ME 423: FLUIDS ENGINEERING

Gas Pipeline Hydraulics

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Compressor Stations

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Compressor Station

Compressor stations are installed on gas pipelines to provide the pressure needed to transport gas from one location to another. Due to limitations of pipeline pressures, multiple compressor stations may be needed to transport a given volume through a long-distance pipeline. The locations and pressures at which these compressor stations operate are determined by allowable **pipe pressures, power available, and environmental and geotechnical factors**.

The maximum pressure that a pipeline can withstand (without being ruptured) is know an **maximum allowable operating pressure (MAOP) of the pipeline.** This MAOP is normally taken as 1200 psig.

Calculations would show that in order to deliver the 100 MMSCFD of gas at Leeds at required terminus pressure of 500 psig, the pressure required at Dover would have to be 1580 psig which is greater that MAOP (1200 psig). Accordingly, we need to install more that one compressor station.

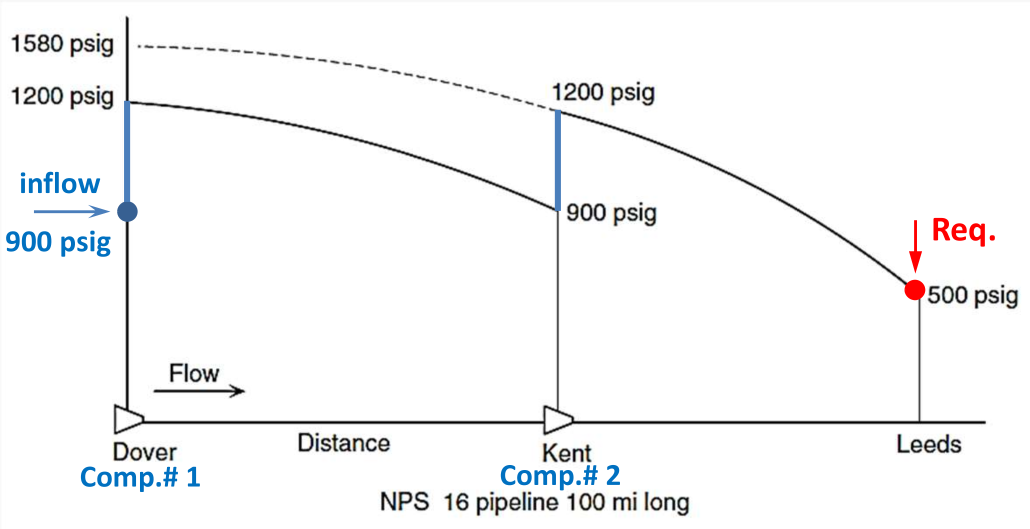


Fig. Gas pipeline with two compressor stations.

Compressor Station

Compression ratio is simply the ratio of the compressor discharge pressure to its suction pressure, both pressures being expressed in absolute units:

$$\text{Compression ratio}, r = \frac{P_d}{P_s}$$

where the suction and discharge pressures P_s and P_d are in absolute units.

An acceptable compression ratio for centrifugal compressors is about 1.5. A larger number requires more compressor horsepower, whereas a smaller compression ratio means less horsepower required.

In gas pipelines, it is desirable to keep the average pipeline pressure as high as possible to reduce compression power.

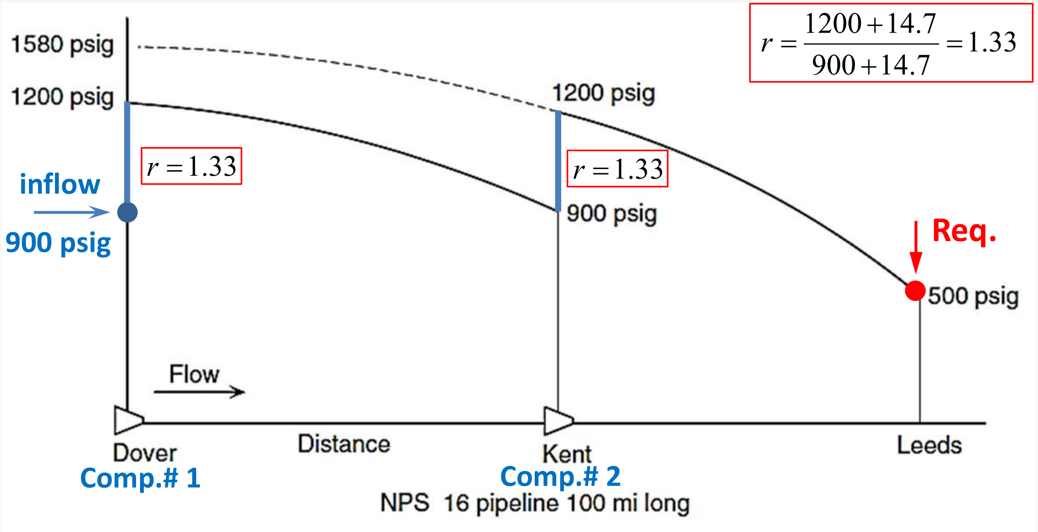


Fig. Gas pipeline with two compressor stations.

Problem

Example 1 (p-142)

A natural gas pipeline, 140 miles long from Dover to Leeds, is constructed of NPS 16, 0.250 in. wall thickness pipe, with an MOP of 1200 psig. The gas specific gravity and viscosity are 0.6 and 8×10^{-6} lb/ft-s, respectively. The pipe roughness can be assumed to be 700 μin., and the base pressure and base temperature are 14.7 psia and 60°F, respectively. The gas flow rate is 175 MMSCFD at 80°F, and the delivery pressure required at Leeds is 800 psig. Determine the number and locations of compressor stations required, neglecting elevation difference along the pipeline. Assume $Z = 0.85$.

Solution:

We need to use the Colebrook-White equation in conjunction the general flow equation to calculate the pressure drop in the gas pipeline from Dover to Leeds which is 140 mi long.

First, calculate the Reynolds number, Re

$$Re = 0.0004778\left(\frac{P_b}{T_b}\right)\left(\frac{GQ}{\mu D}\right) \quad \text{(USCS units)} \qquad (2.34)$$

$$\Rightarrow \text{Re} = 0.0004778\left(\frac{14.7}{60+460}\right)\left(\frac{0.6\times175\times10^{6}}{8\times10^{-6}\times15.5}\right) = 11{,}437{,}412$$

Relative roughness: $e = \dfrac{700\times10^{-6}}{15.5} = 4.5161\times10^{-5}$

Colebrook-White equation:

$$\frac{1}{\sqrt{f}} = -2\,\text{Log}_{10}\left(\frac{e}{3.7D} + \frac{2.51}{Re\sqrt{f}}\right) \quad \text{for } Re > 4000 \qquad (2.39)$$

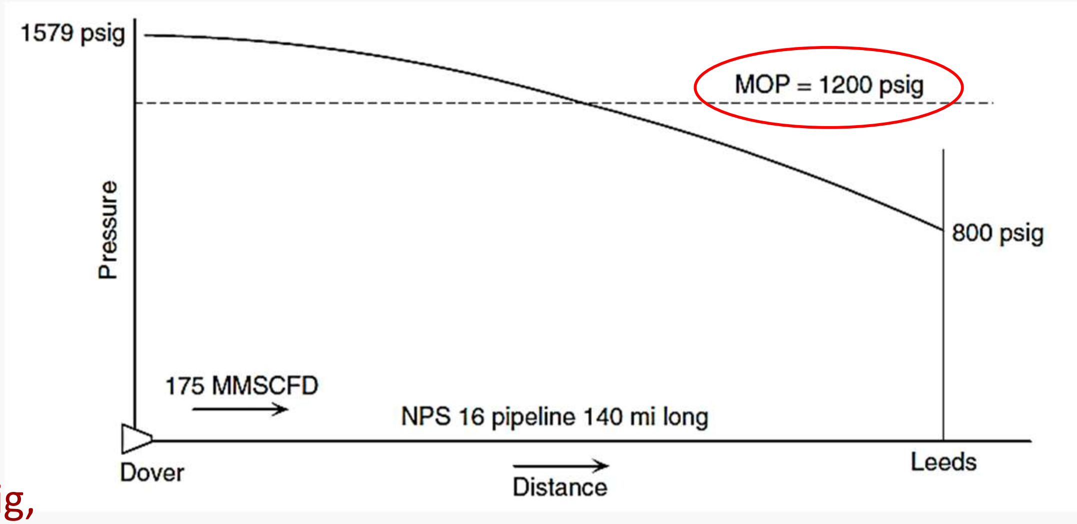
$$\Rightarrow \frac{1}{\sqrt{f}} = -2Log_{10}\left[\frac{4.516\times10^{-5}}{3.7} + \frac{2.51}{11{,}437{,}412\sqrt{f}}\right] \qquad \rightarrow f = 0.0107$$

Using general flow equation

$$Q = 77.54\left(\frac{T_b}{P_b}\right)\left(\frac{P_1^2 - P_2^2}{GT_f LZf}\right)^{0.5} D^{2.5} \quad \text{(USCS units)} \qquad (2.2)$$

$$\Rightarrow 175\times10^6 = 77.54\left(\frac{60+460}{14.7}\right)\left(\frac{P_1^2-(800+14.7)^2}{0.6\times(80+520)\times140\times0.85\times0.0107}\right)^{0.5} \times (15.5)^{2.5}$$

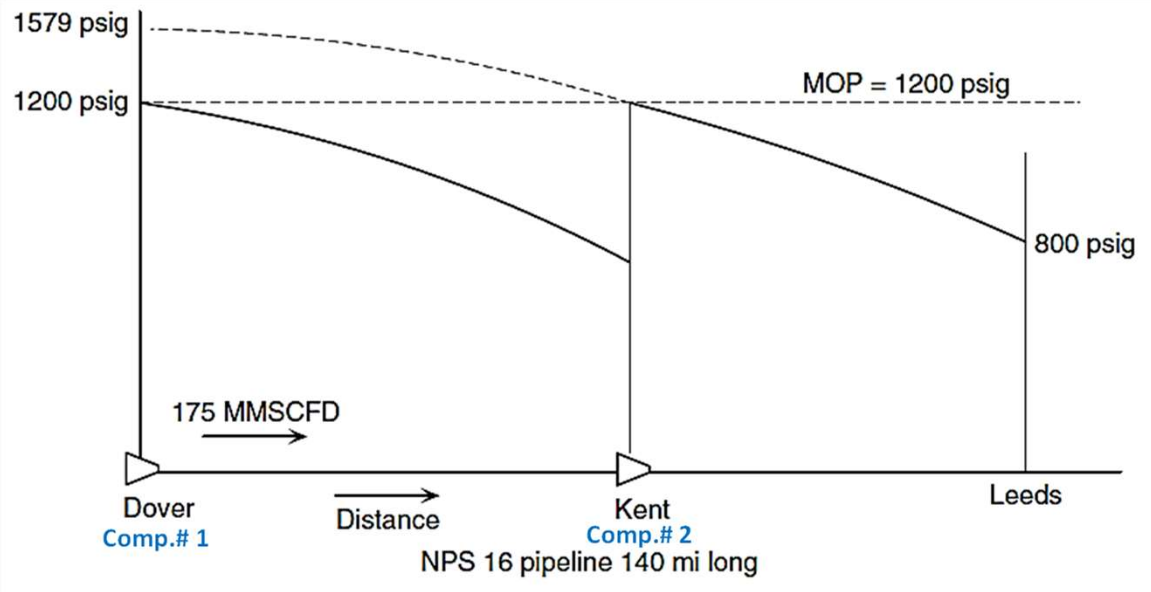
$$\Rightarrow P_1 = 1594 \text{ psia} = 1579.3 \text{ psig} \quad > \text{MOP}\,(1200\,\text{psig})$$



It can be seen from Figure that since the MOP is 1200 psig, we cannot discharge at 1579.3 psig at Dover.
This will not satisfy the requirment of gas pipeline MOP.
Thus, compressor (s) is (are) required to be installed in between Dover and Leeds.

We will need to reduce the discharge pressure at Dover to 1200 psig and install an additional compressor station at some point between Dover and Leeds, as shown in Figure below.

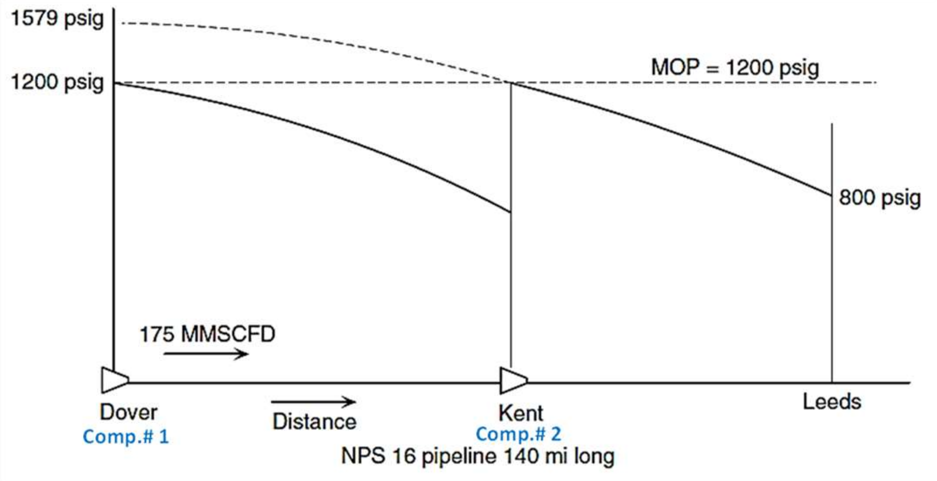
We will initially assume that the intermediate compressor station will be located at Kent, **halfway between Dover and Leeds (70 mi)**. For the pipe segment from Dover to Kent, we will calculate the suction pressure at the Kent compressor station as follows.



$$\Rightarrow 175\times10^{6} = 77.54\left(\frac{60+460}{14.7}\right)\left(\frac{(1200+14.7)^{2}-P_2^{2}}{0.6\times(80+520)\times70\times0.85\times0.0107}\right)^{0.5} \times (15.5)^{2.5}$$

$$\Rightarrow P_2 = 733 \text{ psia} = 718 \text{ psig} \quad \text{at Kent (suction pressure)}$$

At Kent, if we boost the gas pressure from 718 psig to 1200 psig (MOP), the compression ratio at Kent is $\frac{1214.7}{733} = 1.66$. This is a reasonable compression ratio for a centrifugal compressor. Next, we will see if the 1200 psig pressure at Kent will give the desired 800 psig delivery pressure at Leeds.



Considering the 70 mi segment from Kent to Leeds, using the General Flow equation we get

$$\Rightarrow 175\times10^6 = 77.54\left(\frac{60+460}{14.7}\right)\left(\frac{(1200+14.7)^2 - P_2^2}{0.6\times(80+520)\times70\times0.85\times0.0107}\right)^{0.5} \times (15.5)^{2.5}$$

$$\Rightarrow P_2 = 733 \text{ psia} = 718 \text{ psig} \quad \text{at Leeds (delivery pressure)}$$

This is less than the 800 psig desired. Hence, we must move the location of the Kent compressor station slightly toward Leeds so that the 800 psig delivery pressure can be achieved.

We will calculate the distance *L* required between Kent and Leeds. To achieve this, using General Flow Equation 2.2

$$175\times10^{6}=77.54\left(\frac{60+460}{14.7}\right)\left(\frac{(1200+14.7)^{2}-(800+14.7)^{2}}{0.6\times(80+520)\times L_{K\to L}\times0.85\times0.0107}\right)^{0.5}\times(15.5)^{2.5}$$

$$\Rightarrow L_{K\to L}=60.57\text{ miles}$$

Therefore, Kent must be located approximately 61 miles from Leeds **(not halfway)**. We must now recalculate the suction pressure at the Kent compressor station based on the pipe length of (140−60.57) = 79.43 miles between Dover and Kent.

From this suction pressure, we must also check the compression ratio.

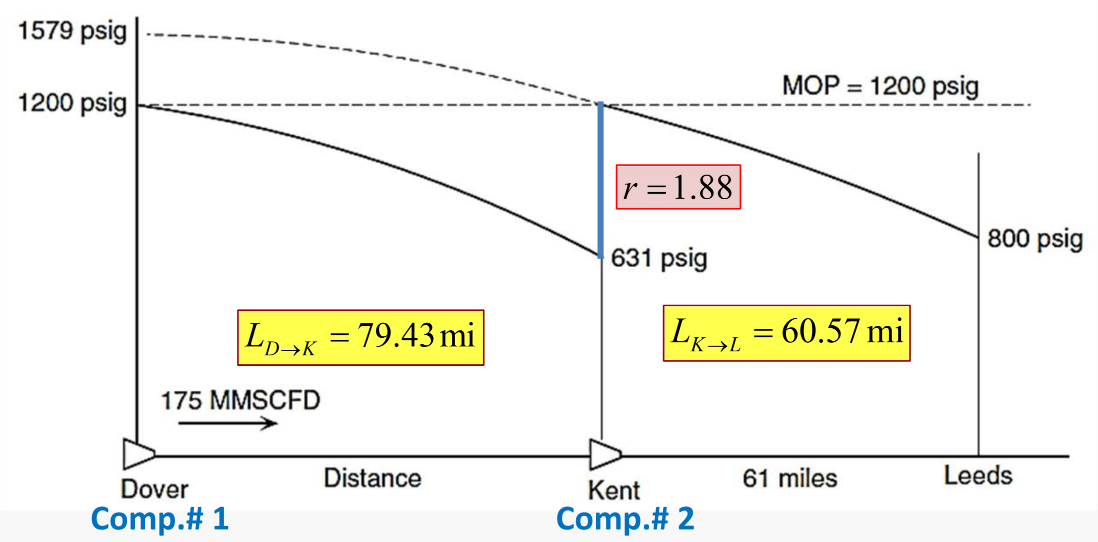
using General Flow Equation 2.2, for pipe segment between Dover and Kent,

$$175\times10^6 = 77.54\left(\frac{60+460}{14.7}\right)\left(\frac{(1200+14.7)^2 - P_2^2}{0.6\times(80+520)\times79.43\times0.85\times0.0107}\right)^{0.5}\times(15.5)^{2.5}$$

$$\Rightarrow P_2 = 645.49\,\text{psia} \quad = 630.79\,\text{psig} \quad \text{at Kent (suction pressure)}$$

Therefore, the suction pressure at Kent = 630.79 psig. The compression ratio at Kent = $\frac{1214.7}{645.49} = 1.88$.

The compression ratio is slightly more than the 1.5 we would like to see. However, for now, we will go ahead with this compression ratio.



Dover to Leeds pipeline with relocated Kent compressor station.

Horse Power Required

The amount of energy input to the gas by the compressors is dependent upon the pressure of the gas and flow rate. The **horsepower (*HP*),** which represents the energy per unit time, also depends upon the gas pressure and the flow rate. As the flow rate increases, the pressure also increases and, hence, the horsepower needed will also increase. Since energy is defined as work done by a force, we can state the power required in terms of the gas flow rate and the discharge pressure of the compressor station.

The ***head*** developed by the compressor is defined as the amount of energy supplied to the gas per unit mass of gas. Therefore, by multiplying the mass flow rate of gas by the compressor head, we can calculate the total energy supplied to the gas. Dividing this by compressor efficiency, we will get the horsepower required to compress the gas.

Horse Power Required

More commonly used formula for **compressor horsepower** that takes into account the compressibility of gas is as follows:

$$HP = 0.0857\left(\frac{\gamma}{\gamma-1}\right)QT_1\left(\frac{Z_1+Z_2}{2}\right)\left(\frac{1}{\eta_a}\right)\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}-1\right] \qquad (4.15)$$

where

HP = compressor horsepower

γ = ratio of specific heats of gas, dimensionless

Q = gas flow rate, MMSCFD

T_1 = suction temperature of gas, °R

P_1 = suction pressure of gas, psia

P_2 = discharge pressure of gas, psia

Z_1 = compressibility of gas at suction conditions, dimensionless

Z_2 = compressibility of gas at discharge conditions, dimensionless

η_a = compressor adiabatic (isentropic) efficiency, decimal value

Horse Power Required

The adiabatic efficiency η_a generally ranges from 0.75 to 0.85. By considering a mechanical efficiency η_m of the compressor driver, we can calculate the brake horsepower (*BHP*) required to run the compressor as follows:

$$BHP = \frac{HP}{\eta_m} \quad (4.17)$$

where *HP* is the horsepower calculated from the preceding equations, taking into account the adiabatic efficiency η_a of the compressor. The mechanical efficiency η_m of the driver can range from 0.95 to 0.98. The overall efficiency, η_o, is defined as the product of the adiabatic efficiency, η_a, and the mechanical efficiency, η_m:

$$\eta_o = \eta_a \times \eta_m \quad (4.18)$$

Horse Power Required

From the **adiabatic compression**, the discharge temperature of the gas is related to the suction temperature and the compression ratio by means of the following equation:

$$\left(\frac{T_2}{T_1}\right)=\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \qquad (4.19)$$

The adiabatic efficiency, η_a, can also be defined as the ratio of the adiabatic temperature rise to the actual temperature rise. Thus, if the gas temperature due to compression increases from T_1 to T_2, the actual temperature rise is $(T_2 - T_1)$.

The adiabatic efficiency can be calculated as

$$\eta_a=\left(\frac{T_1}{T_2-T_1}\right)\left[\left(\frac{Z_1}{Z_2}\right)\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}-1\right] \qquad (4.23)$$

* Derivation Homework

Problem

Example 9

Calculate the compressor horsepower required for an adiabatic compression of 106 MMSCFD gas with inlet temperature of 68°F and 725 psia pressure. The discharge pressure is 1305 psia. Assume the compressibility factors at suction and discharge conditions to be $Z_1 = 1.0$ and $Z_2 = 0.85$, respectively, and the adiabatic exponent $\gamma = 1.4$, with the adiabatic efficiency $\eta_a = 0.8$. If the mechanical efficiency of the compressor driver is 0.95, what *BHP* is required? Calculate the outlet temperature of the gas.

Homework

Ans: HP = 3550 HP

BHP = 3737 HP

T_2 = 326.46°F

Compressor Performance Curve

The performance curve of a centrifugal compressor that can be driven at varying speeds typically shows a graphic plot of the inlet flow rate in actual cubic feet per minute (ACFM) against the head or pressure generated at various percentages of the design speed. Figure 4.11 shows a typical centrifugal compressor performance curve or performance map.

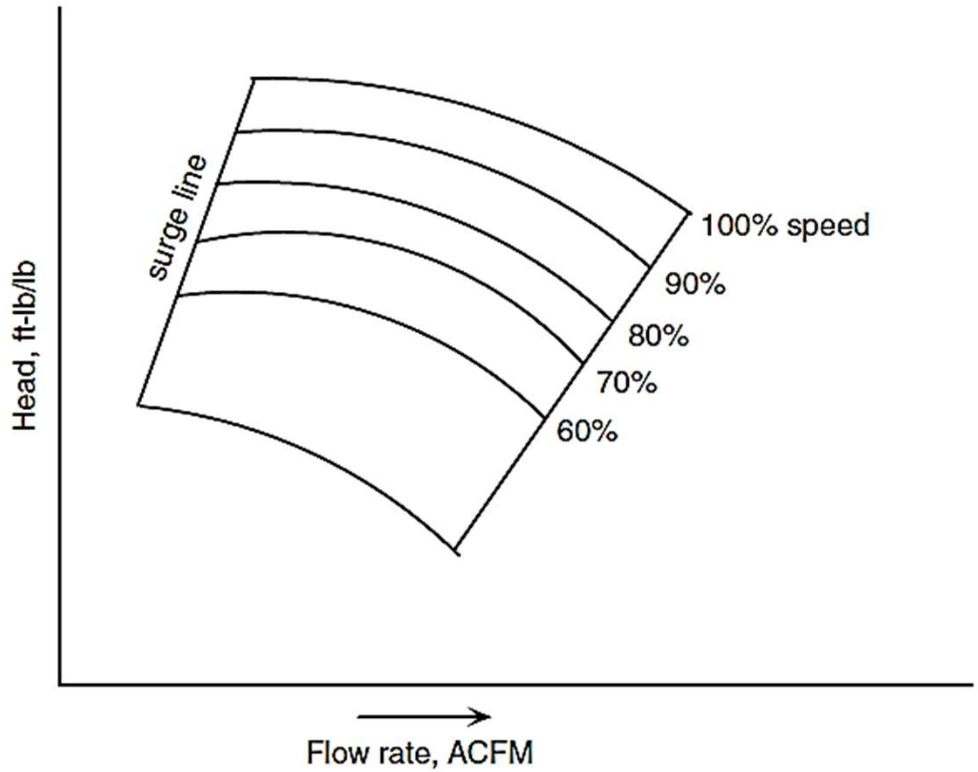


Figure 4.11 Typical centrifugal compressor performance curve.