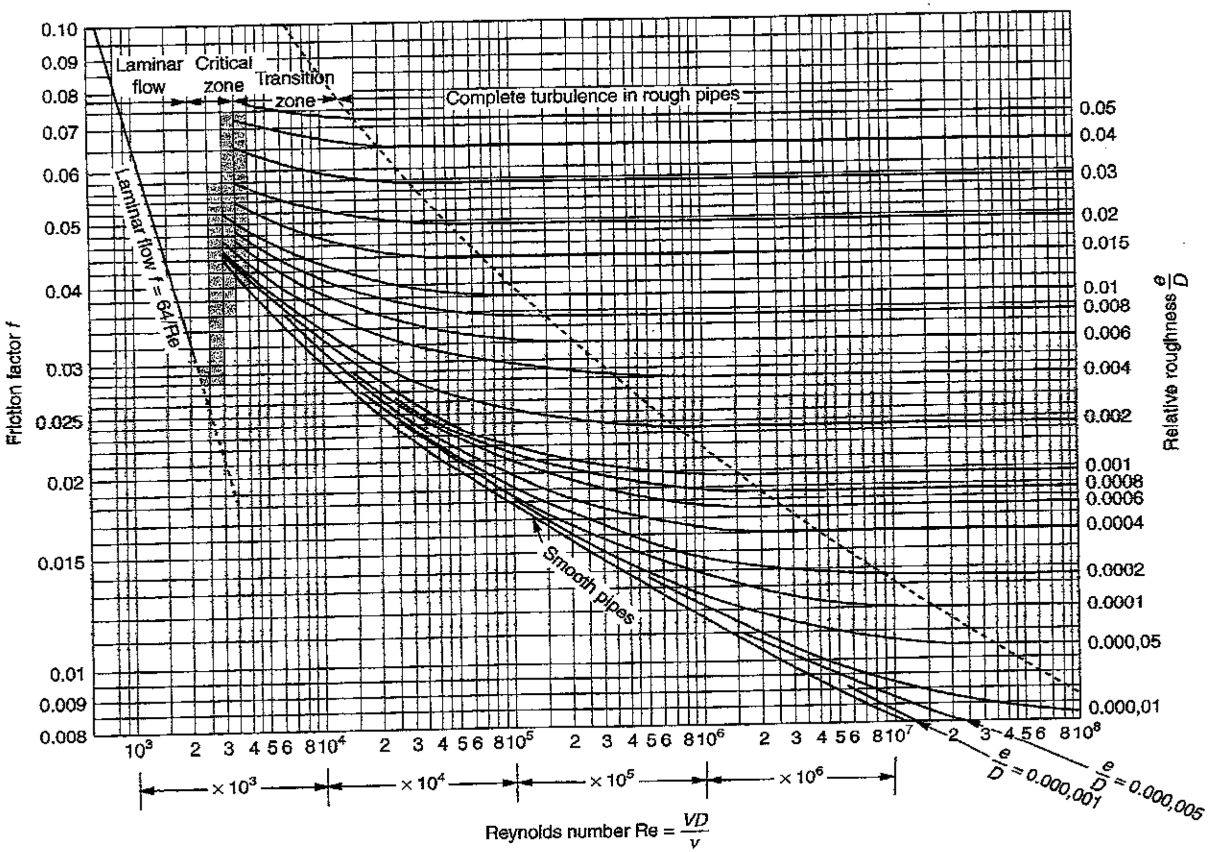


Figure 6.8 Moody diagram.

the transmission factor F will range between 6 and 22. Having introduced the transmission factor F we can now rewrite the Colebrook-White equation in terms of the transmission factor as

$$F = -4\log_{10}\left(\frac{e}{3.7D} + \frac{1.255F}{R}\right) \quad \text{for turbulent flow } R > 4000 \quad (6.53)$$

As we did before with the friction factor f, the transmission factor F must also be calculated from Eq. (6.53) by successive iteration. We assume an initial value for F (for example, $F = 10.0$) and calculate a new value of F by substituting this initial value in the right-hand side of Eq. (6.53). This will result in a second approximation for F, which is then used to recalculate a better value of F. By successive iteration, a satisfactory value of F can be calculated.

The U.S. Bureau of Mines proposed a modified version of the Colebrook-White equation. This is expressed in terms of the transmission factor.

$$F = -4\log_{10}\left(\frac{e}{3.7D} + 1.4125\,\frac{F}{R}\right) \quad \text{for turbulent flow } R > 4000 \quad (6.54)$$

By comparing the modified version in Eq. (6.54) with the original Colebrook-White equation (6.53), we see that the modified Colebrook-White equation uses the constant 1.4125 instead of 1.255. This modification causes a more conservative value of the transmission factor. In other words the modified Colebrook-White equation yields a higher pressure drop for the same flow rate compared to the original Colebrook-White equation.

Example 6.16 A petroleum oil with 0.85 specific gravity and 10 cSt viscosity flows through an NPS 16 (0.250-in wall thickness) pipeline at a flow rate of 4000 bbl/h. The absolute roughness of the pipe may be assumed to be 0.002 in. Calculate the Darcy friction factor and pressure loss due to friction in a mile of pipe length using the Colebrook-White equation. What is the transmission factor?

Solution The inside diameter of an NPS 16 (0.250-in wall thickness) pipe is

$$16.00 - 2 \times 0.250 = 15.50 \text{ in}$$

Next we will calculate the Reynolds number R to determine the flow regime (laminar or turbulent). The Reynolds number from Eq. (6.36) is

$$R = 2213.76\frac{4000}{15.5 \times 10.0} = 57{,}129$$

Since $R > 4000$, the flow is turbulent and we can use the Colebrook-White equation to calculate the friction factor. We can also use the Moody diagram to read the friction factor based on R and the pipe relative roughness e/D.

From the Colebrook-White equation (6.51), the friction factor f is

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{0.002}{3.7 \times 15.5} + \frac{2.51}{57{,}129\sqrt{f}}\right)$$

This equation must be solved for f by trial and error.

First assume that $f = 0.02$. Substituting in the preceding equation, we get a better approximation for f as follows:

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{0.002}{3.7 \times 15.5} + \frac{2.51}{57{,}129\sqrt{0.02}}\right) = 0.0209$$

Recalculating using this value

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{0.002}{3.7 \times 15.5} + \frac{2.51}{57{,}129\sqrt{0.0209}}\right) = 0.0208$$

And finally

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{0.002}{3.7 \times 15.5} + \frac{2.51}{57{,}129\sqrt{0.0208}}\right) = 0.0208$$

Thus $f = 0.0208$ is the solution. The transmission factor is

$$F = \frac{2}{\sqrt{f}} = 13.87$$

Next calculate the average flow velocity needed for the Darcy equation for head loss:

$$\text{Average flow velocity } V = 0.2859 \times \frac{4000}{(15.5)^2} = 4.76 \text{ ft/s} \qquad \text{from Eq. (6.29)}$$

The head loss due to friction can now be calculated using the Darcy equation (6.47), considering a mile of pipe:

$$h = 0.1863\left(0.0208 \times 5280 \times \frac{4.76^2}{15.5}\right)$$

$$= 29.908 \text{ ft of liquid head per mile of pipe}$$

Converting liquid head to pressure in psi using Eq. (6.25) we get

$$\text{Pressure drop } P_m = 29.908 \times \frac{0.85}{2.31} = 11.01 \text{ psi/mi}$$

We could have also calculated the pressure drop per mile directly in psi/mi using the version of the Darcy equation shown in Eq. (6.48).

$$P_m = 34.87 \times 0.0208 \times (4000)^2 \times \frac{0.85}{15.5^5}$$

Therefore,

$$P_m = 11.03 \text{ psi/mi}$$

The slight difference between the two values for P_m is due to rounding off in unit conversions. If we used the Moody diagram to find the friction factor, we would use the Reynolds number of 57,129 and the relative roughness $e/D = 0.002/15.5 = 0.000129$ and read the value of the friction factor $f = 0.021$ approximately. After that, the pressure drop calculation will still be the same as described previously.

Example 6.17 A DN 500 (10-mm wall thickness) steel pipe is used to transport gasoline from a refinery to a storage tank 15 km away. Neglecting any difference in elevations, calculate the friction factor and pressure loss due to friction (kPa/km) at a flow rate of 990 m^3/h. Assume an internal pipe roughness of 0.05 mm. A delivery pressure of 4 kPa must be maintained at the delivery point, and the storage tank is at an elevation of 200 m above that of the refinery. Calculate the pump pressure required at the refinery to transport the given volume of gasoline to the storage tank location. Assume the specific gravity of gasoline is 0.736 and the viscosity is 0.6 cSt.

Solution The DN 500 (10-mm wall thickness) pipe has an inside diameter of

$$D = 500 - 2 \times 10 = 480 \text{ mm}$$

First calculate the Reynolds number from Eq. (6.38):

$$R = \frac{353{,}678Q}{\nu D}$$

$$= \frac{353{,}678 \times 990}{0.6 \times 480} = 1{,}215{,}768$$

Therefore the flow is turbulent and we can use the Colebrook-White equation or the Moody diagram to determine the friction factor.

$$\text{Relative roughness } \frac{e}{D} = \frac{0.05}{480} = 0.0001$$

Using the preceding values for the relative roughness and Reynolds number, from the Moody diagram we get $f = 0.013$. The pressure drop due to friction can now be calculated using the Darcy equation (6.50):

$$P_{\text{km}} = (6.2475 \times 10^{10}) \left(0.013 \times 990^2 \times \frac{0.736}{480^5}\right)$$

$$= 22.99 \text{ kPa/km}$$

The pressure required at the pumping facility is calculated by adding the pressure drop due to friction to the delivery pressure required and the static elevation head between the pumping facility and storage tank. The static head difference is 200 m. This is converted to pressure in kPa, using Eq. (6.26),

$$\text{Pressure drop due to friction in 15 km of pipe} = 15 \times 22.99 = 344.85 \text{ kPa}$$

$$\text{Pressure due to elevation head} = 200 \times \frac{0.736}{0.102} = 1443.14 \text{ kPa}$$

$$\text{Minimum pressure required at delivery point} = 4 \text{ kPa}$$

Therefore adding all three numbers, the total pressure required at the refinery is

$$P_t = P_f + P_{\text{elev}} + P_{\text{del}}$$

where P_t = total pressure required at pump
P_f = frictional pressure drop
P_{elev} = pressure head due to elevation difference
P_{del} = delivery pressure at storage tank

Therefore,

$$P_t = 344.85 + 1443.14 + 4.0 = 1792 \text{ kPa}$$

Thus the pump pressure required at the refinery is 1792 kPa.

6.11.5 Hazen-Williams equation

The Hazen-Williams equation has been used for the calculation of pressure drop in water pipelines and water distribution networks. This equation has also been successfully applied to the calculation of pressure drop in refined petroleum product pipelines, such as gasoline and diesel pipelines. Using the Hazen-Williams method a coefficient C, known as the Hazen-Williams C factor, is used to account for the internal pipe roughness or efficiency. Unlike the Moody diagram or Colebrook-White equation, the Hazen-Williams equation does not use the Reynolds number or viscosity of the liquid to calculate the pressure drop. The Hazen-Williams C factor is a number that is based on experience with a particular product and pipeline. For example, one product pipeline company may use $C = 125$ for diesel and $C = 150$ for gasoline. The higher the C factor, the higher will be the flow rate through the pipeline and the lower the pressure drop due to friction. It may be thought of as an opposite of the friction factor. The Hazen-Williams equation is not used for crude oil and heavier liquids. The Colebrook-White equation gives a better correlation with field data when applied to crude oil pipelines and heated oil pipelines.

The Hazen-Williams equation is generally expressed as follows

$$h = \frac{4.73\, L(Q/C)^{1.852}}{D^{4.87}} \tag{6.55}$$

where h = frictional head loss, ft of liquid head
L = length of pipe, ft
D = pipe inside diameter, ft
Q = flow rate, ft^3/s
C = Hazen-Williams C factor, dimensionless

TABLE 6.4 Hazen-Williams *C* Factor

Pipe material	*C* factor
Smooth pipes (all metals)	130–140
Cast iron (old)	100
Iron (worn/pitted)	60–80
Polyvinyl chloride (PVC)	150
Brick	100
Smooth wood	120
Smooth masonry	120
Vitrified clay	110

The values of the C factor for various applications are listed in Table 6.4. However, it must be noted that when applied to refined petroleum product pipelines these factors have to be adjusted based on experience, since these factors were originally intended for water pipelines.

On examining the Hazen-Williams equation, it can be seen that the head loss due to friction is calculated in feet of liquid head, similar to the Darcy equation. The value of the head loss h can be converted to psi using the head-to-psi conversion equation (6.25). Although using the Hazen-Williams equation appears to be simpler than using the Colebrook-White and Darcy equations to calculate the pressure drop, the unknown term C can cause uncertainties in the pressure drop calculation.

Usually, the C factor is determined based on experience with the particular liquid and the piping system. When designing a new petroleum product pipeline, using the Hazen-Williams equation, we must carefully select the C factor since considerable variation in pressure drop can occur by choosing a particular value of C compared to another. Because of the inverse proportionality effect of C on the head loss, using $C = 120$ instead of $C = 100$ will result in $[1 - (100/120)^{1.852}]$ or 29 percent less pressure drop. Therefore, it is important that the C value be chosen judiciously.

The Hazen-Williams equation (6.55) is not convenient to use when dealing with petroleum pipelines due to the units employed in the original form. Therefore, more acceptable forms of the Hazen-Williams equation have been used in practice. These modified versions of the equation use flow rates in gal/min, bbl/h, and bbl/day with pressure drops expressed in psi/mi and diameter in inches in USCS units. In the following formulas the presented Hazen-Williams equations have been rearranged to calculate the flow rate from a given pressure drop. The versions of the equations to calculate the pressure drop from a given flow rate are also shown.

A modified version of the Hazen-Williams equation in pipeline units is

$$Q = (6.755 \times 10^{-3})CD^{2.63}(h)^{0.54} \qquad (6.56)$$

where Q = flow rate, gal/min
h = friction loss, ft of liquid per 1000 ft of pipe
D = inside diameter of pipe, in
C = Hazen-Williams C factor, dimensionless

Other variants in petroleum pipeline units are as follows:

$$Q = (6.175 \times 10^{-3}) C D^{2.63} \left(\frac{P_m}{\mathrm{Sg}}\right)^{0.54} \tag{6.57}$$

$$P_m = 12{,}352 \left(\frac{Q}{C}\right)^{1.852} \frac{\mathrm{Sg}}{D^{4.87}} \tag{6.58}$$

and

$$P_f = 2339 \left(\frac{Q}{C}\right)^{1.852} \frac{\mathrm{Sg}}{D^{4.87}} \tag{6.59}$$

where Q = flow rate, bbl/h
D = pipe inside diameter, in
P_m = frictional pressure drop, psi/mi
P_f = frictional pressure drop, psi per 1000 ft of pipe length
Sg = liquid specific gravity
C = Hazen-Williams C factor, dimensionless

In SI units, the Hazen-Williams equation is expressed as follows:

$$Q = (9.0379 \times 10^{-8}) C D^{2.63} \left(\frac{P_{km}}{\mathrm{Sg}}\right)^{0.54} \tag{6.60}$$

and

$$P_{\mathrm{km}} = (1.1101 \times 10^{13}) \left(\frac{Q}{C}\right)^{1.852} \frac{\mathrm{Sg}}{D^{4.87}} \tag{6.61}$$

where Q = flow rate, $\mathrm{m^3/h}$
D = pipe inside diameter, mm
P_{km} = frictional pressure drop, kPa/km
Sg = liquid specific gravity (water = 1.00)
C = Hazen-Williams C factor, dimensionless

Example 6.18 Gasoline (specific gravity = 0.74 and viscosity = 0.7 cSt) flows through an NPS 16 (0.250-in wall thickness) pipeline at 4000 gal/min. Using the Hazen-Williams equation with a C factor of 150, calculate the pressure loss due to friction in a mile of pipe.

Solution The flow rate is

$$Q = 4000 \text{ gal/min} = \frac{4000 \times 60}{42} \text{ bbl/h} = 5714.29 \text{ bbl/h}$$

The NPS 16 (0.25-in wall thickness) pipeline has an inside diameter = $16 - 2 \times 0.25 = 15.5$ in

$$P_m = 12{,}352 \left(\frac{5714.29}{150}\right)^{1.852} \frac{0.74}{15.5^{4.87}} \quad \text{psi/mi} \qquad \text{from Eq. (6.58)}$$

Thus the pressure loss due to friction per mile of pipe is 12.35 psi/mi.

Example 6.19 A DN 400 (8-mm wall thickness) steel pipe is used to transport jet fuel (specific gravity = 0.82 and viscosity = 2.0 cSt) from a pumping facility to a storage tank 10 km away. Neglecting differences in elevations, calculate the pressure loss due to friction in bar/km at a flow rate of 700 m^3/h. Use the Hazen-Williams equation with a C factor of 130. If a delivery pressure of 3.5 bar must be maintained at the delivery point and the storage tank is at an elevation of 100 m above that of the pumping facility, calculate the pressure required at the pumping facility at the given flow rate.

Solution The inside diameter = $400 - 2 \times 8 = 384$ mm. Using the Hazen-Williams equation (6.61) we get

$$P_{\text{km}} = (1.1101 \times 10^{13}) \left(\frac{700}{130}\right)^{1.852} \times \frac{0.82}{(384)^{4.87}}$$

$$= 53.40 \text{ kPa/km}$$

Pressure loss due to friction = 53.4 kPa/km = 0.534 bar/km

Total pressure drop in
10 km of pipe length = $0.534 \times 10 = 5.34$ bar

The pressure required at the pumping facility is calculated by adding the pressure drop due to friction to the delivery pressure required and the static elevation head between the pumping facility and storage tank.

$$P_t = P_f + P_{\text{elev}} + P_{\text{del}} \tag{6.62}$$

where P_t = total pressure required at pump
P_f = friction pressure
P_{elev} = pressure head due to elevation difference
P_{del} = delivery pressure at storage tank

$$P_t = 5.34 + \frac{100 \times 1.0/0.102}{100} + 3.5 = 18.64 \text{ bar}$$

Therefore the pressure required at the pumping facility is 18.64 bar, or 1864 kPa.

6.11.6 Miller equation

The Miller equation, or the Benjamin Miller formula, is used for calculating pressure drop in crude oil pipelines. Unlike the Colebrook-White equation this formula does not use the pipe roughness. It can be used to calculate the flow rate for a given pipe size and liquid properties, given the pressure drop due to friction. One form of the Miller equation is as follows:

$$Q = 4.06M\left(\frac{D^5 P_m}{\mathrm{Sg}}\right)^{0.5} \tag{6.63}$$

where the parameter M is defined as

$$M = \log_{10}\left(\frac{D^3 \mathrm{Sg} P_m}{\nu^2}\right) + 4.35 \tag{6.64}$$

and

where Q = flow rate, bbl/day
D = pipe inside diameter, in
P_m = pressure drop, psi/mi
Sg = liquid specific gravity
ν = liquid viscosity, cP

Rearranging the equation to solve for pressure drop, we get

$$P_m = \frac{0.0607(Q/M)^2 \mathrm{Sg}}{D^5} \tag{6.65}$$

where the symbols are as defined before.

In SI Units, the Miller equation has the following form:

$$Q = (3.996 \times 10^{-6})M\left(\frac{D^5 P_m}{\mathrm{Sg}}\right)^{0.5} \tag{6.66}$$

where the parameter M is calculated from

$$M = \log_{10}\left(\frac{D^3 \mathrm{Sg} P_m}{\nu^2}\right) - 0.4965 \tag{6.67}$$

and

where Q = flow rate, m³/h
D = pipe internal diameter, mm
P_m = frictional pressure drop, kPa/km
Sg = liquid specific gravity
ν = liquid viscosity, cP

Reviewing the Miller equation, we see that to calculate the pressure drop P_m given a flow rate Q is not a straightforward process. The intermediate parameter M depends on the unknown pressure drop P_m. We have to solve the problem by successive iteration. We assume an initial value of the pressure drop P_m (say 5 psi/mi) and calculate a starting value for M. Using this value of M in Eq. (6.65), we calculate the second approximation for pressure drop P_m. Next using this newfound value of P_m we recalculate the new value of M and the process is continued until successive values of the pressure drop P_m are within some tolerance such as 0.001 psi/mi.

Example 6.20 An NPS 18 (0.375-in wall thickness) crude oil pipeline flows at the rate of 5000 bbl/h. Calculate the pressure drop per mile using the Miller equation. Assume the specific gravity of crude oil is 0.892 at 60°F and the viscosity is 20 cSt at 60°F. Compare the results using the Colebrook equation with a pipe roughness of 0.002.

Solution Since the Miller equation requires viscosity in centipoise, calculate that first:

$$\text{Liquid viscosity (cP)} = \text{viscosity (cSt)} \times \text{specific gravity}$$
$$= 20 \times 0.892 = 17.84 \text{ cP}$$

The inside diameter of the pipe is

$$D = 18 - 2 \times 0.375 = 17.25 \text{ in}$$

Assume an initial value for the pressure drop of 10 psi/mi. Next calculate the parameter M from Eq. (6.64).

$$M = \log_{10}\left(\frac{17.25^3 \times 0.892 \times 10}{17.84^2}\right) + 4.35 = 6.5079$$

Substituting this value of M in Eq. (6.65) we calculate the pressure drop as

$$P_m = 0.0607 \times \left(\frac{5000 \times 24}{6.5079}\right)^2 \times \frac{0.892}{17.25^5}$$
$$= 12.05 \text{ psi/mi}$$

Using this value of P_m a new value for M is calculated:

$$M = \log_{10}\left(\frac{17.25^3 \times 0.892 \times 12.05}{17.84^2}\right) + 4.35 = 6.5889$$

Recalculate the pressure drop with this value of M:

$$P_m = 0.0607 \times \left(\frac{5000 \times 24}{6.5889}\right)^2 \times \frac{0.892}{17.25^5} = 11.76 \text{ psi/mi}$$

Continuing the iterations a couple of times more, we get the final answer for $P_m = 11.79$. Thus the pressure drop per mile is 11.79 psi/mi.

Next, for comparison, we calculate the pressure drop using the Colebrook equation.

$$\text{Relative roughness } \frac{e}{D} = \frac{0.002}{17.25} = 0.0001$$

Calculate the Reynolds number from Eq. (6.36):

$$R = 2213.76 \times \frac{5000}{17.25 \times 20} = 32{,}083$$

Using the Colebrook equation (6.51) we get the friction factor f as follows:

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{0.0001}{3.7} + \frac{2.51}{32{,}083\sqrt{f}}\right)$$

Solving for f by successive iteration, we get $f = 0.0234$. Using the Darcy equation (6.48) for pressure drop,

$$P_m = 34.87 \times \frac{0.0234 \times 5000^2 \times 0.892}{(17.25)^5}$$

$$= 11.91 \text{ psi/mi}$$

Therefore the pressure drop per mile using the Colebrook equation is 11.91 psi/mi. This compares with a pressure drop of 11.79 psi/mi using the Miller formula.

6.11.7 Shell-MIT equation

The Shell-MIT equation, also known as the MIT equation, was initially used by the Shell pipeline company for modeling the flow of high-viscosity heated crude oil pipelines. This equation for pressure drop uses a modified Reynolds number R_m, which is a multiple of the normal Reynolds number. From R_m a friction factor is calculated depending on whether the flow is laminar or turbulent. The calculation method is as follows. The Reynolds number of flow is first calculated from

$$R = \frac{92.24Q}{D\nu} \tag{6.68}$$

From the preceding, a modified Reynolds number is defined as

$$R_m = \frac{R}{7742} \tag{6.69}$$

where R = Reynolds number, dimensionless
R_m = modified Reynolds number, dimensionless
Q = flow rate, bbl/day
D = pipe inside diameter, in
ν = liquid kinematic viscosity, cSt

Next, a friction factor is calculated from one of the following equations:

$$f = \begin{cases} \dfrac{0.00207}{R_m} & \text{for laminar flow} \quad (6.70) \\ 0.0018 + 0.00662\left(\dfrac{1}{R_m}\right)^{0.355} & \text{for turbulent flow} \quad (6.71) \end{cases}$$

The laminar flow limit is the same as before: Reynolds number $R < 2100$ approximately.

The friction factor f in Eqs. (6.70) and (6.71) is not the Darcy friction factor we have used so far with the Colebrook equation. Therefore we cannot directly use it in the Darcy equation (6.45) to calculate the pressure drop.

The pressure drop due to friction with the Shell-MIT equation is then calculated as follows:

$$P_m = \frac{0.241(f\mathrm{Sg}Q^2)}{D^5} \tag{6.72}$$

where P_m = pressure drop due to friction, psi/mi
f = Shell-MIT equation friction factor, dimensionless
Sg = liquid specific gravity
Q = liquid flow rate, bbl/day
D = pipe inside diameter, in

With flow rate in bbl/h, the pressure drop due to friction is calculated using the following modified version of the Darcy equation:

$$P_m = \frac{138.82(f\mathrm{Sg}Q^2)}{D^5} \tag{6.73}$$

where P_m = pressure drop due to friction, psi/mi
f = Shell-MIT equation friction factor, dimensionless
Sg = liquid specific gravity
Q = liquid flow rate, bbl/h
D = pipe inside diameter, in

In SI units the MIT equation is expressed as follows:

$$P_m = (6.2191 \times 10^{10})\frac{f\mathrm{Sg}Q^2}{D^5} \tag{6.74}$$

where P_m = frictional pressure drop, kPa/km
f = Shell-MIT equation friction factor, dimensionless
Sg = liquid specific gravity
Q = liquid flow rate, m^3/h
D = pipe inside diameter, mm

Example 6.21 A 400-mm outside diameter (8-mm wall thickness) crude oil pipeline is used for shipping a heavy crude oil between two storage terminals at a flow rate of 600 m^3/h at 80°C. Calculate, using the MIT equation, the frictional pressure drop assuming the crude oil has a specific gravity of 0.895 and a viscosity of 100 cSt at 80°C. Compare the result using the Moody diagram method.

Solution The inside diameter of pipe $D = 400 - 2 \times 8 = 384$ mm. From Eq. (6.38), the Reynolds number is first calculated:

$$R = \frac{353{,}678 \times 600}{100 \times 384} = 5526$$

Since $R > 2100$, the flow is in the turbulent zone. Calculate the Shell-MIT modified Reynolds number using Eq. (6.69).

$$R_m = \frac{5526}{7742} = 0.7138$$

Calculate the friction factor from Eq. (6.71).

$$\text{Friction factor} = 0.0018 + 0.00662\left(\frac{1}{0.7138}\right)^{0.355} = 0.0093$$

Finally, we calculate the pressure drop from Eq. (6.74) as follows:

$$P_m = (6.2191 \times 10^{10})\frac{0.0093 \times 0.895 \times 600 \times 600}{(384)^5}$$

$$= 22.23 \text{ kPa/km}$$

6.11.8 Other pressure drop equations

Two other equations for friction factor calculations are the Churchill equation and the Swamee-Jain equation. These equations are explicit equations in friction factor calculation, unlike the Colebrook-White equation, which requires solution by trial and error.

Churchill equation. This equation for the friction factor was proposed by Stuart Churchill and published in *Chemical Engineering* magazine in November 1977. It is as follows:

$$f = \left[\left(\frac{8}{R}\right)^{12} + \frac{1}{(A+B)^{3/2}}\right]^{1/12} \tag{6.75}$$

where

$$A = 2.457 \log_e \left[\frac{1}{(7/R)^{0.9} + (0.27e/D)}\right]^{16} \tag{6.76}$$

$$B = \left(\frac{37{,}530}{R}\right)^{16} \tag{6.77}$$

The Churchill equation for the friction factor yields results that compare closely with that obtained using the Colebrook-White equation or the Moody diagram.

Swamee-Jain equation. This is another explicit equation for calculating the friction factor. It was first presented by P. K. Swamee and A. K. Jain in 1976 in *Journal of the Hydraulics Division of ASCE*. This equation is the easiest of all equations for calculating the friction factor. The Swamee-Jain equation is as follows:

$$f = \frac{0.25}{[\log_{10}(e/3.7D + 5.74/R^{0.9})]^2} \tag{6.78}$$

Similar to the Churchill equation friction factor, the Swamee-Jain equation correlates fairly well with the friction factor calculated using the Colebrook-White equation or the Moody diagram.

6.12 Minor Losses

So far, we have calculated the pressure drop per unit length in straight pipe. We also calculated the total pressure drop considering several miles of pipe from a pump station to a storage tank. Minor losses in a petroleum product pipeline are classified as those pressure drops that are associated with piping components such as valves and fittings. Fittings include elbows and tees. In addition there are pressure losses associated with pipe diameter enlargement and reduction. A pipe nozzle exiting from a storage tank will have entrance and exit losses. All these pressure drops are called *minor losses*, as they are relatively small compared to friction loss in a straight length of pipe.

Generally, minor losses are included in calculations by using the equivalent length of the valve or fitting or using a resistance factor or K factor multiplied by the velocity head $v^2/2g$. The term minor losses can be applied only where the pipeline lengths and hence the friction losses are relatively large compared to the pressure drops in the fittings and valves. In a situation such as plant piping and tank farm piping the pressure drop in the straight length of pipe may be of the same order of magnitude as that due to valves and fittings. In such cases the term minor losses is really a misnomer. Regardless, the pressure losses through valves, fittings, etc., can be accounted for approximately using the equivalent length or K times the velocity head method. It must be noted that this way of calculating the minor losses is valid only in turbulent flow. No data are available for laminar flow.

6.12.1 Valves and fittings

If Table 6.5 shows the equivalent lengths of commonly used valves and fittings in a petroleum pipeline system. It can be seen from this table

TABLE 6.5 Equivalent Lengths of Valves and Fittings

Description	L/D
Gate valve	8
Globe valve	340
Angle valve	55
Ball valve	3
Plug valve straightway	18
Plug valve 3-way through-flow	30
Plug valve branch flow	90
Swing check valve	100
Lift check valve	600
Standard elbow	
90°	30
45°	16
Long radius 90°	16
Standard tee	
Through-flow	20
Through-branch	60
Miter bends	
$\alpha = 0$	2
$\alpha = 30$	8
$\alpha = 60$	25
$\alpha = 90$	60

that a gate valve has an L/D ratio of 8 compared to straight pipe. Therefore, a 20-in-diameter gate valve may be replaced with a $20 \times 8 =$ 160-in-long piece of pipe that will match the frictional pressure drop through the valve.

Example 6.22 A piping system is 2000 ft of NPS 20 pipe that has two 20-in gate valves, three 20-in ball valves, one swing check valve, and four 90° standard elbows. Using the equivalent length concept, calculate the total pipe length that will include all straight pipe and valves and fittings.

Solution Using Table 6.5, we can convert all valves and fittings in terms of 20-in pipe as follows:

$$\text{Two 20-in gate valves} = 2 \times 20 \times 8 = 320 \text{ in of 20-in pipe}$$

$$\text{Three 20-in ball valves} = 3 \times 20 \times 3 = 180 \text{ in of 20-in pipe}$$

$$\text{One 20-in swing check valve} = 1 \times 20 \times 50 = 1000 \text{ in of 20-in pipe}$$

$$\text{Four 90° elbows} = 4 \times 20 \times 30 = 2400 \text{ in of 20-in pipe}$$

$$\text{Total for all valves and fittings} = 4220 \text{ in of 20-in pipe}$$

$$= 351.67 \text{ ft of 20-in pipe}$$

Adding the 2000 ft of straight pipe, the total equivalent length of straight pipe and all fittings is

$$L_e = 2000 + 351.67 = 2351.67 \text{ ft}$$

The pressure drop due to friction in the preceding piping system can now be calculated based on 2351.67 ft of pipe. It can be seen in this example that the valves and fittings represent roughly 15 percent of the total pipeline length. In plant piping this percentage may be higher than that in a long-distance petroleum pipeline. Hence, the reason for the term *minor losses*.

Another approach to accounting for minor losses is using the resistance coefficient or K factor. The K factor and the velocity head approach to calculating pressure drop through valves and fittings can be analyzed as follows using the Darcy equation. From the Darcy equation (6.45), the pressure drop in a straight length of pipe is given by

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

The term $f(L/D)$ may be substituted with a head loss coefficient K (also known as the resistance coefficient) and the preceding equation then becomes

$$h = K\frac{v^2}{2g} \qquad (6.79)$$

In Eq. (6.79), the head loss in a straight piece of pipe is represented as a multiple of the velocity head $v^2/2g$. Following a similar analysis, we can state that the pressure drop through a valve or fitting can also be represented by $K(v^2/2g)$, where the coefficient K is specific to the valve or fitting. Note that this method is only applicable to turbulent flow through pipe fittings and valves. No data are available for laminar flow in fittings and valves. Typical K factors for valves and fittings are listed in Table 6.6. It can be seen that the K factor depends on the nominal pipe size of the valve or fitting. The equivalent length, on the other hand, is given as a ratio of L/D for a particular fitting or valve.

From Table 6.6, it can be seen that a 6-in gate valve has a K factor of 0.12, while a 20-in gate valve has a K factor of 0.10. However, both sizes of gate valves have the same equivalent length–to–diameter ratio of 8. The head loss through the 6-in valve can be estimated to be $0.12\,(v^2/2g)$ and that in the 20-in valve is $0.10\,(v^2/2g)$. The velocities in both cases will be different due to the difference in diameters.

If the flow rate was 1000 gal/min, the velocity in the 6-in valve will be approximately

$$v_6 = 0.4085\,\frac{1000}{6.125^2} = 10.89 \text{ ft/s}$$

Similarly, at 1000 gal/min, the velocity in the 20-in valve will be approximately

$$v_6 = 0.4085\,\frac{1000}{19.5^2} = 1.07 \text{ ft/s}$$

TABLE 6.6 Friction Loss in Valves—Resistance Coefficient K

Description	L/D	Nominal pipe size, in										
		$\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	2	$2\frac{1}{2}$–3	4	6	8–10	12–16	18–24
Gate valve	8	0.22	0.20	0.18	0.18	0.15	0.15	0.14	0.14	0.12	0.11	0.10	0.10
Globe valve	340	9.20	8.50	7.80	7.50	7.10	6.50	6.10	5.80	5.10	4.80	4.40	4.10
Angle valve	55	1.48	1.38	1.27	1.21	1.16	1.05	0.99	0.94	0.83	0.77	0.72	0.66
Ball valve	3	0.08	0.08	0.07	0.07	0.06	0.06	0.05	0.05	0.05	0.04	0.04	0.04
Plug valve straightway	18	0.49	0.45	0.41	0.40	0.38	0.34	0.32	0.31	0.27	0.25	0.23	0.22
Plug valve 3-way through-flow	30	0.81	0.75	0.69	0.66	0.63	0.57	0.54	0.51	0.45	0.42	0.39	0.36
Plug valve branch flow	90	2.43	2.25	2.07	1.98	1.89	1.71	1.62	1.53	1.35	1.26	1.17	1.08
Swing check valve	50	1.40	1.30	1.20	1.10	1.10	1.00	0.90	0.90	0.75	0.70	0.65	0.60
Lift check valve	600	16.20	15.00	13.80	13.20	12.60	11.40	10.80	10.20	9.00	8.40	7.80	7.22
Standard elbow													
90°	30	0.81	0.75	0.69	0.66	0.63	0.57	0.54	0.51	0.45	0.42	0.39	0.36
45°	16	0.43	0.40	0.37	0.35	0.34	0.30	0.29	0.27	0.24	0.22	0.21	0.19
Long radius 90°	16	0.43	0.40	0.37	0.35	0.34	0.30	0.29	0.27	0.24	0.22	0.21	0.19
Standard tee													
Through-flow	20	0.54	0.50	0.46	0.44	0.42	0.38	0.36	0.34	0.30	0.28	0.26	0.24
Through-branch	60	1.62	1.50	1.38	1.32	1.26	1.14	1.08	1.02	0.90	0.84	0.78	0.72
Mitre bends													
$\alpha = 0$	2	0.05	0.05	0.05	0.04	0.04	0.04	0.04	0.03	0.03	0.03	0.03	0.02
$\alpha = 30$	8	0.22	0.20	0.18	0.18	0.17	0.15	0.14	0.14	0.12	0.11	0.10	0.10
$\alpha = 60$	25	0.68	0.63	0.58	0.55	0.53	0.48	0.45	0.43	0.38	0.35	0.33	0.30
$\alpha = 90$	60	1.62	1.50	1.38	1.32	1.26	1.14	1.08	1.02	0.90	0.84	0.78	0.72

Therefore,

$$\text{Head loss in 6-in gate valve} = \frac{0.12\,(10.89)^2}{64.4} = 0.22\ \text{ft}$$

and

$$\text{Head loss in 20-in gate valve} = \frac{0.10\,(1.07)^2}{64.4} = 0.002\ \text{ft}$$

These head losses appear small since we have used a relatively low flow rate in the 20-in valve. In reality the flow rate in the 20-in valve may be as high as 6000 gal/min and the corresponding head loss will be 0.072 ft.

6.12.2 Pipe enlargement and reduction

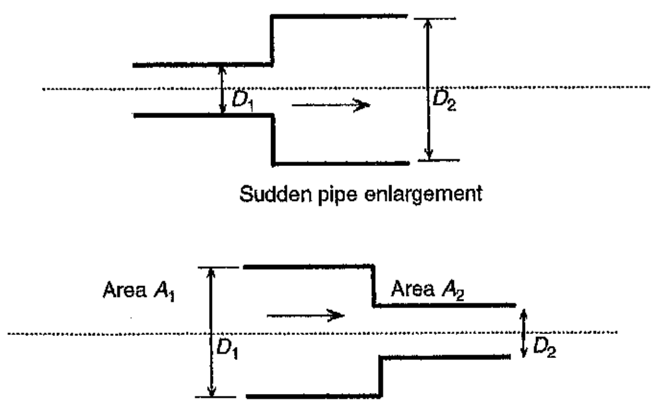
Pipe enlargements and reductions contribute to head loss that can be included in minor losses. For sudden enlargement of pipes, the following head loss equation may be used:

$$h_f = \frac{(v_1 - v_2)^2}{2g} \tag{6.80}$$

where v_1 and v_2 are the velocities of the liquid in the two pipe sizes D_1 and D_2, respectively. Writing Eq. (6.80) in terms of pipe cross-sectional areas A_1 and A_2,

$$h_f = \left(1 - \frac{A_1}{A_2}\right)^2 \frac{{v_1}^2}{2g} \tag{6.81}$$

for sudden enlargement. This is illustrated in Fig. 6.9.



A_1/A_2	0.00	0.10	0.20	0.30	0.40	0.50	0.60	0.70	0.80	0.90	1.00
C_c	0.585	0.624	0.632	0.643	0.659	0.681	0.712	0.755	0.813	0.892	1.000

Figure 6.9 Sudden pipe enlargement and pipe reduction.

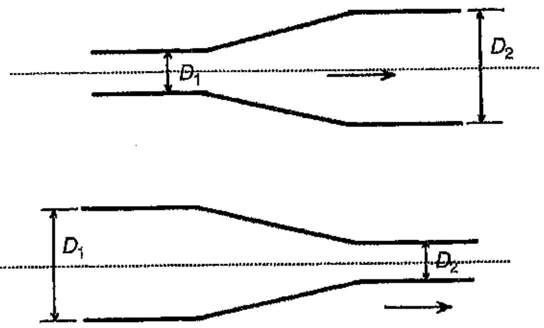


Figure 6.10 Gradual pipe enlargement and pipe reduction.

For sudden contraction or reduction in pipe size as shown in Fig. 6.9, the head loss is calculated from

$$h_f = \left(\frac{1}{C_c} - 1\right)\frac{v_2^{\,2}}{2g} \tag{6.82}$$

where the coefficient C_c depends on the ratio of the two pipe cross-sectional areas A_1 and A_2 as shown in Fig. 6.9.

Gradual enlargement and reduction of pipe size, as shown in Fig. 6.10, cause less head loss than sudden enlargement and sudden reduction. For gradual expansions, the following equation may be used:

$$h_f = \frac{C_c(v_1 - v_2)^2}{2g} \tag{6.83}$$

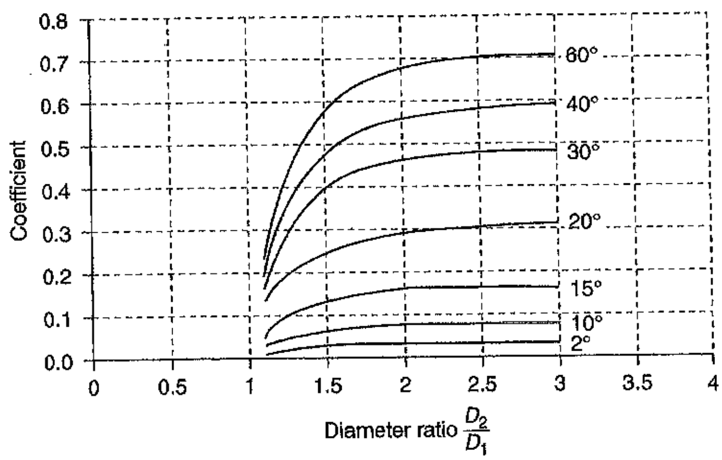


Figure 6.11 Gradual pipe expansion head loss coefficient.

diameter will be selected as the diameter of the first pipe section D_1. Since equivalent length is based on the same pressure drop in the equivalent pipe as the original pipe diameter, we will calculate the equivalent length of section 2 by finding that length of diameter D_1 that will match the pressure drop in a length L_2 of pipe diameter D_2. Using the Darcy equation and converting velocities in terms of flow rate from Eq. (6.31), we can write

$$\text{Head loss} = \frac{f(L/D)(0.4085Q/D_2)^2}{2g}$$

For simplicity, assuming the same friction factor,

$$\frac{L_e}{{D_1}^5} = \frac{L_2}{{D_2}^5}$$

Therefore, the equivalent length of section 2 based on diameter D_1 is

$$L_e = L_2\left(\frac{D_1}{D_2}\right)^5$$

Similarly, the equivalent length of section 3 based on diameter D_1 is

$$L_e = L_3\left(\frac{D_1}{D_3}\right)^5$$

The total equivalent length of all three pipe sections based on diameter D_1 is therefore

$$L_t = L_1 + L_2\left(\frac{D_1}{D_2}\right)^5 + L_3\left(\frac{D_1}{D_3}\right)^5$$

The total pressure drop in the three sections of pipe can now be calculated based on a single pipe of diameter D_1 and length L_t.

Example 6.23 Three pipes with 14-, 16-, and 18-in diameters, respectively, are connected in series with pipe reducers, fittings, and valves as follows:

14-in pipeline, 0.250-in wall thickness, 2000 ft long

16-in pipeline, 0.375-in wall thickness, 3000 ft long

18-in pipeline, 0.375-in wall thickness, 5000 ft long

One 16 × 14 in reducer

One 18 × 16 in reducer

Two 14-in 90° elbows

Four 16-in 90° elbows

Six 18-in 90° elbows

One 14-in gate valve

where C_c depends on the diameter ratio D_2/D_1 and the cone angle β in the gradual expansion. A graph showing the variation of C_c with β and the diameter ratio is shown in Fig. 6.11.

6.12.3 Pipe entrance and exit losses

The K factors for computing the head loss associated with pipe entrance and exit are as follows:

$$K = \begin{cases} 0.5 & \text{for pipe entrance, sharp edged} \\ 1.0 & \text{for pipe exit, sharp edged} \\ 0.78 & \text{for pipe entrance, inward projecting} \end{cases}$$

6.13 Complex Piping Systems

So far we have discussed straight length of pipe with valves and fittings. Complex piping systems include pipes of different diameters in series and parallel configuration.

6.13.1 Series piping

Series piping in its simplest form consists of two or more different pipe sizes connected end to end as illustrated in Fig. 6.12. Pressure drop calculations in series piping may be handled in one of two ways. The first approach would be to calculate the pressure drop in each pipe size and add them together to obtain the total pressure drop. Another approach is to consider one of the pipe diameters as the base size and convert other pipe sizes into equivalent lengths of the base pipe size. The resultant equivalent lengths are added together to form one long piece of pipe of constant diameter equal to the base diameter selected. The pressure drop can now be calculated for this single-diameter pipeline. Of course, all valves and fittings will also be converted to their respective equivalent pipe lengths using the L/D ratios from Table 6.6.

Consider three sections of pipe joined together in series. Using subscripts 1, 2, and 3 and denoting the pipe length as L, inside diameter as D, flow rate as Q, and velocity as V, we can calculate the equivalent length of each pipe section in terms of a base diameter. This base

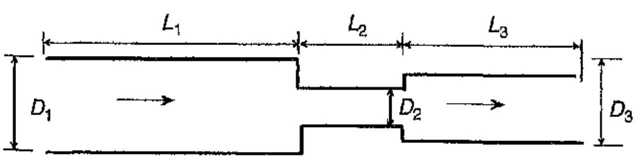


Figure 6.12 Series piping.

One 16-in ball valve

One 18-in gate valve

(a) Use the Hazen-Williams equation with a C factor of 140 to calculate the total pressure drop in the series piping system at a flow rate of 3500 gal/min. The product transported is gasoline with a specific gravity of 0.74. Flow starts in the 14-in piping and ends in the 18-in piping.

(b) If the flow rate is increased to 6000 gal/min, estimate the new total pressure drop in the piping system, keeping everything else the same.

Solution

(a) Since we are going to use the Hazen-Williams equation, the pipes in series analysis will be based on the pressure loss being inversely proportional to $D^{4.87}$, where D is the inside diameter of pipe, per Eq. (6.55).

We will first calculate the total equivalent lengths of all 14-in pipe, fittings, and valves in terms of the 14-in-diameter pipe.

$$\text{Straight pipe: 14 in, 2000 ft} = \text{2000 ft of 14-in pipe}$$

$$\text{Two 14-in } 90^\circ \text{ elbows} = \frac{2 \times 30 \times 14}{12} = \text{70 ft of 14-in pipe}$$

$$\text{One 14-in gate valve} = \frac{1 \times 8 \times 14}{12} = \text{9.33 ft of 14-in pipe}$$

Therefore, the total equivalent length of 14-in pipe, fittings, and valves = 2079.33 ft of 14-in pipe.

Similarly we get the total equivalent length of 16-in pipe, fittings, and valve as follows:

$$\text{Straight pipe: 16-in, 3000 ft} = \text{3000 ft of 16-in pipe}$$

$$\text{Four 16-in } 90^\circ \text{ elbows} = \frac{4 \times 30 \times 16}{12} = \text{160 ft of 16-in pipe}$$

$$\text{One 16-in ball valve} = \frac{1 \times 3 \times 16}{12} = \text{4 ft of 16-in pipe}$$

Therefore, the total equivalent length of 16-in pipe, fittings, and valve = 3164 ft of 16-in pipe.

Finally, we calculate the total equivalent length of 18-in pipe, fittings, and valve as follows:

$$\text{Straight pipe: 18-in, 5000 ft} = \text{5000 ft of 18-in pipe}$$

$$\text{Six 18-in } 90^\circ \text{ elbows} = \frac{6 \times 30 \times 18}{12} = \text{270 ft of 18-in pipe}$$

$$\text{One 18-in gate valve} = \frac{1 \times 8 \times 18}{12} = \text{12 ft of 18-in pipe}$$

Therefore, the total equivalent length of 18-in pipe, fittings, and valve = 5282 ft of 18-in pipe.

Next we convert all the preceding pipe lengths to the equivalent 14-in pipe based on the fact that the pressure loss is inversely proportional to $D^{4.87}$, where D is the inside diameter of pipe.

$$2079.33 \text{ ft of 14-in pipe} = 2079.33 \text{ ft of 14-in pipe}$$

$$3164 \text{ ft of 16-in pipe} = 3164 \times \left(\frac{13.5}{15.25}\right)^{4.87} = 1748 \text{ ft of 14-in pipe}$$

$$5282 \text{ ft of 18-in pipe} = 5282 \times \left(\frac{13.5}{17.25}\right)^{4.87} = 1601 \text{ ft of 14-in pipe}$$

Therefore adding all the preceding lengths we get

$$\text{Total equivalent length in terms of 14-in pipe} = 5429 \text{ ft of 14-in pipe}$$

We still have to account for the 16 × 14 in and 18 × 16 in reducers. The reducers can be considered as sudden enlargements for the approximate calculation of the head loss, using the K factor and velocity head method. For sudden enlargements, the resistance coefficient K is found from

$$K = \left[1 - \left(\frac{d_1}{d_2}\right)^2\right]^2$$

where d_1 is the smaller diameter and d_2 is the larger diameter.

For the 16 × 14 in reducer,

$$K = \left[1 - \left(\frac{13.5}{15.25}\right)^2\right]^2 = 0.0468$$

and for the 18 × 16 in reducer,

$$K = \left[1 - \left(\frac{15.25}{17.25}\right)^2\right]^2 = 0.0477$$

The head loss through the reducers will then be calculated based on $K(V^2/2g)$.

Flow velocities in the three different pipe sizes at 3500 gal/min will be calculated using Eq. (6.31):

$$\text{Velocity in 14-in pipe: } V_{14} = \frac{0.4085 \times 3500}{(13.5)^2} = 7.85 \text{ ft/s}$$

$$\text{Velocity in 16-in pipe: } V_{16} = \frac{0.4085 \times 3500}{(15.25)^2} = 6.15 \text{ ft/s}$$

$$\text{Velocity in 18-in pipe: } V_{18} = \frac{0.4085 \times 3500}{(17.25)^2} = 4.81 \text{ ft/s}$$

The head loss through the 16 × 14 in reducer is

$$h_1 = 0.0468\,\frac{7.85^2}{64.4} = 0.0448\text{ ft}$$

and the head loss through the 18 × 16 in reducer is

$$h_1 = 0.0477\,\frac{6.15^2}{64.4} = 0.028\text{ ft}$$

These head losses are insignificant and hence can be neglected in comparison with the head loss in straight length of pipe. Therefore, the total head loss in the entire piping system will be based on a total equivalent length of 5429 ft of 14-in pipe.

Using the Hazen-Williams equation (6.59) the pressure drop at 3500 gal/min (equal to 3500/0.7 bbl/h) is

$$P_f = 2339\left(\frac{5000}{140}\right)^{1.852}\frac{0.74}{(13.5)^{4.87}} = 4.07\text{ psi per 1000 ft of pipe}$$

Therefore, for the 5429 ft of equivalent 14-in pipe, the total pressure drop is

$$P_f = 4.07\,\frac{5429}{1000} = 22.1\text{ psi}$$

(b) When the flow rate is increased to 6000 gal/min, we can use proportions to estimate the new total pressure drop in the piping as follows:

$$P_f = \left(\frac{6000}{3500}\right)^{1.852} \times 4.07 = 11.04\text{ psi per 1000 ft of pipe}$$

Therefore, the total pressure drop in 5429 ft of 14-in. pipe is

$$P_f = 11.04 \times \frac{5429}{1000} = 59.94\text{ psi}$$

Example 6.24 Two pipes with 400- and 600-mm diameters, respectively, are connected in series with pipe reducers, fittings, and valves as follows:

400-mm pipeline, 6-mm wall thickness, 600 m long

600-mm pipeline, 10-mm wall thickness, 1500 m long

One 600 × 400 mm reducer

Two 400-mm 90° elbows

Four 600-mm 90° elbows

One 400-mm gate valve

One 600-mm gate valve

Use the Hazen-Williams equation with a C factor of 120 to calculate the total pressure drop in the series oil piping system at a flow rate of 250 L/s. Liquid specific gravity is 0.82 and viscosity is 2.5 cSt.

Solution The total equivalent length on 400-mm-diameter pipe is the sum of the following:

$$\text{Straight pipe length} = 600 \text{ m}$$

$$\text{Two } 90^\circ \text{ elbows} = \frac{2 \times 30 \times 400}{1000} = 24 \text{ m}$$

$$\text{One gate valve} = \frac{1 \times 8 \times 400}{1000} = 3.2 \text{ m}$$

Thus,

$$\text{Total equivalent length on 400-mm-diameter pipe} = 627.2 \text{ m}$$

The total equivalent length on 600-mm-diameter pipe is the sum of the following:

$$\text{Straight pipe length} = 1500 \text{ m}$$

$$\text{Four } 90^\circ \text{ elbows} = \frac{4 \times 30 \times 600}{1000} = 72 \text{ m}$$

$$\text{One gate valve} = \frac{1 \times 8 \times 600}{1000} = 4.8 \text{ m}$$

Thus,

$$\text{Total equivalent length on 600-mm-diameter pipe} = 1576.8 \text{ m}$$

Reducers will be neglected since they have insignificant head loss. Convert all pipe to 400-mm equivalent diameter.

$$1576.8 \text{ m of 600-mm pipe} = 1576.8 \left(\frac{388}{580}\right)^{4.87}$$

$$= 222.6 \text{ m of 400-mm pipe}$$

$$\text{Total equivalent length on 400-mm-diameter pipe} = 627.2 + 222.6 = 849.8 \text{ m}$$

$$Q = 250 \times 10^{-3} \times 3600 = 900 \text{ m}^3/\text{h}$$

The pressure drop from Eq. (6.61) is

$$P_m = 1.1101 \times 10^{13} \left(\frac{900}{120}\right)^{1.852} \times \frac{0.82}{(388)^{4.87}}$$

$$= 93.79 \text{ kPa/km}$$

$$\text{Total pressure drop} = \frac{93.79 \times 849.8}{1000} = 79.7 \text{ kPa}$$

6.13.2 Parallel piping

Liquid pipelines in parallel configured such that the multiple pipes are connected so that the liquid flow splits into the multiple pipes at

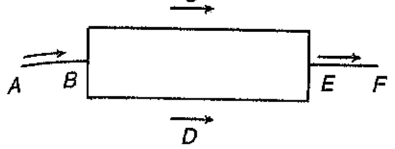


Figure 6.13 Parallel piping.

the beginning and the separate flow streams subsequently rejoin downstream into another single pipe as depicted in Fig. 6.13.

Figure 6.13 shows a parallel piping system in the horizontal plane with no change in pipe elevations. Liquid flows through a single pipe *AB*, and at the junction *B* the flow splits into two pipe branches *BCE* and *BDE*. At the downstream end at junction *E*, the flows rejoin to the initial flow rate and subsequently flow through the single pipe *EF*.

To calculate the flow rates and pressure drop due to friction in the parallel piping system, shown in Fig. 6.13, two main principles of parallel piping must be followed. These are flow conservation at any junction point and common pressure drop across each parallel branch pipe.

Based on flow conservation, at each junction point of the pipeline, the incoming flow must exactly equal the total outflow. Therefore, at junction *B*, the flow *Q* entering the junction must exactly equal the sum of the flow rates in branches *BCE* and *BDE*.

Thus,

$$Q = Q_{BCE} + Q_{BDE} \tag{6.84}$$

where Q_{BCE} = flow through branch *BCE*
Q_{BDE} = flow through branch *BDE*
Q = incoming flow at junction *B*

The other requirement in parallel pipes concerns the pressure drop in each branch piping. Based on this the pressure drop due to friction in branch *BCE* must exactly equal that in branch *BDE*. This is because both branches have a common starting point (*B*) and a common ending point (*E*). Since the pressure at each of these two points is a unique value, we can conclude that the pressure drop in branch pipe *BCE* and that in branch pipe *BDE* are both equal to $P_B - P_E$, where P_B and P_E represent the pressure at the junction points *B* and *E*, respectively.

Another approach to calculating the pressure drop in parallel piping is the use of an equivalent diameter for the parallel pipes. For example in Fig. 6.13, if pipe *AB* has a diameter of 14 in and branches *BCE* and *BDE* have diameters of 10 and 12 in, respectively, we can find some equivalent diameter pipe of the same length as one of the branches that will have the same pressure drop between points B and C as the two branches. An approximate equivalent diameter can be calculated using the Darcy equation.

The pressure loss in branch BCE (10-in diameter) can be calculated as

$$h_1 = \frac{f(L_1/D_1)v_1^2}{2g} \tag{6.85}$$

where the subscript 1 is used for branch BCE and subscript 2 for branch BDE.

Similarly, for branch BDE

$$h_2 = \frac{f(L_2/D_2)v_2^2}{2g} \tag{6.86}$$

For simplicity we have assumed the same friction factors for both branches. Since h_1 and h_2 are equal for parallel pipes, and representing the velocities v_1 and v_2 in terms of the respective flow rates Q_1 and Q_2, using Eq. (6.85) we have the following equations:

$$\frac{f(L_1/D_1)v_1^2}{2g} = \frac{f(L_2/D_2)v_2^2}{2g}$$

$$v_1 = 0.4085\,\frac{Q_1}{D_1^2}$$

$$v_2 = 0.4085\,\frac{Q_2}{D_2^2}$$

In these equations we are assuming flow rates in gal/min and diameters in inches.

Simplifying the equations, we get

$$\frac{L_1}{D_1}\left(\frac{Q_1}{D_1^2}\right)^2 = \frac{L_2}{D_2}\left(\frac{Q_2}{D_2^2}\right)^2$$

or

$$\frac{Q_1}{Q_2} = \left(\frac{L_2}{L_1}\right)^{0.5}\left(\frac{D_1}{D_2}\right)^{2.5} \tag{6.87}$$

Also by conservation of flow

$$Q_1 + Q_2 = Q \tag{6.88}$$

Using Eqs. (6.87) and (6.88), we can calculate the flow through each branch in terms of the inlet flow Q. The equivalent pipe will be designated as D_e in diameter and L_e in length. Since the equivalent pipe will have the same pressure drop as each of the two branches, we can write

$$\frac{L_e}{D_e}\left(\frac{Q_e}{D_e^2}\right)^2 = \frac{L_1}{D_1}\left(\frac{Q_1}{D_1^2}\right)^2 \tag{6.89}$$

where Q_e is the same as the inlet flow Q since both branches have been replaced with a single pipe. In Eq. (6.89), there are two unknowns L_e and D_e. Another equation is needed to solve for both variables. For simplicity, we can set L_e to be equal to one of the lengths L_1 or L_2. With this assumption, we can solve for the equivalent diameter D_e as follows.

$$D_e = D_1\left(\frac{Q}{Q_1}\right)^{0.4} \tag{6.90}$$

Example 6.25 A gasoline pipeline consists of a 2000-ft section of NPS 12 pipe (0.250-in wall thickness) starting at point A and terminating at point B. At point B, two pieces of pipe (4000 ft long each and NPS 10 pipe with 0.250-in wall thickness) are connected in parallel and rejoin at a point D. From D, 3000 ft of NPS 14 pipe (0.250-in wall thickness) extends to point E. Using the equivalent diameter method calculate the pressures and flow rate throughout the system when transporting gasoline (specific gravity = 0.74 and viscosity = 0.6 cSt) at 2500 gal/min. Compare the results by calculating the pressures and flow rates in each branch.

Solution Since the pipe loops between B and D are each NPS 10 and 4000 ft long, the flow will be equally split between the two branches. Each branch pipe will carry 1250 gal/min.

The equivalent diameter for section BD is found from Eq. (6.90):

$$D_e = D_1\left(\frac{Q}{Q_1}\right)^{0.4} = 10.25 \times (2)^{0.4} = 13.525 \text{ in}$$

Therefore we can replace the two 4000-ft NPS 10 pipes between B and D with a single pipe that is 4000 ft long and has a 13.525-in inside diameter.

The Reynolds number for this pipe at 2500 gal/min is found from Eq. (6.35):

$$R = \frac{3162.5 \times 2500}{13.525 \times 0.6} = 974{,}276$$

Considering that the pipe roughness is 0.002 in for all pipes:

$$\text{Relative roughness } \frac{e}{D} = \frac{0.002}{13.525} = 0.0001$$

From the Moody diagram, the friction factor $f = 0.0138$. The pressure drop in section BD is [using Eq. (6.48)]

$$P_m = 71.16\,\frac{0.0138 \times (2500)^2 \times 0.74}{(13.525)^5} = 10.04 \text{ psi/mi}$$

Therefore,

$$\text{Total pressure drop in } BD = \frac{10.04 \times 4000}{5280} = 7.61 \text{ psi}$$

For section AB we have,

$$R = \frac{3162.5 \times 2500}{12.25 \times 0.6} = 1{,}075{,}680$$

$$\text{Relative roughness } \frac{e}{D} = \frac{0.002}{12.25} = 0.0002$$

From the Moody diagram, the friction factor $f = 0.0148$. The pressure drop in section AB is [using Eq. (6.48)]

$$P_m = 71.16 \frac{0.0148 \times (2500)^2 \times 0.74}{(12.25)^5} = 17.66 \text{ psi/mi}$$

Therefore,

$$\text{Total pressure drop in } AB = \frac{17.66 \times 2000}{5280} = 6.69 \text{ psi}$$

Finally, for section DE we have,

$$R = \frac{3162.5 \times 2500}{13.5 \times 0.6} = 976{,}080$$

$$\text{Relative roughness } \frac{e}{D} = \frac{0.002}{13.5} = 0.0001$$

From the Moody diagram, the friction factor $f = 0.0138$. The pressure drop in section DE is [using Eq. (6.48)]

$$P_m = 71.16 \frac{0.0138 \times (2500)^2 \times 0.74}{(13.5)^5} = 10.13 \text{ psi/mi}$$

Therefore,

$$\text{Total pressure drop in } DE = \frac{10.13 \times 3000}{5280} = 5.76 \text{ psi}$$

Finally,

$$\begin{aligned}\text{Total pressure drop in entire piping system} &= 6.69 + 7.61 + 5.76 \\ &= 20.06 \text{ psi}\end{aligned}$$

Next for comparison we will analyze the branch pressure drops considering each branch separately flowing at 1250 gal/min.

$$R = \frac{3162.5 \times 1250}{10.25 \times 0.6} = 642{,}785$$

$$\text{Relative roughness } \frac{e}{D} = \frac{0.002}{10.25} = 0.0002$$

From the Moody diagram, the friction factor $f = 0.015$. The pressure drop in section BD is [using Eq. (6.48)]

$$P_m = 71.16\,\frac{0.015 \times (1250)^2 \times 0.74}{(10.25)^5} = 10.65 \text{ psi/mi}$$

This compares with the pressure drop of 10.04 psi/mi we calculated using an equivalent diameter of 13.525. It can be seen that the difference between the two pressure drops is approximately 6 percent.

Example 6.26 A 5000-m-long crude oil pipeline is composed of three sections A, B, and C. Section A has a 200-m inside diameter and is 1500 m long. Section C has a 400-mm inside diameter and is 2000 m long. The middle section B consists of two parallel pipes each 1500 m long. One of the parallel pipes has a 150-mm inside diameter and the other has a 200-mm inside diameter. Calculate the pressures and flow rates in this piping system at a flow rate of 500 m^3/h. The specific gravity of the liquid is 0.87, the viscosity is 10 cSt, and the pipe roughness is 0.05 mm.

Solution For the center section B, the flow rates will be distributed between the two branches according to Eq. (6.87):

$$\frac{Q_1}{Q_2} = \left(\frac{L_2}{L_1}\right)^{0.5}\left(\frac{D_1}{D2}\right)^{2.5} = 1 \times \left(\frac{200}{150}\right)^{2.5}$$
$$= 2.053$$

Also

$$Q_1 + Q_2 = Q = 500$$

Solving for Q_1 and Q_2, we get

$$Q_1 = 336.23 \text{ m}^3\text{/h and } Q_2 = 163.77 \text{ m}^3\text{/h}$$

Therefore the flow rates in section B are 336.23 m^3/h through 200-mm-diameter pipe and 163.77 m^3/h through 150-mm-diameter pipe.

Section A consists of 200-mm-diameter pipe that flows at 500 m^3/h. The Reynolds number from Eq. (6.38) is

$$R = \frac{353{,}678 \times 500}{10 \times 200} = 88{,}420$$

Therefore flow is turbulent.

$$\text{Relative roughness} = \frac{e}{D} = \frac{0.05}{200} = 0.0003 \text{ in}$$

From the Moody diagram the friction factor $f = 0.0195$. The pressure drop from Eq. (6.50) is

$$P_m = 6.2475 \times \frac{10^{10} \times 0.0195 \times (500)^2 \times 0.87}{(200)^5} = 828.04 \text{ kPa/km}$$

Therefore the total pressure drop in section A is

$$\Delta P_a = 1.5 \times 828.04 = 1242 \text{ kPa}$$

Section B consists of 200-mm-diameter pipe that flows at 336.23 m^3/h (one branch). The Reynolds number from Eq. (6.38) is

$$R = \frac{353{,}678 \times 336.23}{10 \times 200} = 59{,}459$$

Therefore flow is turbulent.

$$\text{Relative roughness} = \frac{e}{D} = \frac{0.05}{200} = 0.0003 \text{ in}$$

From the Moody diagram the friction factor $f = 0.0205$. The pressure drop from Eq. (6.50) is

$$P_m = (6.2475 \times 10^{10}) \times \frac{0.0205 \times (336.23)^2 \times 0.87}{(200)^5} = 393.64 \text{ kPa/km}$$

Therefore the total pressure drop in section B is

$$\Delta P_b = 1.5 \times 393.64 = 590.46 \text{ kPa}$$

Finally section C consists of 400-mm-diameter pipe that flows at 500 m^3/h. The Reynolds number from Eq. (6.38) is

$$R = \frac{353{,}678 \times 500}{10 \times 400} = 44{,}210$$

Therefore flow is turbulent.

$$\text{Relative roughness} = \frac{e}{D} = \frac{0.05}{400} = 0.0001 \text{ in}$$

From the Moody diagram the friction factor $f = 0.022$. The pressure drop from Eq. (6.50) is

$$P_m = (6.2475 \times 10^{10}) \times \frac{0.022 \times (500)^2 \times 0.87}{(400)^5} = 29.19 \text{ kPa/km}$$

Therefore the total pressure drop in section C is

$$\Delta P_c = 2.0 \times 29.19 = 58.38 \text{ kPa}$$

Total pressure drop in entire pipeline system $= 1242 + 590.46 + 58.38 = 1891$ kPa.

6.14 Total Pressure Required

So far we have examined the frictional pressure drop in petroleum systems piping consisting of pipe, fittings, valves, etc. We also calculated the total pressure required to pump oil through a pipeline up to a delivery station at an elevated point. The total pressure required at the beginning of a pipeline, for a specified flow rate, consists of three distinct components:

1. Frictional pressure drop
2. Elevation head
3. Delivery pressure

$$P_t = P_f + P_{\text{elev}} + P_{\text{del}} \tag{6.91}$$

The first item is simply the total frictional head loss in all straight pipe, fittings, valves, etc. The second item accounts for the pipeline elevation difference between the origin of the pipeline and the delivery terminus. If the origin of the pipeline is at a lower elevation than that of the pipeline terminus or delivery point, a certain amount of positive pressure is required to compensate for the elevation difference. On the other hand, if the delivery point were at a lower elevation than the beginning of the pipeline, gravity will assist the flow, and the pressure required at the beginning of the pipeline will be reduced by this elevation difference. The third component, delivery pressure at the terminus, simply ensures that a certain minimum pressure is maintained at the delivery point, such as a storage tank.

For example, if an oil pipeline requires 800 psi to compensate for frictional losses and the minimum delivery pressure required is 25 psi, the total pressure required at the beginning of the pipeline is calculated as follows. If there were no elevation difference between the beginning of the pipeline and the delivery point, the elevation head (component 2) is zero. Therefore, the total pressure P_t required is

$$P_t = 800 + 0 + 25 = 825 \text{ psi}$$

Next consider elevation changes. If the elevation at the beginning is 100 ft and the elevation at the delivery point is 600 ft, then

$$P_t = 800 + \frac{(600 - 100) \times 0.82}{2.31} + 25 = 1002.49 \text{ psi}$$

The middle term in this equation represents the static elevation head difference converted to psi. Finally, if the elevation at the beginning is 600 ft and the elevation at the delivery point is 100 ft, then

$$P_t = 800 + \frac{(100 - 600) \times 0.82}{2.31} + 25 = 647.51 \text{ psi}$$

It can be seen from the preceding that the 500-ft advantage in elevation in the final case reduces the total pressure required by approximately 178 psi compared to the situation where there was no elevation difference between the beginning of the pipeline and delivery point (825 psi versus 647.51 psi).

6.14.1 Effect of elevation

The preceding discussion illustrated a liquid pipeline that had a flat elevation profile compared to an uphill pipeline and a downhill pipeline. There are situations where the ground elevation may have drastic peaks and valleys that require careful consideration of the pipeline topography. In some instances, the total pressure required to transport a given volume of liquid through a long pipeline may depend more on the ground elevation profile than on the actual frictional pressure drop. In the preceding we calculated the total pressure required for a flat pipeline as 825 psi and an uphill pipeline to be 1002 psi. In the uphill case the static elevation difference contributed to 17 percent of the total pressure required. Thus the frictional component was much higher than the elevation component. We will examine a case where the elevation differences in a long pipeline dictate the total pressure required more than the frictional head loss.

Example 6.27 A 20-in, 500-mi-long (0.375-in wall thickness) oil pipeline has a ground elevation profile as shown in Fig. 6.14. The elevation at Corona is 600 ft and at Red Mesa is 2350 ft. Calculate the total pressure required at the Corona pump station to transport 200,000 bbl/day of oil (specific gravity = 0.895 and viscosity = 35 cSt) to the Red Mesa storage tanks, with a minimum delivery pressure of 50 psi at Red Mesa.

Use the Colebrook equation for calculation of the friction factor. If the pipeline operating pressure cannot exceed 1400 psi, how many pumping stations besides Corona will be required to transport the given flow rate? Use a pipe roughness of 0.002 in.

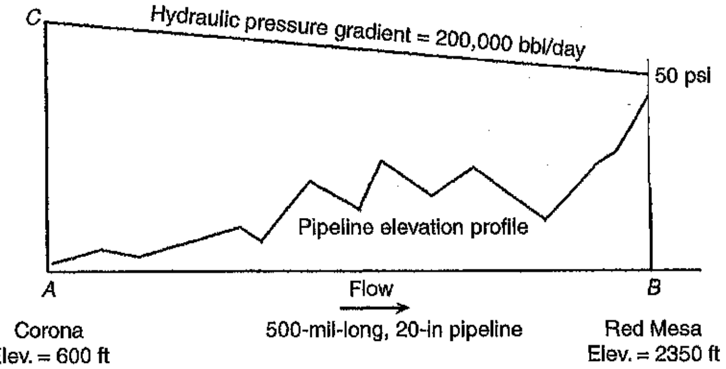


Figure 6.14 Corona to Red Mesa pipeline.

Solution First, calculate the Reynolds number from Eq. (6.37):

$$R = \frac{92.24 \times 200{,}000}{19.25 \times 35} = 27{,}381$$

Therefore the flow is turbulent.

$$\text{Relative pipe roughness} = \frac{e}{D} = \frac{0.002}{19.25} = 0.0001$$

Next, calculate the friction factor f using the Colebrook equation (6.51).

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{0.0001}{3.7} + \frac{2.51}{27{,}381\sqrt{f}}\right)$$

Solving for f by trial and error, $f = 0.0199$. We can now find the pressure loss due to friction using Eq. (6.48) as follows:

$$P_m = 0.0605 \times \frac{0.0199 \times (200{,}000)^2 \times 0.895}{(19.25)^5}$$
$$= 16.31 \text{ psi/mi}$$

The total pressure required at Corona is calculated by adding the pressure drop due to friction to the delivery pressure required at Red Mesa and the static elevation head between Corona and Red Mesa.

$$P_t = P_f + P_{\text{elev}} + P_{\text{del}} \qquad \text{from Eq. (6.91)}$$
$$= (16.31 \times 500) + (2350 - 600) \times \frac{0.895}{2.31} + 50$$
$$= 8155 + 678 + 50 = 8883 \text{ psi}$$

Since a total pressure of 8883 psi at Corona far exceeds the maximum operating pressure of 1400 psi, it is clear that we need additional intermediate booster pump stations besides Corona. The approximate number of pump stations required without exceeding the pipeline pressure of 1400 psi is

$$\text{Number of pump stations} = \frac{8883}{1400} = 6.35, \quad \text{or 7 pump stations}$$

Therefore, we will need six additional booster pump stations besides Corona. With seven pump stations the average pressure per pump station will be

$$\text{Average pump station discharge pressure} = \frac{8883}{7} = 1269 \text{ psi}$$

6.14.2 Tight line operation

When there are drastic elevation differences in a long pipeline, sometimes the last section of the pipeline toward the delivery terminus may operate in an open-channel flow. This means that the pipeline section will not be full of liquid and there will be a vapor space above the liquid.

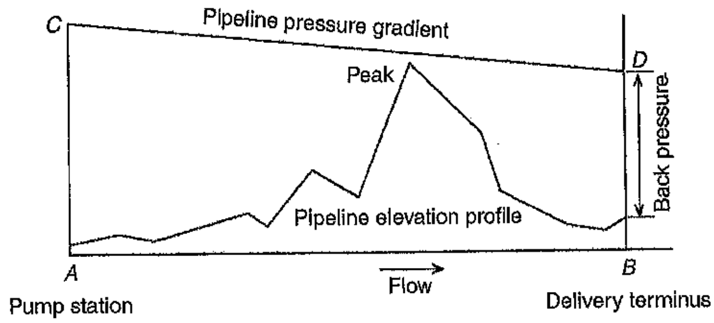


Figure 6.15 Tight line operation.

Such situations are acceptable in ordinary petroleum liquid (gasoline, diesel, and crude oil) pipelines compared to high vapor pressure liquids such as liquefied petroleum gas (LPG) and propane. To prevent such open-channel flow or slack line conditions, we must pack the line by providing adequate back pressure at the delivery terminus as illustrated in Fig. 6.15.

6.15 Hydraulic Gradient

The graphical representation of the pressures along the pipeline, as shown in Fig. 6.16, is the hydraulic gradient. Since elevation is measured in feet, the pipeline pressures are converted to feet of head of liquid and plotted against the distance along the pipeline superimposed on the elevation profile. If we assume a beginning elevation of 100 ft, a delivery terminus elevation of 500 ft, a total pressure of 1000 psi required at the beginning, and a delivery pressure of 25 psi at the terminus, we can plot the hydraulic pressure gradient graphically by the following method.

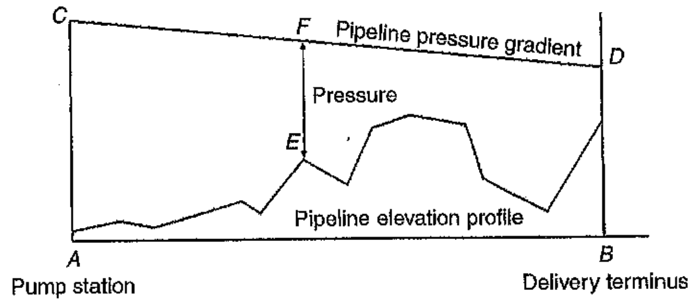


Figure 6.16 Hydraulic gradient.

At the beginning of the pipeline the point C representing the total pressure will be plotted at a height of

$$100\text{ ft} + \frac{1000 \times 2.31}{0.85} = 2818\text{ ft}$$

where the liquid specific gravity is 0.85. Similarly, at the delivery terminus the point D representing the total head at delivery will be plotted at a height of

$$500 + \frac{25 \times 2.31}{0.85} = 568\text{ ft}$$

The line connecting the points C and D represents the variation of the total head in the pipeline and is termed the *hydraulic gradient*. At any intermediate point such as E along the pipeline the pipeline pressure will be the difference between the total head represented by point F on the hydraulic gradient and the actual elevation of the pipeline at E.

If the total head at F is 1850 ft and the pipeline elevation at E is 250 ft, the actual pipeline pressure at E is

$$(1850 - 250)\text{ ft} = \frac{1600 \times 0.85}{2.31} = 589\text{ psi}$$

It can be seen that the hydraulic gradient clears all peaks along the pipeline. If the elevation at E were 2000 ft, we would have a negative pressure in the pipeline at E equivalent to

$$(1850 - 2000)\text{ ft} \quad \text{or} \quad -150\text{ ft} = \frac{-150 \times 0.85}{2.31} = -55\text{ psi}$$

Since a negative pressure is not acceptable, the total pressure at the beginning of the pipeline will have to be higher by 55 psi.

$$\text{Revised total head at } A = 2818 + 150 = 2968\text{ ft}$$

This will result in zero gauge pressure in the pipeline at peak E. The actual pressure in the pipeline will therefore be equal to the atmospheric pressure at that location. Since we would like to always maintain some positive pressure above the atmospheric pressure, in this case the total head at A will be slightly higher than 2968 ft. Assuming a 10-psi positive pressure is desired at the highest peak such as E (2000 ft elevation), the revised total pressure at A would be

$$\text{Total pressure at } A = 1000 + 55 + 10 = 1065\text{ psi}$$

Therefore,

$$\text{Total head at } C = 100 + \frac{1065 \times 2.31}{0.85} = 2994\text{ ft}$$

The difference between 2994 ft and 2968 ft is 26 ft, which is approximately 10 psi.

6.16 Pumping Horsepower

In the previous sections we calculated the total pressure required at the beginning of the pipeline to transport a given volume of liquid over a certain distance. We will now calculate the pumping horsepower (HP) required to accomplish this.

Consider Example 6.27 in which we calculated the total pressure required to pump 200,000 bbl/day of oil from Corona to Red Mesa through a 500-mi-long, 20-in pipeline. We calculated the total pressure required to be 8883 psi. Since the maximum allowable working pressure in the pipeline was limited to 1400 psi, we concluded that six additional pump stations besides Corona were required. With a total of seven pump stations, each pump station would be discharging at a pressure of approximately 1269 psi.

At the Corona pump station oil would enter the pump at some minimum pressure, say 50 psi, and the pumps would boost the pressure to the required discharge pressure of 1269 psi. Effectively, the pumps would add the energy equivalent of (1269 − 50) or 1219 psi at a flow rate of 200,000 bbl/day (5833.33 gal/min). The water horsepower (WHP) required is calculated as

$$\text{WHP} = \frac{(1219 \times 2.31/0.895) \times 5833.33 \times 0.895}{3960} = 4148 \text{ HP}$$

In general the WHP, also known as hydraulic horsepower (HHP), based on 100 percent pump efficiency, is calculated from the following equation:

$$\text{WHP} = \frac{\text{ft of head} \times \text{gal/min} \times \text{liquid specific gravity}}{3960}$$

Assuming a pump efficiency of 80 percent, the pump brake horsepower (BHP) required at Corona is

$$\text{BHP} = \frac{4148}{0.8} = 5185 \text{ HP}$$

The general formula for calculating the BHP of a pump is

$$\text{BHP} = \frac{\text{ft of head} \times \text{gal/min} \times \text{liquid specific gravity}}{3960 \times \text{effy}} \tag{6.92}$$

where effy is the pump efficiency expressed as a decimal value.

If the pump is driven by an electric motor with a motor efficiency of 95 percent, the drive motor HP required will be

$$\text{Motor HP} = \frac{5185}{0.95} = 5458 \text{ HP}$$

The nearest standard size motor of 6000 HP would be adequate for this application. Of course, this assumes that the entire pumping requirement at the Corona pump station is handled by a single pump-motor unit. In reality, to provide for operational flexibility and maintenance two or more pumps will be configured in series or parallel to provide the necessary pressure at the specified flow rate. Let us assume that two pumps are configured in parallel to provide the necessary head pressure of 1219 psi (3146 ft) at the Corona pump station. Each pump will be designed for one-half the total flow rate, or 2917 gal/min, and a pressure of 3146 ft. If the pumps selected had an efficiency of 80 percent, we could calculate the BHP required for each pump as follows:

$$\begin{aligned} \text{BHP} &= \frac{3146 \times 2917 \times 0.895}{3960 \times 0.80} \qquad \text{from Eq. (6.92)} \\ &= 2593 \text{ HP} \end{aligned}$$

Alternatively, if the pumps were configured in series instead of parallel, each pump would be designed for the full flow rate of 5833.33 gal/min but at half the total head required or 1573 ft. The BHP required per pump will still be the same as for the parallel configuration. Pumps are discussed in more detail in Sec. 6.17.

6.17 Pumps

Pumps are installed on petroleum products pipelines to provide the necessary pressure at the beginning of the pipeline to compensate for pipe friction and any elevation head and provide the necessary delivery pressure at the pipeline terminus. Pumps used on petroleum pipelines are either positive displacement (PD) type or centrifugal pumps.

PD pumps generally have higher efficiency, higher maintenance cost, and a fixed volume flow rate at any pressure within allowable limits. Centrifugal pumps on the other hand are more flexible in terms of flow rates but have lower efficiency and lower operating and maintenance cost. The majority of liquid pipelines today are driven by centrifugal pumps.

Since pumps are designed to produce pressure at a given flow rate, an important characteristic of a pump is its performance curve. The performance curve is a graphic representation of how the pressure generated by a pump varies with its flow rate. Other parameters, such as efficiency and horsepower, are also considered as part of a pump performance curve.

6.17.1 Positive displacement pumps

Positive displacement (PD) pumps include piston pumps, gear pumps, and screw pumps. These are used generally in applications where a constant volume of liquid must be pumped against a fixed or variable pressure.

PD pumps can effectively generate any amount of pressure at the fixed flow rate, which depends on its geometry, as long as equipment pressure limits are not exceeded. Since a PD pump can generate any pressure required, we must ensure that proper pressure control devices are installed to prevent rupture of the piping on the discharge side of the PD pump. As indicated earlier, PD pumps have less flexibility with flow rates and higher maintenance cost. Because of these reasons, PD pumps are not popular in long-distance and distribution liquid pipelines. Centrifugal pumps are preferred due to their flexibility and low operating cost.

6.17.2 Centrifugal pumps

Centrifugal pumps consist of one or more rotating impellers contained in a casing. The centrifugal force of rotation generates the pressure in the liquid as it goes from the suction side to the discharge side of the pump. Centrifugal pumps have a wide range of operating flow rates with fairly good efficiency. The operating and maintenance cost of a centrifugal pump is lower than that of a PD pump. The performance curves of a centrifugal pump consist of head versus capacity, efficiency versus capacity, and BHP versus capacity. The term *capacity* is used synonymously with flow rate in connection with centrifugal pumps. Also the term *head* is used in preference to pressure when dealing with centrifugal pumps. Figure 6.17 shows a typical performance curve for a centrifugal pump.

Generally, the head-capacity curve of a centrifugal pump is a drooping curve. The highest head is generated at zero flow rate (shutoff head) and the head decreases with an increase in the flow rate as shown in Fig. 6.17. The efficiency increases with flow rate up to the best efficiency point (BEP) after which the efficiency drops off. The BHP calculated using Eq. (6.92) also generally increases with flow rate but may taper off or start decreasing at some point depending on the head-capacity curve.

The head generated by a centrifugal pump depends on the diameter of the pump impeller and the speed at which the impeller runs. The affinity laws of centrifugal pumps may be used to determine pump performance at different impeller diameters and pump speeds. These laws can be mathematically stated as follows:

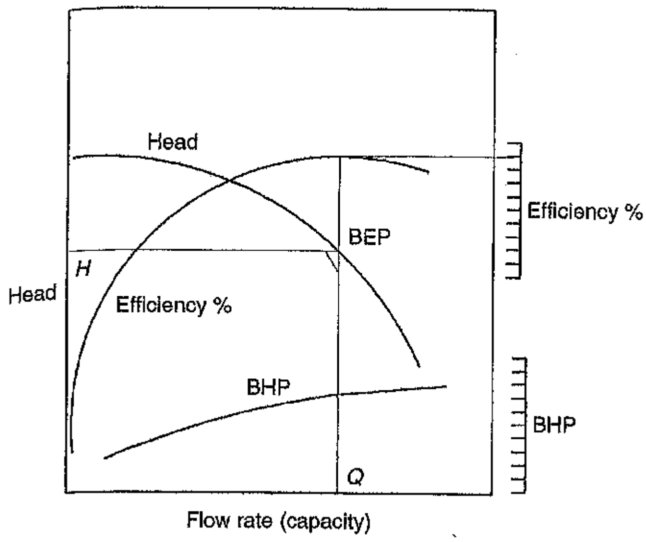


Figure 6.17 Performance curve for centrifugal pump.

For impeller diameter change:

$$\text{Flow rate:} \quad \frac{Q_1}{Q_2} = \frac{D_1}{D_2} \tag{6.93}$$

$$\text{Head:} \quad \frac{H_1}{H_2} = \left(\frac{D_1}{D_2}\right)^2 \tag{6.94}$$

$$\text{BHP:} \quad \frac{\text{BHP}_1}{\text{BHP}_2} = \left(\frac{D_1}{D_2}\right)^3 \tag{6.95}$$

For impeller speed change:

$$\text{Flow rate:} \quad \frac{Q_1}{Q_2} = \frac{N_1}{N_2} \tag{6.96}$$

$$\text{Head:} \quad \frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2 \tag{6.97}$$

$$\text{BHP:} \quad \frac{\text{BHP}_1}{\text{BHP}_2} = \left(\frac{N_1}{N_2}\right)^3 \tag{6.98}$$

where subscript 1 refers to initial conditions and subscript 2 to final conditions. It must be noted that the affinity laws for impeller diameter change are accurate only for small changes in diameter. However, the affinity laws for impeller speed change are accurate for a wide range of impeller speeds.

Using the affinity laws, if the performance of a centrifugal pump is known at a particular diameter, the corresponding performance at a slightly smaller diameter or slightly larger diameter can be calculated very easily. Similarly, if the pump performance for a 10-in impeller at 3500 revolutions per minute (r/min) impeller speed is known, we can easily calculate the performance of the same pump at 4000 r/min.

Example 6.28 The performance of a centrifugal pump with a 10-in impeller is as shown in the following table.

Capacity Q, gal/min	Head H, ft	Efficiency E, %
0	2355	0
1600	2340	57.5
2400	2280	72.0
3200	2115	79.0
3800	1920	80.0
4000	1845	79.8
4800	1545	76.0

(a) Determine the revised pump performance with a reduced impeller size of 9 in.

(b) If the given performance is based on an impeller speed of 3560 r/min, calculate the revised performance at an impeller speed of 3000 r/min.

Solution

(a) The ratio of impeller diameters is $\frac{9}{10} = 0.9$. Therefore, the Q values will be multiplied by 0.9 and the H values will be multiplied by $0.9 \times 0.9 = 0.81$. Revised performance data are given in the following table.

Capacity Q, gal/min	Head H, ft	Efficiency E, %
0	1907	0
1440	1895	57.5
2160	1847	72.0
2880	1713	79.0
3420	1555	80.0
3600	1495	79.8
4320	1252	76.0

(b) When speed is changed from 3560 to 3300 r/min, the speed ratio $= 3000/3560 = 0.8427$. Therefore, Q values will be multiplied by 0.8427 and H values will be multiplied by $(0.8247)^2 = 0.7101$. Therefore, the revised pump performance is as shown in the following table.

Capacity Q, gal/min	Head H, ft	Efficiency E, %
0	1672	0
1348	1662	57.5
2022	1619	72.0
2697	1502	79.0
3202	1363	80.0
3371	1310	79.8
4045	1097	76.0

Example 6.29 For the same pump performance described in Example 6.28, calculate the impeller trim necessary to produce a head of 2000 ft at a flow rate of 3200 gal/min. If this pump had a variable-speed drive and the given performance was based on an impeller speed of 3560 r/min, what speed would be required to achieve the same design point of 2000 ft of head at a flow rate of 3200 gal/min?

Solution Using the affinity laws, the diameter required to produce 2000 ft of head at 3200 gal/min is as follows:

$$\left(\frac{D}{10}\right)^2 = \frac{2000}{2115}$$

$$D = 10 \times 0.9724 = 9.72 \text{ in}$$

The speed ratio can be calculated from

$$\left(\frac{N}{3560}\right)^2 = \frac{2000}{2115}$$

Solving for speed,

$$N = 3560 \times 0.9724 = 3462 \text{ r/min}$$

Strictly speaking, this approach is only approximate since the affinity laws have to be applied along iso-efficiency curves. We must create the new H-Q curves at the reduced impeller diameter (or speed) to ensure that at 3200 gal/min the head generated is 2000 ft. If not, adjustment must be made to the impeller diameter (or speed). This is left as an exercise for the reader.

6.17.3 Net positive suction head

An important parameter related to the operation of centrifugal pumps is the net positive suction head (NPSH). This represents the absolute minimum pressure at the suction of the pump impeller at the specified flow rate to prevent pump cavitation. Below this value the pump impeller may be damaged and render the pump useless. The calculation of NPSH available for a particular pump and piping configuration requires knowledge of the pipe size on the suction side of the pump, the elevation of the liquid source and the pump impeller, along with the atmospheric pressure and vapor pressure of the liquid being pumped. This will be illustrated using an example.

Example 6.30 Figure 6.18 shows a centrifugal pump installation where liquid is pumped out of a storage tank which is located at an elevation of 25 ft above that of the centerline of the pump. The piping from the storage tank to the pump suction consists of straight pipe, valves, and fittings. Calculate the NPSH available at a flow rate of 3200 gal/min. The liquid being pumped has a specific gravity of 0.825 and a viscosity of 15 cSt. If flow rate increases to 5000 gal/min, what is the new NPSH available?

Solution The NPSH available is calculated as follows:

$$\text{NPSH} = (P_a - P_v)\frac{2.31}{\text{Sg}} + H + E_1 - E_2 - h_f \tag{6.99}$$

where P_a = atmospheric pressure, psi
P_v = liquid vapor pressure at flowing temperature, psia
Sg = liquid specific gravity
H = liquid head in tank, ft
E_1 = elevation of tank bottom, ft
E_2 = elevation of pump suction, ft
h_f = friction loss in suction piping from tank to pump suction, ft

All terms in Eq. (6.99) are known except the head loss h_f. This item must be calculated considering the flow rate, pipe size, and liquid properties. The Reynolds number at 3200 gal/min in the 16-in pipe, using Eq. (6.35), is

$$R = \frac{3162.5 \times 3200}{15.5 \times 15} = 43{,}527$$

The friction factor will be found from the Moody diagram. Assume the pipe absolute roughness is 0.002 in. Then

$$\text{Relative roughness } \frac{e}{D} = \frac{0.002}{15.5} = 0.0001$$

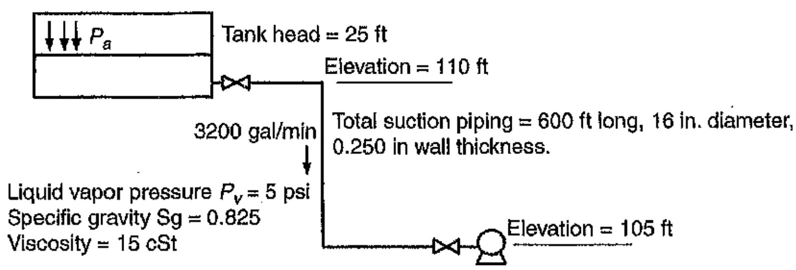


Figure 6.18 NPSH calculations.

From the Moody diagram $f = 0.0215$. The flow velocity from Eq. (6.31) is

$$v = \frac{0.4085 \times 3200}{(15.5)^2} = 5.44 \text{ ft/s}$$

The pressure loss in the suction piping from the tank to the pump will be calculated using the Darcy equation (6.47):

$$h_f = \frac{0.1863\, fLv^2}{D}$$

$$= \frac{0.1863 \times 0.0215 \times 600 \times (5.44)^2}{15.5} = 4.59 \text{ ft}$$

Substituting these values in Eq. (6.99), we obtain

$$\text{NPSH} = (14.73 - 5) \times \frac{2.31}{0.825} + 25 + 110 - 105 - 4.59$$

$$= 27.24 + 25 + 110 - 105 - 4.59 = 52.65$$

The required NPSH for the pump must be less than this value. If the flow rate increases to 5000 gal/min and the liquid level in turn drops to 1 ft, the revised NPSH available is calculated as follows. With flow rate increasing from 3200 to 5000 gal/min, the head loss due to friction h_f is approximately,

$$h_f = \left(\frac{5000}{3200}\right)^2 \times 4.59 = 11.2 \text{ ft}$$

Therefore,

$$\text{NPSH} = 27.24 + 1 + 110 - 105 - 11.2 = 22.04 \text{ ft}$$

It can be seen that the NPSH available dropped off dramatically with the reduction in liquid level in the tank and the increased friction loss in the suction piping at the higher flow rate.

The required NPSH for the pump (based on vendor data) must be lower than the available NPSH calculations just obtained. If the pump data show 30 ft NPSH is required at 5000 gal/min, the preceding calculation indicates that the pump will cavitate since the NPSH available is only 22.04 ft.

6.17.4 Specific speed

An important parameter related to centrifugal pumps is the specific speed. The *specific speed* of a centrifugal pump is defined as the speed at which a geometrically similar pump must be run such that it will produce a head of 1 ft at a flow rate of 1 gal/min. Mathematically, the specific speed is defined as follows:

$$N_S = \frac{NQ^{1/2}}{H^{3/4}} \tag{6.100}$$

where N_S = specific speed
N = impeller speed, r/min
Q = flow rate, gal/min
H = head, ft

It must be noted that in Eq. (6.100) for specific speed, the capacity Q and head H must be measured at the best efficiency point (BEP) for the maximum impeller diameter of the pump. For a multistage pump the value of the head H must be calculated per stage. It can be seen from Eq. (6.100) that low specific speed is attributed to high head pumps and high specific speed for pumps with low head.

Similar to the specific speed, another term known as *suction specific speed* is also applied to centrifugal pumps. It is defined as follows:

$$N_{SS} = \frac{NQ^{1/2}}{(\text{NPSH}_R)^{3/4}} \tag{6.101}$$

where N_{SS} = suction specific speed
N = impeller speed, r/min
Q = flow rate, gal/min
NPSH_R = NPSH required at best efficiency point

With single or double suction pumps the full capacity Q is used in Eq. (6.101) for specific speed. For double suction pumps one-half the value of Q is used in calculating the suction specific speed.

Example 6.31 Calculate the specific speed of a four-stage double suction centrifugal pump with a 12-in-diameter impeller that runs at 3500 r/min and generates a head of 2300 ft at a flow rate of 3500 gal/min at the BEP. Calculate the suction specific speed of this pump, if the NPSH required is 23 ft.

Solution From Eq. (6.100), the specific speed is

$$N_S = \frac{NQ^{1/2}}{H^{3/4}}$$

$$= \frac{3500\,(3500)^{1/2}}{(2300/4)^{3/4}} = 1763$$

The suction specific speed is calculated using Eq. (6.101).

$$N_{SS} = \frac{NQ^{1/2}}{\text{NPSH}_R{}^{3/4}}$$

$$= \frac{3500\,(3500/2)^{1/2}}{(23)^{3/4}} = 13{,}941$$

6.17.5 Effect of viscosity and gravity on pump performance

Generally pump vendors provide centrifugal pump performance based on water as the pumped liquid. Thus the head versus capacity, efficiency versus capacity, and BHP versus capacity curves for a typical centrifugal pump as shown in Fig. 6.17 is really the performance when pumping water. When pumping a petroleum product, the head generated at a particular flow will be slightly less than that with water. The degree of departure from the water curve depends on the viscosity of the petroleum product. For example, when pumping gasoline, jet fuel, or diesel, the head generated will practically be the same as that obtained with water, since these three liquids do not have appreciably high viscosity compared to water.

Generally, if the viscosity is greater than 10 cSt (50 SSU), the performance with the petroleum product will degrade compared to the water performance. Thus when pumping ANS crude with a viscosity of 200 SSU at 60°F, the head-capacity curve will be located below that for water as shown in Fig. 6.19. The Hydraulic Institute chart can be used to correct the water performance curve of a centrifugal pump when pumping high-viscosity liquid. It must be noted that with a high-viscosity

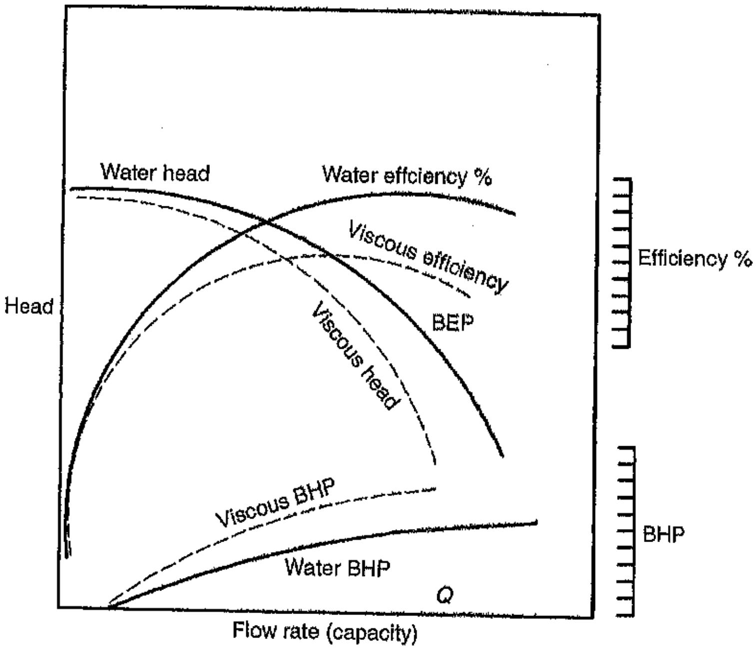


Figure 6.19 Head-capacity curves.

liquid, the pump efficiency degrades faster than the pump head. This can be seen in the comparative performance curve for water and high-viscosity liquid shown in Fig. 6.19.

Several software programs are available to calculate the performance of a centrifugal pump when pumping a high-viscosity liquid. These programs use the Hydraulic Institute chart method to correct the head, efficiency, and BHP from the water performance data. One such program is PUMPCALC published by SYSTEK Technologies, Inc. (*www.systek.us*). Appendix C includes a sample printout and graphic of a viscosity corrected pump performance curve using PUMPCALC.

Positive displacement pumps such as screw pumps and gear pumps tend to perform better with high-viscosity liquids. In fact the higher the viscosity of the pumped liquid, the less would be the slip in these types of pumps. For example, if a screw pump is rated at 5000 gal/min, the volume flow rate will be closer to this number with a 2000-SSU viscosity liquid compared to a 500-SSU viscosity liquid. In contrast centrifugal pump performance degrades from water to 500 SSU viscosity to the lowest performance with the 2000-SSU viscosity liquid.

The BHP required by the pump is a function of the liquid specific gravity, flow rate, head, and pump efficiency [from Eq. (6.92)]. We can therefore conclude that the BHP required increases with higher specific gravity liquids. Thus water (specific gravity = 1.0) may require a BHP of 1500 HP at a particular flow rate. The same pump pumping diesel (specific gravity = 0.85) at the same flow rate and head will require less BHP according to the pump curve. Actually, due to the higher viscosity of diesel (approximately 5.0 cSt compared to that of water at 1.0 cSt) the head required to pump the same volume of diesel will be higher than that of water. From this standpoint the BHP required with diesel will be higher than water. However, when reviewing the pump performance curve, the BHP required is directly proportional to the specific gravity and hence the BHP curve, for diesel will be below that of water. The BHP curve for gasoline will be lower than diesel since gasoline has a specific gravity of 0.74.

6.18 Valves and Fittings

Oil pipelines include several appurtenances as part of the pipeline system. Valves, fittings, and other devices are used in a pipeline system to accomplish certain features of pipeline operations. Valves may be used to communicate between the pipeline and storage facilities as well as between pumping equipment and storage tanks. There are many different types of valves, each performing a specific function. Gate valves and ball valves are used in the main pipeline as well as within pump stations and tank farms. Pressure relief valves are used to protect piping systems and facilities from overpressure due to upsets in operational conditions. Pressure regulators and control valves are used to reduce pressures in certain sections of piping systems as well as when delivering petroleum product to third-party pipelines that may be designed for lower operating pressures. Check valves are found in pump stations and tank farms to prevent backflow as well as separating the suction piping from the discharge side of a pump installation. On long-distance pipelines with multiple pump stations, the pigging process necessitates a complex series of piping and valves to ensure that the pig passes through the pump station piping without getting stuck.

All valves and fittings such as elbows and tees contribute to the frictional pressure loss in a pipeline system. Earlier we referred to some of these head losses as minor losses. As described earlier each valve and fitting is converted to an equivalent length of straight pipe for the purpose of calculating the head loss in the pipeline system.

A control valve functions as a pressure-reducing device and is designed to maintain a specified pressure at the downstream side as shown in Fig. 6.20. If P_1 is the upstream pressure and P_2 the downstream pressure, the control valve is designed to handle a given flow rate Q at these pressures. A coefficient of discharge C_v is typical of the control valve design and is related to the pressures and flow rates by the following equation:

$$Q = C_v A (P_1 - P_2)^{1/2} \tag{6.102}$$

where A is a constant.

Generally, the control valve is selected for a specific application based on P_1, P_2, and Q. For example, a particular situation may require 800 psi upstream pressure, 400 psi downstream pressure, and a flow rate of 3000 gal/min. Based on these numbers, we may calculate a $C_v = 550$. We would then select the correct size of a particular vendor's control valve that can provide this C_v value at a specified flow rate and pressures.

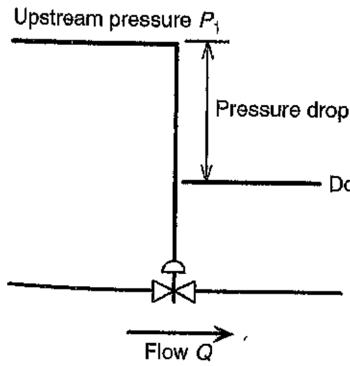


Figure 6.20 Control valve.

For example, a 10-in valve from vendor A may have a C_v of 400, while a 12-in valve may have a $C_v = 600$. Therefore, in this case we would choose a 12-in valve to satisfy our requirement of $C_v = 550$.

6.19 Pipe Stress Analysis

The pipe used to transport petroleum product must be strong enough to withstand the internal pressure necessary to move liquid at the desired flow rate. The wall thickness T necessary to safely withstand an internal pressure of P depends upon the pipe diameter D and yield strength of the pipe material and is generally calculated based upon Barlow's equation:

$$S_h = \frac{PD}{2T} \tag{6.103}$$

where S_h represents the hoop stress in the circumferential direction in the pipe material. Another stress, termed the axial stress or longitudinal stress, acts perpendicular to the cross section of the pipe. The axial stress is one-half the magnitude of the hoop stress. Hence the governing stress is the hoop stress from Eq. (6.103).

Applying a safety factor and including the yield strength of the pipe material, Barlow's equation is modified for use in petroleum piping calculation as follows:

$$P = \frac{2T \times S \times E \times F}{D} \tag{6.104}$$

where P = internal pipe design pressure, psig
D = pipe outside diameter, in
T = pipe wall thickness, in
S = specified minimum yield strength (SMYS) of pipe material, psig
E = seam joint factor = 1.0 for seamless and submerged arc welded (SAW) pipes (see Table 6.7 for other joint types)
F = design factor, usually 0.72 for liquid pipelines

The design factor is sometimes reduced from the 0.72 value in the case of offshore platform piping or when certain city regulations require buried pipelines to be operated at a lower pressure. Equation (6.104) for calculating the internal design pressure is found in the Code of Federal Regulations, Title 49, Part 195, published by the U.S. Department of Transportation (DOT). You will also find reference to this equation in ASME standard B31.4 for design and transportation of liquid pipelines.

In SI units, the internal design pressure equation is the same as shown in Eq. 6.104, except the pipe diameter and wall thickness are in

TABLE 6.7 Pipe Design Joint Factors

Pipe specification	Pipe category	Joint factor E
ASTM A53	Seamless	1.00
	Electric resistance welded	1.00
	Furnace lap welded	0.80
	Furnace butt welded	0.60
ASTM A106	Seamless	1.00
ASTM A134	Electric fusion arc welded	0.80
ASTM A135	Electric Resistance Welded	1.00
ASTM A139	Electric fusion welded	0.80
ASTM A211	Spiral welded pipe	0.80
ASTM A333	Seamless	1.00
ASTM A333	Welded	1.00
ASTM A381	Double submerged arc welded	1.00
ASTM A671	Electric fusion welded	1.00
ASTM A672	Electric fusion welded	1.00
ASTM A691	Electric fusion welded	1.00
API 5L	Seamless	1.00
	Electric resistance welded	1.00
	Electric flash welded	1.00
	Submerged arc welded	1.00
	Furnace lap welded	0.80
	Furnace butt welded	0.60
API 5LX	Seamless	1.00
	Electric resistance welded	1.00
	Electric flash welded	1.00
	Submerged arc welded	1.00
API 5LS	Electric resistance welded	1.00
	Submerged arc welded	1.00

millimeters. The SMYS of pipe material and the internal design pressures are both expressed in kilopascals.

Petroleum pipelines are constructed of steel pipe conforming to American Petroleum Institute (API) standards 5L and 5LX specifications. Some piping may also be constructed of steel pipe conforming to ASTM and ANSI standards. High-strength steel pipe may be designated as API 5LX-52, 5LX-60, or 5LX-80. The last two digits of the pipe specification denote the SMYS of the pipe material. Thus 5LX-52 pipe has a yield strength of 52,000 psi.

Example 6.32 Calculate the allowable internal design pressure for a 16-inch (0.250-in wall thickness) pipeline constructed of API 5LX-52 steel. What wall thickness will be required if an internal working pressure of 1400 psi is required?

Solution Using Eq. (6.104),

$$P = \frac{2 \times 0.250 \times 52{,}000 \times 0.72 \times 1.0}{16} = 1170 \text{ psi}$$

For an internal working pressure of 1400 psi, the wall thickness required is

$$1400 = \frac{2 \times T \times 52{,}000 \times 0.72 \times 1.0}{16}$$

Solving for T, we get

$$\text{Wall thickness } T = 0.299 \text{ in}$$

The nearest standard pipe wall thickness is 0.312 in.

6.20 Pipeline Economics

In pipeline economics we are interested in determining the most economical pipe size and material to be used for transporting a given volume of a petroleum product from a source to a destination. The criterion would be to minimize the capital investment as well as annual operating and maintenance cost. In addition to selecting the pipe itself to handle the flow rate we must also evaluate the optimum size of pumping equipment required. By installing a smaller-diameter pipe we may reduce the pipe material cost and installation cost. However, the smaller pipe size would result in a larger pressure drop due to friction and hence a higher horsepower, which would require larger, more costly pumping equipment. On the other hand, selecting a larger pipe size would increase the capital cost of the pipeline itself but would reduce the pump horsepower required and hence the capital cost of pumping equipment. Larger pumps and motors will also result in increased annual operating and maintenance cost. Therefore, we need to determine the optimum pipe size and pumping power required based on some approach that will minimize both capital investment as well as annual operating costs. The least present value approach, which considers the total capital investment, the annual operating costs over the life of the pipeline, time value of money, borrowing cost, and income tax rate, seems to be an appropriate method in this regard.

Example 6.33 A 25-mi-long crude oil pipeline is used to transport 200,000 bbl/day of light crude (specific gravity = 0.850 and viscosity = 15 cSt) from a pumping station at Parker to a storage tank at Danby. Determine the optimum pipe size for this application based on the least initial cost. Consider three different pipe sizes—NPS 16, NPS 20, and NPS 24. Use the Colebrook-White equation or the Moody diagram for friction factor calculations. Assume the pipeline is on fairly flat terrain. Use 85 percent pump efficiency, \$700 per ton for pipe material cost, and \$1500 per HP for pump station installation cost. The labor costs for installing the three pipe sizes are \$80, \$100, and \$110 per ft. The pipeline will be designed for an operating pressure of 1400 psi. The pipe absolute roughness $e = 0.002$ in.

Solution Based on a 1400-psi design pressure, the wall thickness of NPS 16 pipe will be calculated first. Assuming API 5LX-52 pipe, the wall thickness required for a 1400-psi operating pressure is calculated from Eq. (6.104):

$$T = \frac{1400 \times 16}{2 \times 52{,}000 \times 0.72} = 0.299 \text{ in}$$

The nearest standard size is 0.312 in. The Reynolds number is calculated from Eq. (6.37) as follows:

$$R = \frac{92.24 \times 200{,}000}{15.376 \times 15} = 79{,}986$$

Therefore, the flow is turbulent.

$$\frac{e}{D} = \frac{0.002}{15.376} = 0.0001$$

The friction factor f is found from the Moody diagram as

$$f = 0.0195$$

The pressure drop per mile per Eq. (6.48) is

$$P_m = 0.0605 \times \frac{0.0195 \times (200{,}000)^2 \times 0.85}{(15.376)^5} = 46.67 \text{ psi/mi}$$

$$\text{Total pressure drop in 25 mi} = 25 \times 46.67 = 1167 \text{ psi}$$

Assuming a 50-psi delivery pressure and a 50-psi pump suction pressure,

$$\text{Pump head required at Parker} = \frac{1167 \times 2.31}{0.85} = 3172 \text{ ft}$$

$$\text{Pump flow rate} = \frac{200{,}000 \times 0.7}{24} = 5833.33 \text{ gal/min}$$

$$\text{Pump HP required at Parker} = \frac{3172 \times 5833.33 \times 0.85}{3960 \times 0.85} = 4673 \text{ HP}$$

Therefore a 5000-HP pump unit will be required. Next we will calculate the total pipe required. The total tonnage of NPS 16 pipe is calculated as follows:

$$\text{Pipe weight per ft} = 10.68 \times 0.312(16 - 0.312) = 52.275$$

$$\text{Total pipe tonnage for 25 mi} = \frac{25 \times 52.275 \times 5280}{2000} = 3450 \text{ tons}$$

Increasing this by 5 percent for contingency and considering a material cost of $700 per ton,

$$\text{Total pipe material cost} = 700 \times 3450 \times 1.05 = \$2.54 \text{ million}$$

$$\text{Labor cost for installing NPS 16 pipeline} = 80 \times 25 \times 5280 = \$10.56 \text{ million}$$

$$\text{Pump station cost} = 1500 \times 5000 = \$7.5 \text{ million}$$

Therefore,

$$\text{Total capital cost of NPS 16 pipeline} = \$2.54 + \$10.56 + \$7.5 = \$20.6 \text{ million}$$

Next we calculate the pressure and HP required for the NPS 20 pipeline:

$$T = \frac{1400 \times 20}{2 \times 52{,}000 \times 0.72} = 0.374 \text{ in}$$

The nearest standard size is 0.375 in. The Reynolds number is calculated from Eq. (6.37) as follows:

$$R = \frac{92.24 \times 200{,}000}{19.25 \times 15} = 63{,}889$$

Therefore, the flow is turbulent.

$$\frac{e}{D} = \frac{0.002}{19.25} = 0.0001$$

The friction factor f is found from the Moody diagram as

$$f = 0.020$$

The pressure drop per mile per Eq. (6.48) is

$$P_m = 0.0605 \times \frac{0.020 \times (200{,}000)^2 \times 0.85}{(19.25)^5} = 15.56 \text{ psi/mi}$$

$$\text{Total pressure drop in 25 mi} = 25 \times 15.56 = 389 \text{ psi}$$

Assuming a 50-psi delivery pressure and a 50-psi pump suction pressure,

$$\text{Pump head required at Parker} = \frac{389 \times 2.31}{0.85} = 1057 \text{ ft}$$

$$\text{Pump flow rate} = \frac{200{,}000 \times 0.7}{24} = 5833.33 \text{ gal/min}$$

$$\text{Pump HP required at Parker} = \frac{1057 \times 5833.33 \times 0.85}{3960 \times 0.85} = 1557 \text{ HP}$$

Therefore a 1750-HP pump unit will be required.

Next we will calculate the total pipe required. The total tonnage of NPS 20 pipe is calculated as follows:

$$\text{Pipe weight per ft} = 10.68 \times 0.375\,(20 - 0.375) = 78.6$$

$$\text{Total pipe tonnage for 25 mi} = 25 \times 78.6 \times \frac{5280}{2000} = 5188 \text{ tons}$$

Increasing this by 5 percent for contingency and considering a material cost of $700 per ton,

$$\text{Total pipe material cost} = 700 \times 5188 \times 1.05 = \$3.81 \text{ million}$$

$$\text{Labor cost for installing NPS 20 pipeline} = 100 \times 25 \times 5280 = \$13.2 \text{ million}$$

$$\text{Pump station cost} = 1500 \times 1750 = \$2.63 \text{ million}$$

Therefore,

Total capital cost of
NPS 20 pipeline = \$3.81 + \$13.2 + \$2.63 = \$19.64 million

Next we calculate the pressure and HP required for NPS 24 pipeline.

$$T = \frac{1400 \times 24}{2 \times 52{,}000 \times 0.72} = 0.449 \text{ in}$$

The nearest standard size is 0.500 in. The Reynolds number is calculated from Eq. (6.37) as follows:

$$R = \frac{92.24 \times 200{,}000}{23.0 \times 15} = 53{,}473$$

Therefore, the flow is turbulent.

$$\frac{e}{D} = \frac{0.002}{23.0} = 0.0001$$

The friction factor f is found from the Moody diagram as

$$f = 0.021$$

The pressure drop per mile per Eq. (6.48) is

$$P_m = 0.0605 \times \frac{0.021 \times (200{,}000)^2 \times 0.85}{(23.0)^5} = 6.71 \text{ psi/mi}$$

Total pressure drop
in 25 mi = 25 × 6.71 = 167.8 psi

Assuming a 50-psi delivery pressure and a 50-psi pump suction pressure,

$$\text{Pump head required at Parker} = \frac{167.8 \times 2.31}{0.85} = 456 \text{ ft}$$

$$\text{Pump flow rate} = \frac{200{,}000 \times 0.7}{24} = 5833.33 \text{ gal/min}$$

$$\text{Pump HP required at Parker} = \frac{456 \times 5833.33 \times 0.85}{3960 \times 0.85} = 672 \text{ HP}$$

Therefore an 800-HP pump unit will be required.

Next we will calculate the total pipe required. The total tonnage of NPS 24 pipe is calculated as follows:

$$\text{Pipe weight per ft} = 10.68 \times 0.5\,(24 - 0.5) = 125.5$$

$$\text{Total pipe tonnage for 25 mi} = \frac{25 \times 125.5 \times 5280}{2000} = 8283 \text{ tons}$$

Increasing this by 5 percent for contingency and considering a material cost of \$700 per ton,

Total pipe material cost = 700 × 8283 × 1.05 = \$6.09 million

Labor cost for installing
NPS 24 pipeline = 110 × 25 × 5280 = \$14.52 million

Pump station cost = 1500 × 800 = \$1.2 million

Therefore,

$$\text{Total capital cost of NPS 24 pipeline} = \$6.09 + \$14.52 + \$1.2 = \$21.81 \text{ million}$$

In summary, capital costs of the NPS 16, NPS 20, and NPS 24 pipelines are

$$\text{NPS 16} = \$20.6 \text{ million}$$

$$\text{NPS 20} = \$19.64 \text{ million}$$

$$\text{NPS 24} = \$21.81 \text{ million}$$

Therefore, based on initial cost alone it appears that NPS 20 is the preferred pipe size.

Example 6.34 A 68-mi-long refined petroleum products pipeline is constructed of NPS 24 (0.375-in wall thickness) pipe and is used for transporting 10,000 bbl/h of diesel from Hampton pump station to a delivery tank at Derry. The delivery pressure required at Derry is 30 psi. The elevation at Hampton is 150 ft and at Derry it is 250 ft. Calculate the pumping horsepower required at 80 percent pump efficiency. This pipeline system needs to be expanded to handle increased capacity from 10,000 bbl/h to 20,000 bbl/h. One option would be to install a parallel NPS 24 (0.375-in wall thickness) pipeline and provide upgraded pumps at Hampton. Another option would require expanding the capacity of the existing pipeline by installing an intermediate booster pump station. Determine the more economical alternative for the expansion. Diesel has a specific gravity of 0.85 and a viscosity of 5.5 cSt.

Solution First calculate the Reynolds number from Eq. (6.36):

$$R = \frac{2213.76 \times 10{,}000}{23.25 \times 5.5} = 173{,}119$$

Assuming relative roughness $e/D = 0.0001$, from the Moody diagram we get the friction factor as

$$f = 0.017$$

Pressure drop is calculated using Eq. (6.48).

$$P_m = 34.87 \times \frac{0.017 \times (10{,}000)^2 \times 0.85}{(23.25)^5} = 7.42 \text{ psi/mi}$$

The total pressure required is the sum of friction head, elevation head, and delivery head using Eq. (6.91).

$$P_T = \frac{(68 \times 7.42) + (250 - 150) \times 0.85}{2.31} + 30 = 571.36 \text{ psi}$$

Assuming a 50-psi suction pressure, the pump head required at Hampton is

$$H = \frac{(571.36 - 50) \times 2.31}{0.85} = 1417 \text{ ft}$$

$$\text{Pump flow rate } Q = 10{,}000 \text{ bbl/h} = 7000 \text{ gal/min}$$

Therefore, the pump HP required using Eq. (6.92) is

$$\text{BHP} = \frac{1417 \times 7000 \times 0.85}{3960 \times 0.8} = 2662$$

When the flow rate increases to 20,000 bbl/h from 10,000 bbl/h, the new Reynolds number is

$$R = 2 \times 173{,}119 = 346{,}238$$

Assuming relative roughness $e/D = 0.0001$, from the Moody diagram we get the friction factor as

$$f = 0.0154$$

The pressure drop is calculated using Eq. (6.48):

$$P_m = 34.87 \times \frac{0.0154 \times (20{,}000)^2 \times 0.85}{(23.25)^5} = 26.87 \text{ psi/mi}$$

The total pressure required at Hampton is

$$P_T = (68 \times 26.87) + \frac{(250 - 150) \times 0.85}{2.31} + 30 = 1894 \text{ psi}$$

Since this pressure is higher than a maximum allowable operating pressure (MAOP) of 1400 psi, we will need to install an intermediate booster pump station between Hampton and Derry.

Assuming the total HP required in this case is equally distributed between the two pump stations, we will calculate the pump HP required at each station as follows:

$$\text{Pump station discharge pressure} = \frac{1894 - 50}{2} = 922 \text{ psi}$$

$$\text{Pump head} = \frac{(922 - 50) \times 2.31}{0.85} = 2370 \text{ ft}$$

$$\text{Pump flow rate} = 20{,}000 \text{ bbl/h} = 14{,}000 \text{ gal/min}$$

Therefore, the pump HP required from Eq. (6.92) is

$$\text{BHP} = \frac{2370 \times 14{,}000 \times 0.85}{3960 \times 0.8} = 8903$$

Thus each pump station requires a 9000-HP pump for a total of 18,000 HP.

If we achieve the increased throughput by installing an NPS 24 parallel pipe, the flow through each 24-in pipe will be 10,000 bbl/h, the same as before expansion. Therefore, comparison between the two options of installing a parallel pipe versus adding an intermediate booster pump station must be based on the cost comparison of 68 mi of additional NPS 24 pipe versus increased HP at Hampton and an additional 9000 HP at the new pump station.

Initially, at 10,000 bbl/h, Hampton required 2662, or approximately 3000, HP installed. In the second phase Hampton must be upgraded to 9000 HP and a new 9000-HP booster station must be installed.

$$\text{Incremental HP required for expansion} = 18{,}000 - 3000 = 15{,}000 \text{ HP}$$

$$\text{Capital cost of incremental HP at \$1500 per HP} = 1500 \times 15{,}000 = \$22.5 \text{ million}$$

Compared to installing the booster station, looping the existing NPS 24 line will be calculated on the basis of $700 per ton of pipe material and $100 per ft labor cost.

$$\text{Pipe weight per ft} = 10.68 \times 0.375 \times (24 - 0.375) = 94.62 \text{ lb/ft}$$

$$\text{Material cost for 68 mi of pipe} = \frac{700 \times 94.62 \times 5280 \times 68}{2000} = \$11.9 \text{ million}$$

$$\text{Labor cost for installing 68 mi of NPS 24 pipe} = 68 \times 5280 \times 100 = \$35.9 \text{ million}$$

$$\text{Total cost of NPS 24 pipe loop} = 11.9 + 35.9 = \$47.8 \text{ million}$$

Therefore, based on capital cost alone, it is more economical to install the booster pump station.