# GAS PIPELINE HYDRAULICS 

## E. Shashi Menon

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## Preface

Gas Pipeline Hydraulics is a practical handbook for engineers, technicians, and others involved in the design and operation of pipelines transporting natural gas and other compressible fluids. It is based on the author's 30-year experience in the oil and gas industry. This book will help readers better understand and apply the principles of fluid mechanics to their daily work in the gas pipeline transmission and distribution industry. The book is divided into 10 chapters with several example problems solved fully, as well as additional problems provided as exercises.

Chapter 1 introduces the basic properties of natural gas and other compressible fluids that are important in understanding how gas behaves under various conditions of pressure and temperature as it flows through a pipeline. The properties of hydrocarbon gas mixtures, such as gravity, viscosity, and compressibility, are reviewed, and both analytical and graphical methods are explained with illustrative examples.

In Chapter 2, the methods of calculating the pressure drop in a gas pipeline are discussed. The General Flow equation is introduced as the basic equation, and the various correlations for friction factor and transmission factors, such as Colebrook and AGA, are explained. Other flow equations, such as Panhandle and Weymouth, are also covered using examples.

Chapter 3 extends the concepts of pressure drop calculations further to determine the total pressure required for transporting gas in pipelines under various configurations, such as series and parallel pipelines. The effects of intermediate delivery volumes and injection rates along a distribution pipeline are reviewed. The importance of the contract delivery pressures and the necessity of regulating pressures using a control valve or pressure regulator are also discussed. The effect of gas temperature on the pressure drop in a transmission pipeline is reviewed with example output reports from a commercial gas hydraulics simulation model. Equivalent lengths in series piping and equivalent diameters in parallel piping are covered, as well as pipe looping to increase gas pipeline flow rate. The quantity of gas contained in a section of a pipeline and the calculation of line pack are also reviewed.

Chapter 4 discusses compressor stations required to transport gas in a pipeline and how to calculate their numbers and optimum locations. Centrifugal and positive displacement compressors are explained, and their performances are compared. Typical performance characteristics of a centrifugal compressor are analyzed. Isothermal, adiabatic, and polytropic compression processes and horsepower required are discussed with sample calculations. The discharge temperature of the compressed gas and its impact on pipeline throughput, along with the necessity of gas cooling, are explained.

In Chapter 5, installing pipe loops to increase the throughput in a gas pipeline is explored. Looping is compared to the option of building intermediate compressor stations. The advantages and disadvantages of looping a pipeline versus installing compressor stations are discussed.

Chapter 6 covers the mechanical strength of a pipeline. The effects of pipe diameter, wall thickness, material of construction, and specific safety requirements
dictated by design codes and state and federal regulations are reviewed. Hydrostatic testing requirements and classification of pipelines based upon their proximity to human dwellings and industrial establishments and population density are also covered.

Chapter 7 introduces readers to thermal hydraulic analysis. For simplicity, long distance gas pipelines can be treated as isothermal flows. With compressor stations, the higher discharge temperature causes heat transfer between the pipeline gas and its surroundings. The effects of soil thermal conductivity, burial depth of pipeline, and the soil temperature are analyzed in determining the temperature and pressure profile of a gas pipeline. The Joule-Thompson cooling effect is reviewed, and the results from a commercial hydraulic simulation model are discussed.

Chapter 8 introduces transient pressure analysis. This is an area that is quite complex, and a complete discussion of the transient hydraulic analysis of gas pipelines requires a separate book. Readers are referred to some excellent references on this subject.

Chapter 9 covers valves and flow measurement. The different types of valves used in a gas pipeline and their characteristics are explained. The importance of accurate flow measurement in gas pipeline transactions is stressed. The codes and standards used to ensure proper design, construction, and operation of orifice flow meters are reviewed.

Chapter 10 explores economic aspects of gas pipeline systems. Determining the optimum pipe size for a particular gas flow rate, taking into account the initial capital cost and annual operating and maintenance cost, is explained. For a typical gas pipeline system, the various capital cost components are reviewed, along with the recurring annual costs such as operation and maintenance, fuel, and administrative costs. Also, the calculation methodology for determining transportation cost or tariff is covered.

At the end of each chapter, additional problems are provided as exercises. A list of references for the material covered in each chapter is included as well.

Appendices at the end of the book include tables of conversion factors from USCS units to SI units and vice versa, along with tables of properties of natural gas. Also included are tables showing commonly used pipe sizes, listing allowable internal pressures and hydrotest pressures. A section containing a summary of all hydraulic formulas used in the book is provided as a handy reference.

I enjoyed working with the fine staff at CRC Press. In particular, I want to thank Cindy Carelli, acquisitions editor; Theresa Del Forn, project coordinator; and Marsha Hecht, project editor, for their meticulous and prompt attention to all matters concerning the production of this book. They are indeed a very professional group and one of the best I have worked with over the years. I am indebted to my wife Pramila for the enormous amount of typing she did preparing the manuscript and for helping me proofread the final document and check for mathematical accuracy. I also appreciate all the good comments and suggestions that I received from practicing professional engineers such as Ken Zipp, Dan Bhavsar, and Charles Tateosian. In addition, I am thankful to Charles Peterson, Ken Zipp, and Ron Wood for agreeing to review
the manuscript, which resulted in some valuable comments that enhanced the quality of this book.

Finally, I would like to dedicate this book to my parents, who encouraged me in all my endeavors throughout my childhood and adult life.
E. Shashi Menon

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## CHAPTER 1

## Gas Properties

In this chapter we will discuss the properties of gases that influence gas flow through a pipeline. We will explore the relationship of pressure, volume, and temperature of a gas and how the gas properties such as density, viscosity, and compressibility change with the temperature and pressure. Starting with the ideal or perfect gases that obey the ideal gas equation, we will examine how real gases differ from ideal gases. The concept of compressibility factor, or gas deviation factor, will be introduced and methods of calculating the compressibility factor using some popular graphical correlation and calculation methods explained. The properties of a mixture of gases will be discussed, and how these are calculated will be covered. Understanding the gas properties is an important first step toward analysis of gas pipeline hydraulics.

A fluid can be a liquid or a gas. Liquids are generally considered almost incompressible. A gas is classified as a homogenous fluid with low density and viscosity. It expands to fill the vessel that contains the gas. The molecules that constitute the gas are spaced farther apart in comparison with a liquid and, therefore, a slight change in pressure affects the density of gas more than that of a liquid. Gases, therefore, have higher compressibility than liquids. This implies that gas properties such as density, viscosity, and compressibility factor change with pressure.

### 1.1 MASS AND WEIGHT

Mass is the quantity of matter in a substance. It is sometimes used interchangeably with weight. Strictly speaking, mass is a scalar quantity, whereas weight is a force and, therefore, a vector quantity. Mass is independent of the geographic location, whereas weight depends upon the acceleration due to gravity and, therefore, varies slightly with geographic location. Mass is measured in slugs in the U.S. Customary System (USCS) of units and kilograms (kg) in Systeme International (SI) units.

However, for most purposes, we say that a $10-\mathrm{lb}$ mass has a weight of 10 lb . The pound (lb) is a more convenient unit for mass, and to distinguish between mass and weight, the terms pound mass (lbm) and pound force (lbf) are sometimes used. A slug is equal to approximately 32.2 lb .

If some gas is contained in a certain volume and the temperature and pressure change, the mass will remain constant unless some gas is taken out or added to the container. This is known as the principle of conservation of mass. Weight is measured in pounds (lb) in USCS units and in Newton (N) in SI units. Sometimes we talk about mass flow rate through a pipeline rather than volume flow rate. Mass flow rate is measured in $\mathrm{lb} / \mathrm{hr}$ in USCS units or $\mathrm{kg} / \mathrm{hr}$ in SI units.

### 1.2 VOLUME

Volume of a gas is the space a given mass of gas occupies at a particular temperature and pressure. Since gas is compressible, it will expand to fill the available space. Therefore, the gas volume will vary with temperature and pressure. Hence, a certain volume of a given mass of gas at some temperature and pressure will decrease in volume as the pressure is increased and vice versa. Suppose a quantity of gas is contained in a volume of $100 \mathrm{ft}^{3}$ at a temperature of $80^{\circ} \mathrm{F}$ and a pressure of 200 psi . If the temperature is increased to $100^{\circ} \mathrm{F}$, keeping the volume constant, the pressure will also increase. Similarly, if the temperature is reduced, gas pressure will also reduce, provided volume remains constant. Charles's law states that for constant volume, the pressure of a fixed mass of gas will vary directly with the temperature. Thus, if temperature increases by $20 \%$, the pressure will also rise by $20 \%$. Similarly, if pressure is maintained constant, the volume will increase in direct proportion with temperature. Charles's law, Boyle's law, and other gas laws will be discussed in detail later in this chapter.

Volume of gas is measured in $\mathrm{ft}^{3}$ in USCS units and $\mathrm{m}^{3}$ in SI units. Other units for volume include thousand $\mathrm{ft}^{3}\left(\mathrm{Mft}^{3}\right)$ and million $\mathrm{ft}^{3}\left(\mathrm{MMft}^{3}\right)$ in USCS units and thousand $\mathrm{m}^{3}\left(\mathrm{~km}^{3}\right)$ and million $\mathrm{m}^{3}\left(\mathrm{Mm}^{3}\right)$ in SI units. When referred to standard conditions (also called base conditions) of temperature and pressure $\left(60^{\circ} \mathrm{F}\right.$ and 14.7 psia in USCS units), the volume is stated as standard volume and, therefore, measured in standard $\mathrm{ft}^{3}$ (SCF) or million standard $\mathrm{ft}^{3}$ (MMSCF). It must be noted that in the USCS units, the practice has been to use $M$ to represent a thousand, and therefore MM refers to a million. This goes back to the Roman days of numerals, when M represented a thousand. In SI units, a more logical step is followed. For thousand, the letter k (for kilo) is used and the letter M (for Mega) is used for a million. Therefore, 500 MSCFD in USCS units refers to 500 thousand standard cubic feet per day ( $500,000 \mathrm{ft}^{3} / \mathrm{day}$ ), whereas $15 \mathrm{Mm}^{3} /$ day means 15 million cubic meters per day in SI units. This distinction in the use of the letter M to denote a thousand in USCS units and M for a million in SI units must be clearly understood.

Volume flow rate of gas is measured per unit time and can be expressed as $\mathrm{ft}^{3} / \mathrm{min}, \mathrm{ft}^{3} / \mathrm{h}, \mathrm{ft}^{3} / \mathrm{day}$, SCFD, MMSCFD, etc. in USCS units. In SI units, gas flow rate is expressed in $\mathrm{m}^{3} / \mathrm{h}$ or $\mathrm{Mm}^{3} /$ day.

### 1.3 DENSITY, SPECIFIC WEIGHT, AND SPECIFIC VOLUME

Density represents the amount of gas that can be packed in a given volume. Therefore, it is measured in terms of mass per unit volume. If 5 lb of a gas is contained in $100 \mathrm{ft}^{3}$ of volume at some temperature and pressure, we say that the gas density is $5 / 100=0.05 \mathrm{lb} / \mathrm{ft}^{3}$.

Strictly speaking, in USCS units density must be expressed as slug/ft ${ }^{3}$ since mass is customarily referred to in slug.

Thus,

$$
\begin{equation*}
\rho=\frac{m}{V} \tag{1.1}
\end{equation*}
$$

where
$\rho=$ density of gas
$m=$ mass of gas
$V=$ volume of gas

Density is expressed in slug/ft ${ }^{3}$ or $\mathrm{lb} / \mathrm{ft}^{3}$ in USCS units and $\mathrm{kg} / \mathrm{m}^{3}$ in SI units.
A companion term called specific weight is also used when referring to the density of gas. Specific weight, represented by the symbol $\gamma$, is the weight of gas per unit volume measured in lb/ft ${ }^{3}$ in USCS units, and is, therefore, contrasted with density, which is measured in slug/ft ${ }^{3}$. In SI units, the specific weight is expressed in Newton per $\mathrm{m}^{3}\left(\mathrm{~N} / \mathrm{m}^{3}\right)$. Quite often, density is also referred to in $\mathrm{lb} / \mathrm{ft}^{3}$ in USCS units.

The reciprocal of the specific weight is known as the specific volume. By definition, therefore, specific volume represents the volume occupied by a unit weight of gas. It is measured in $\mathrm{ft}^{3} / \mathrm{lb}$ in USCS units and $\mathrm{m}^{3} / \mathrm{N}$ in SI units. If the specific weight of a particular gas is $0.06 \mathrm{lb} / \mathrm{ft}^{3}$ at some temperature and pressure, its specific volume is $\frac{1}{0.06}$ or $16.67 \mathrm{ft}^{3} / \mathrm{lb}$.

### 1.4 SPECIFIC GRAVITY

Specific gravity of a gas, sometimes called gravity, is a measure of how heavy the gas is compared to air at a particular temperature. It might also be called relative density, expressed as the ratio of the gas density to the density of air. Because specific gravity is a ratio, it is a dimensionless quantity.

$$
\begin{equation*}
G=\frac{\rho_{g}}{\rho_{\text {air }}} \tag{1.2}
\end{equation*}
$$

where
$G=$ gas gravity, dimensionless
$\rho_{g}=$ density of gas
$\rho_{\text {air }}=$ density of air

Both densities in Equation 1.2 must be in the same units and measured at the same temperature.

For example, natural gas has a specific gravity of 0.60 (air $=1.00$ ) at $60^{\circ} \mathrm{F}$. This means that the gas is $60 \%$ as heavy as air.

If we know the molecular weight of a particular gas, we can calculate its gravity by dividing the molecular weight by the molecular weight of air, as follows.

$$
\begin{equation*}
G=\frac{M_{g}}{M_{\text {air }}}=\frac{M_{g}}{28.9625} \tag{1.3}
\end{equation*}
$$

or

$$
\begin{equation*}
G=\frac{M_{g}}{29} \tag{1.4}
\end{equation*}
$$

Rounding off the molecular weight of air to 29
where
$G \quad=$ specific gravity of gas
$M_{g}=$ molecular weight of gas
$M_{\text {air }}=$ molecular weight of air $=28.9625$

Since natural gas consists of a mixture of several gases (methane, ethane, etc.), the molecular weight $M_{g}$ in Equation 1.4 is referred to as the apparent molecular weight of the gas mixture.

When the molecular weight and the percentage or mole fractions of the individual components of a natural gas mixture are known, we can calculate the molecular weight of the gas mixture by using a weighted average method. Thus, a natural gas mixture consisting of $90 \%$ methane, $8 \%$ ethane, and $2 \%$ propane will have a specific gravity of

$$
G=\frac{(0.9 \times M 1)+(0.08 \times M 2)+(0.02 \times M 3)}{29}
$$

where $M 1, M 2$, and $M 3$ are the molecular weights of methane, ethane, and propane, respectively, and 29 represents the molecular weight of air.

Table 1.1 lists the molecular weights and other properties of several hydrocarbon gases.

### 1.5 VISCOSITY

The viscosity of a fluid represents its resistance to flow. The higher the viscosity, the more difficult it is to flow. Lower-viscosity fluids flow easily in pipes and cause less pressure drop. Liquids have much larger values of viscosity compared to gases. For example, water has a viscosity of 1.0 centiPoise ( cP ), whereas viscosity of natural gas is approximately 0.0008 cP . Even though the gas viscosity is a small number, it
has an important function in determining the type of flow in pipelines. The Reynolds number (discussed in Chapter 2) is a dimensionless parameter that is used to classify flow rate in pipelines. It depends on the gas viscosity, flow rate, pipe diameter, temperature, and pressure. The absolute viscosity, also called the dynamic viscosity, is expressed in lb/ft-s in USCS units and Poise (P) in SI units. A related term is the kinematic viscosity. This is simply the absolute viscosity divided by the density. The two viscosities are related as follows:

$$
\begin{equation*}
v=\frac{\mu}{\rho} \tag{1.5}
\end{equation*}
$$

where, in USCS units,
$v=$ kinematic viscosity, $\mathrm{ft}^{2} / \mathrm{s}$
$\mu=$ dynamic viscosity, lb/ft-s
$\rho=$ density, $\mathrm{lb} / \mathrm{ft}^{3}$
and in SI units
$v=$ kinematic viscosity, St
$\mu=$ dynamic viscosity, P
$\rho=$ density, $\mathrm{kg} / \mathrm{m}^{3}$
Kinematic viscosity is expressed in $\mathrm{ft}^{2} / \mathrm{s}$ in USCS units and Stokes (St) in SI units. Other units of viscosity in SI units include centipoise (cP) for dynamic viscosity and centistokes (cSt) for kinematic viscosity. Appendix A includes conversion factors for converting viscosity from one set of units to another.

The viscosity of a gas depends on its temperature and pressure. Unlike liquids, the viscosity of a gas increases with increase in temperature. Since viscosity represents resistance to flow, as the gas temperature increases, the quantity of gas flow through a pipeline will decrease; hence, more throughput is possible in a gas pipeline at lower temperatures. This is in sharp contrast to liquid flow, where the throughput increases with temperature due to decrease in viscosity and vice versa. It must be noted that, unlike liquids, pressure also affects the viscosity of a gas. Like temperature, the gas viscosity increases with pressure. Figure 1.1 shows the variation of viscosity with temperature for a gas. Table 1.2 lists the viscosities of common gases.

Since natural gas is a mixture of pure gases such as methane and ethane, the following formula is used to calculate the viscosity from the viscosities of component gases:

$$
\begin{equation*}
\mu=\frac{\Sigma\left(\mu_{i} y_{i} \sqrt{M_{i}}\right)}{\Sigma\left(y_{i} \sqrt{M_{i}}\right)} \tag{1.6}
\end{equation*}
$$

where
$\mu=$ dynamic viscosity of gas mixture
$\mu_{i}=$ dynamic viscosity of gas component $i$
$y_{i}=$ mole fraction or percent of gas component $i$
$M_{i}=$ molecular weight of gas component $i$

| Gas | Formula | Molecular Weight | Vapor Pressure psia at $100^{\circ} \mathrm{F}$ | Critical Constants |  |  | Ideal Gas$14.696 \text { psia, } 60^{\circ} \mathrm{F}$ |  | Specific Heat, Btu/lb/ ${ }^{\circ}$ F 14.696 psia, $60^{\circ} \mathrm{F}$ Ideal Gas |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
|  |  |  |  | Pressure psia | Temperature ${ }^{\circ} \mathrm{F}$ | Volume ft ${ }^{3} / \mathrm{lb}$ | $\begin{gathered} \text { Spgr } \\ \text { (air=1.00) } \end{gathered}$ | ft ${ }^{3} / \mathrm{lb}$-gas |  |
| Methane | $\mathrm{CH}_{4}$ | 16.0430 | 5000 | 666.0 | -116.66 | 0.0988 | 0.5539 | 23.654 | 0.52676 |
| Ethane | $\mathrm{C}_{2} \mathrm{H}_{6}$ | 30.0700 | 800 | 707.0 | 90.07 | 0.0783 | 1.0382 | 12.620 | 0.40789 |
| Propane | $\mathrm{C}_{3} \mathrm{H}_{8}$ | 44.0970 | 188.65 | 617.0 | 205.93 | 0.0727 | 1.5226 | 8.6059 | 0.38847 |
| Isobutane | $\mathrm{C}_{4} \mathrm{H}_{10}$ | 58.1230 | 72.581 | 527.9 | 274.4 | 0.0714 | 2.0068 | 6.5291 | 0.38669 |
| n-butane | $\mathrm{C}_{4} \mathrm{H}_{10}$ | 58.1230 | 51.706 | 548.8 | 305.52 | 0.0703 | 2.0068 | 6.5291 | 0.39500 |
| Iso-pentane | $\mathrm{C}_{5} \mathrm{H}_{12}$ | 72.1500 | 20.443 | 490.4 | 368.96 | 0.0684 | 2.4912 | 5.2596 | 0.38448 |
| n-pentane | $\mathrm{C}_{5} \mathrm{H}_{12}$ | 72.1500 | 15.575 | 488.1 | 385.7 | 0.0695 | 2.4912 | 5.2596 | 0.38831 |
| Neo-pentane | $\mathrm{C}_{5} \mathrm{H}_{12}$ | 72.1500 | 36.72 | 464.0 | 321.01 | 0.0673 | 2.4912 | 5.2596 | 0.39038 |
| n -hexane | $\mathrm{C}_{6} \mathrm{H}_{14}$ | 86.1770 | 4.9596 | 436.9 | 453.8 | 0.0688 | 2.9755 | 4.4035 | 0.38631 |
| 2-methyl pentane | $\mathrm{C}_{6} \mathrm{H}_{14}$ | 86.1770 | 6.769 | 436.6 | 435.76 | 0.0682 | 2.9755 | 4.4035 | 0.38526 |
| 3-methyl pentane | $\mathrm{C}_{6} \mathrm{H}_{14}$ | 86.1770 | 6.103 | 452.5 | 448.2 | 0.0682 | 2.9755 | 4.4035 | 0.37902 |
| Neo hexane | $\mathrm{C}_{6} \mathrm{H}_{14}$ | 86.1770 | 9.859 | 446.7 | 419.92 | 0.0667 | 2.9755 | 4.4035 | 0.38231 |
| 2,3-dimethylbutane | $\mathrm{C}_{6} \mathrm{H}_{14}$ | 86.1770 | 7.406 | 454.0 | 440.08 | 0.0665 | 2.9755 | 4.4035 | 0.37762 |
| n -Heptane | $\mathrm{C}_{7} \mathrm{H}_{16}$ | 100.2040 | 1.621 | 396.8 | 512.8 | 0.0682 | 3.4598 | 3.7872 | 0.38449 |
| 2-Methylhexane | $\mathrm{C}_{7} \mathrm{H}_{16}$ | 100.2040 | 2.273 | 396.0 | 494.44 | 0.0673 | 3.4598 | 3.7872 | 0.38170 |
| 3-Methylhexane | $\mathrm{C}_{7} \mathrm{H}_{16}$ | 100.2040 | 2.13 | 407.6 | 503.62 | 0.0646 | 3.4598 | 3.7872 | 0.37882 |
| 3-Ethylpentane | $\mathrm{C}_{7} \mathrm{H}_{16}$ | 100.2040 | 2.012 | 419.2 | 513.16 | 0.0665 | 3.4598 | 3.7872 | 0.38646 |
| 2,2-Dimethylpentane | $\mathrm{C}_{7} \mathrm{H}_{16}$ | 100.2040 | 3.494 | 401.8 | 476.98 | 0.0665 | 3.4598 | 3.7872 | 0.38651 |
| 2,4-Dimethylpentane | $\mathrm{C}_{7} \mathrm{H}_{16}$ | 100.2040 | 3.294 | 397.4 | 475.72 | 0.0667 | 3.4598 | 3.7872 | 0.39627 |
| 3,3-Dimethylpentane | $\mathrm{C}_{7} \mathrm{H}_{16}$ | 100.2040 | 2.775 | 427.9 | 505.6 | 0.0662 | 3.4598 | 3.7872 | 0.38306 |
| Triptane | $\mathrm{C}_{7} \mathrm{H}_{16}$ | 100.2040 | 3.376 | 427.9 | 496.24 | 0.0636 | 3.4598 | 3.7872 | 0.37724 |
| n-octane | $\mathrm{C}_{8} \mathrm{H}_{18}$ | 114.2310 | 0.5371 | 360.7 | 564.15 | 0.0673 | 3.9441 | 3.322 | 0.38334 |
| Di Isobutyl | $\mathrm{C}_{8} \mathrm{H}_{18}$ | 114.2310 | 1.1020 | 361.1 | 530.26 | 0.0676 | 3.9441 | 3.322 | 0.37571 |
| Isooctane | $\mathrm{C}_{8} \mathrm{H}_{18}$ | 114.2310 | 1.7090 | 372.7 | 519.28 | 0.0657 | 3.9441 | 3.322 | 0.38222 |
| n -Nonane | $\mathrm{C}_{9} \mathrm{H}_{20}$ | 128.2580 | 0.17155 | 330.7 | 610.72 | 0.0693 | 4.4284 | 2.9588 | 0.38248 |
| n -Decane | $\mathrm{C}_{10} \mathrm{H}_{22}$ | 142.2850 | 0.06088 | 304.6 | 652.1 | 0.0702 | 4.9127 | 2.6671 | 0.38181 |
| Cyclopentane | $\mathrm{C}_{5} \mathrm{H}_{10}$ | 70.1340 | 9.917 | 653.8 | 461.1 | 0.0594 | 2.4215 | 5.411 | 0.27122 |
| Methylcyclopentane | $\mathrm{C}_{6} \mathrm{H}_{12}$ | 84.1610 | 4.491 | 548.8 | 499.28 | 0.0607 | 2.9059 | 4.509 | 0.30027 |
| Cyclohexane | $\mathrm{C}_{6} \mathrm{H}_{12}$ | 84.1610 | 3.267 | 590.7 | 536.6 | 0.0586 | 2.9059 | 4.509 | 0.29012 |


| Methylcyclohexane | $\mathrm{C}_{7} \mathrm{H}_{14}$ | 98.1880 | 1.609 | 503.4 | 570.2 | 0.0600 | 3.3902 | 3.8649 | 0.31902 |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Ethylene | $\mathrm{C}_{2} \mathrm{H}_{4}$ | 28.0540 | 1400 | 731.0 | 48.54 | 0.0746 | 0.9686 | 13.527 | 0.35789 | 8 |
| Propylene | $\mathrm{C}_{3} \mathrm{H}_{6}$ | 42.0810 | 232.8 | 676.6 | 198.31 | 0.0717 | 1.4529 | 9.0179 | 0.35683 | 0 |
| Butylene | $\mathrm{C}_{4} \mathrm{H}_{8}$ | 56.1080 | 62.55 | 586.4 | 296.18 | 0.0683 | 1.9373 | 6.7636 | 0.35535 | 0 |
| Cis-2-butene | $\mathrm{C}_{4} \mathrm{H}_{8}$ | 56.1080 | 45.97 | 615.4 | 324.31 | 0.0667 | 1.9373 | 6.7636 | 0.33275 | $\bigcirc$ |
| Trans-2-butene | $\mathrm{C}_{4} \mathrm{H}_{8}$ | 56.1080 | 49.88 | 574.9 | 311.8 | 0.0679 | 1.9373 | 6.7636 | 0.35574 | \% |
| Isobutene | $\mathrm{C}_{4} \mathrm{H}_{8}$ | 56.1080 | 64.95 | 580.2 | 292.49 | 0.0681 | 1.9373 | 6.7636 | 0.36636 | $\stackrel{1}{7}$ |
| 1-Pentene | $\mathrm{C}_{5} \mathrm{H}_{10}$ | 70.1340 | 19.12 | 509.5 | 376.86 | 0.0674 | 2.4215 | 5.411 | 0.35944 | © |
| 1,2-Butadene | $\mathrm{C}_{4} \mathrm{H}_{6}$ | 54.0920 | 36.53 | 656.0 | 354 | 0.0700 | 1.8677 | 7.0156 | 0.34347 |  |
| 1,3-Butadene | $\mathrm{C}_{4} \mathrm{H}_{6}$ | 54.0920 | 59.46 | 620.3 | 306 | 0.0653 | 1.8677 | 7.0156 | 0.34223 |  |
| Isoprene | $\mathrm{C}_{5} \mathrm{H}_{8}$ | 68.1190 | 16.68 | 582.0 | 403 | 0.0660 | 2.3520 | 5.571 | 0.35072 |  |
| Acetylene | $\mathrm{C}_{2} \mathrm{H}_{2}$ | 26.0380 |  | 890.4 | 95.29 | 0.0693 | 0.8990 | 14.574 | 0.39754 |  |
| Benzene | $\mathrm{C}_{6} \mathrm{H}_{6}$ | 78.1140 | 3.225 | 710.4 | 552.15 | 0.0531 | 2.6971 | 4.8581 | 0.24295 |  |
| Toluene | $\mathrm{C}_{7} \mathrm{H}_{8}$ | 92.1410 | 1.033 | 595.5 | 605.5 | 0.0549 | 3.1814 | 4.1184 | 0.26005 |  |
| Ethyl-benzene | $\mathrm{C}_{8} \mathrm{H}_{10}$ | 106.1670 | 0.3716 | 523 | 651.22 | 0.0564 | 3.6657 | 3.5744 | 0.27768 |  |
| o-Xylene | $\mathrm{C}_{8} \mathrm{H}_{10}$ | 106.1670 | 0.2643 | 541.6 | 674.85 | 0.0557 | 3.6657 | 3.5744 | 0.28964 |  |
| m-Xylene | $\mathrm{C}_{8} \mathrm{H}_{10}$ | 106.1670 | 0.3265 | 512.9 | 650.95 | 0.0567 | 3.6657 | 3.5744 | 0.27427 |  |
| p-Xylene | $\mathrm{C}_{8} \mathrm{H}_{10}$ | 106.1670 | 0.3424 | 509.2 | 649.47 | 0.0572 | 3.6657 | 3.5744 | 0.27470 |  |
| Styrene | $\mathrm{C}_{8} \mathrm{H}_{8}$ | 104.1520 | 0.2582 | 587.8 | 703 | 0.0534 | 3.5961 | 3.6435 | 0.26682 |  |
| Isopropylbenzene | $\mathrm{C}_{9} \mathrm{H}_{12}$ | 120.1940 | 0.188 | 465.4 | 676.2 | 0.0569 | 4.1500 | 3.1573 | 0.30704 |  |
| Methyl alcohol | $\mathrm{CH}_{4} \mathrm{O}$ | 32.0420 | 4.631 | 1174 | 463.01 | 0.0590 | 1.1063 | 11.843 | 0.32429 |  |
| Ethyl alcohol | $\mathrm{C}_{2} \mathrm{H}_{6} \mathrm{O}$ | 46.0690 | 2.313 | 891.7 | 465.31 | 0.0581 | 1.5906 | 8.2372 | 0.33074 |  |
| Carbon monoxide | CO | 28.0100 |  | 506.8 | -220.51 | 0.0527 | 0.9671 | 13.548 | 0.24847 |  |
| Carbon dioxide | $\mathrm{CO}_{2}$ | 44.0100 |  | 1071 | 87.73 | 0.0342 | 1.5196 | 8.6229 | 0.19909 |  |
| Hydrogen sulfide | $\mathrm{H}_{2} \mathrm{~S}$ | 34.0820 | 394.59 | 1306 | 212.4 | 0.0461 | 1.1768 | 11.134 | 0.23838 |  |
| Sulfur dioxide | $\mathrm{SO}_{2}$ | 64.0650 | 85.46 | 1143 | 315.7 | 0.0305 | 2.2120 | 5.9235 | 0.14802 |  |
| Ammonia | $\mathrm{NH}_{3}$ | 17.0305 | 211.9 | 1647 | 270.2 | 0.0681 | 0.5880 | 22.283 | 0.49678 |  |
| Air | $\mathrm{N}_{2}+\mathrm{O}_{2}$ | 28.9625 |  | 546.9 | -221.29 | 0.0517 | 1.0000 | 13.103 | 0.2398 |  |
| Hydrogen | $\mathrm{H}_{2}$ | 2.0159 |  | 187.5 | -400.3 | 0.5101 | 0.06960 | 188.25 | 3.4066 |  |
| Oxygen | $\mathrm{O}_{2}$ | 31.9988 |  | 731.4 | -181.4 | 0.0367 | 1.1048 | 11.859 | 0.21897 |  |
| Nitrogen | $\mathrm{N}_{2}$ | 28.0134 |  | 493 | -232.48 | 0.0510 | 0.9672 | 13.546 | 0.24833 |  |
| Chlorine | $\mathrm{Cl}_{2}$ | 70.9054 | 157.3 | 1157 | 290.69 | 0.0280 | 2.4482 | 5.3519 | 0.11375 |  |
| Water | $\mathrm{H}_{2} \mathrm{O}$ | 18.0153 | 0.95 | 3200.1 | 705.1 | 0.04975 | 0.62202 | 21.065 | 0.44469 |  |
| Helium | He | 4.0026 |  | 32.99 | -450.31 | 0.2300 | 0.1382 | 94.814 | 1.24040 |  |
| Hydrogen chloride | HCl | 36.4606 | 906.71 | 1205 | 124.75 | 0.0356 | 1.2589 | 10.408 | 0.19086 | $\checkmark$ |



Figure 1.1 Variation of gas viscosity with temperature.

Therefore, a homogeneous mixture that consists of $20 \%$ of a gas A (molecular weight $=18$ ) that has a viscosity $6 \times 10^{-6}$ Poise and $80 \%$ of a gas B (molecular weight $=17$ ) that has a viscosity $8 \times 10^{-6}$ Poise will have a resultant viscosity of

$$
\mu=\frac{(0.2 \times 6 \times \sqrt{18})+(0.8 \times 8 \times \sqrt{17})}{(0.2 \times \sqrt{18})+(0.8 \times \sqrt{17})} \times 10^{-6}=7.59 \times 10^{-6} \text { Poise }
$$

Table 1.2 Viscosities of Common Gases

| Gas | Viscosity (cP) |
| :--- | :---: |
| Methane | 0.0107 |
| Ethane | 0.0089 |
| Propane | 0.0075 |
| i-Butane | 0.0071 |
| n-Butane | 0.0073 |
| i-Pentane | 0.0066 |
| n-Pentane | 0.0066 |
| Hexane | 0.0063 |
| Heptane | 0.0059 |
| Octane | 0.0050 |
| Nonane | 0.0048 |
| Decane | 0.0045 |
| Ethylene | 0.0098 |
| Carbon Monoxide | 0.0184 |
| Carbon Dioxide | 0.0147 |
| Hydrogen Sulphide | 0.0122 |
| Air | 0.0178 |
| Nitrogen | 0.0173 |
| Helium | 0.0193 |

It must be noted that all viscosities must be measured at the same temperature and pressure.

The reader is referred to W. McCain's book for further discussion on calculation of viscosities of natural gas mixtures. See the Reference section for more details.

## Example 1

A natural gas mixture consists of four components $\mathrm{C}_{1}, \mathrm{C}_{2}, \mathrm{C}_{3}$, and $\mathrm{nC}_{4}$. Their mole fractions and viscosities at a particular temperature and pressure are indicated below, along with their molecular weights.

| Component | Mole Fraction $\boldsymbol{y}$ | Viscosity, cP | Molecular Weight |
| :---: | :---: | :---: | :---: |
| $\mathrm{C}_{1}$ | 0.8200 | 0.0130 | 16.04 |
| $\mathrm{C}_{2}$ | 0.1000 | 0.0112 | 30.07 |
| $\mathrm{C}_{3}$ | 0.0500 | 0.0098 | 44.10 |
| $\mathrm{nC}_{4}$ | 0.0300 | 0.0091 | 58.12 |
| Total | 1.000 |  |  |

Calculate the viscosity of the gas mixture.
Solution

Using the given data, we prepare a table as follows. $M$ represents the molecular weight of each component and $\mu$ the viscosity.

| Component | $\boldsymbol{y}$ | $\boldsymbol{M}$ | $\boldsymbol{M}^{1 / 2}$ | $\boldsymbol{y} \boldsymbol{M}^{1 / 2}$ | $\boldsymbol{\mu}$ | $\boldsymbol{\mu} \boldsymbol{y} \boldsymbol{M}^{1 / 2}$ |
| :---: | :--- | :---: | :---: | :---: | :---: | :---: |
| $\mathrm{C}_{1}$ | 0.8200 | 16.04 | 4.00 | 3.2841 | 0.0130 | 0.0427 |
| $\mathrm{C}_{2}$ | 0.1000 | 30.07 | 5.48 | 0.5484 | 0.0112 | 0.0061 |
| $\mathrm{C}_{3}$ | 0.0500 | 44.10 | 6.64 | 0.3320 | 0.0098 | 0.0033 |
| $\mathrm{nC}_{4}$ | 0.0300 | 58.12 | 7.62 | 0.2287 | 0.0091 | 0.0021 |
| Total | 1.000 |  |  | 4.3932 |  | 0.0542 |

From Equation 1.6, the viscosity of the gas mixture is

$$
\mu=\frac{0.0542}{4.3932}=0.0123 \mathrm{cP}
$$

### 1.6 IDEAL GASES

An ideal gas is defined as a fluid in which the volume of the gas molecules is negligible when compared to the volume occupied by the gas. Also, the attraction or repulsion between the individual gas molecules and the container is negligible. Further, in an ideal gas, the molecules are considered to be perfectly elastic, and there is no internal energy loss resulting from collision between the molecules. Such ideal gases are said
to obey several gas laws, such as Boyle's law, Charles's law, and the ideal gas law or the perfect gas equation. We will first discuss the behavior of ideal gases and then follow it up with the behavior of real gases.

If $M$ represents the molecular weight of a gas and the mass of a certain quantity of gas is $m$, the number of moles is given by

$$
\begin{equation*}
n=\frac{m}{M} \tag{1.7}
\end{equation*}
$$

where $n$ is the number that represents the number of moles in the given mass.
For example, the molecular weight of methane is 16.043 . Therefore, 50 lb of methane will contain approximately 3 moles.

The ideal gas law, sometimes referred to as the perfect gas equation, simply states that the pressure, volume, and temperature of the gas are related to the number of moles by the following equation:

$$
\begin{equation*}
P V=n R T \quad \text { (USCS units) } \tag{1.8}
\end{equation*}
$$

where
$P=$ absolute pressure, pounds per square inch absolute (psia)
$V=$ gas volume, $\mathrm{ft}^{3}$
$n=$ number of lb moles as defined in Equation 1.7
$R=$ universal gas constant, psia $\mathrm{ft}^{3} / \mathrm{lb}$ mole ${ }^{\circ} \mathrm{R}$
$T=$ absolute temperature of gas, ${ }^{\circ} \mathrm{R}\left({ }^{\circ} \mathrm{F}+460\right)$
The universal gas constant $R$ has a value of $10.73 \mathrm{psia}^{\mathrm{ft}} / \mathrm{lb}^{\mathrm{mole}}{ }^{\circ} \mathrm{R}$ in USCS units. We can combine Equation 1.7 with Equation 1.8 and express the ideal gas equation as follows:

$$
\begin{equation*}
P V=\frac{m R T}{M} \tag{1.9}
\end{equation*}
$$

All symbols are as defined previously.
It should be noted that the constant $R$ is the same for all ideal gases and, hence, it is called the universal gas constant.

It has been found that the ideal gas equation is correct only at low pressures close to the atmospheric pressure. Since gas pipelines generally operate at pressures higher than atmospheric pressures, we must modify Equation 1.9 to take into account the effect of compressibility. The latter is accounted for by using a term called the compressibility factor, or gas deviation factor. We will discuss real gases and the compressibility factor later in this chapter.

In the ideal gas Equation 1.9, the pressures and temperatures must be in absolute units. Absolute pressure is defined as the gauge pressure (as measured by a gauge) plus the local atmospheric pressure.

Therefore,

$$
\begin{equation*}
P_{\mathrm{abs}}=P_{\mathrm{gauge}}+P_{\mathrm{atm}} \tag{1.10}
\end{equation*}
$$

Thus, if the gas pressure is 200 psig and the atmospheric pressure is 14.7 psia , we get the absolute pressure of the gas as

$$
P_{\mathrm{abs}}=200+14.7=214.7 \mathrm{psia}
$$

Absolute pressure is expressed as psia, whereas the gauge pressure is expressed as psig. The adder to the gauge pressure, which is the local atmospheric pressure, is also called the base pressure. In SI units, 500 kPa gauge pressure is equal to 601 kPa absolute pressure if the base pressure is 101 kPa . Pressure in USCS units is expressed in pounds per square inch, or psi. In SI units, pressure is expressed in kilopascal $(\mathrm{kPa})$, megapascal ( MPa ), or Bar. Refer to Appendix A for unit conversion charts.

The absolute temperature is measured above a certain datum. In USCS units, the absolute scale of temperature is designated as degree Rankin $\left({ }^{\circ} \mathrm{R}\right)$ and is equal to the sum of the temperature in ${ }^{\circ} \mathrm{F}$ and the constant 460 . In SI units the absolute temperature scale is referred to as degree Kelvin (K). Absolute temperature in K is equal to ${ }^{\circ} \mathrm{C}+273$.

Therefore,

$$
\begin{align*}
{ }^{\circ} \mathrm{R} & ={ }^{\circ} \mathrm{F}+460  \tag{1.11}\\
\mathrm{~K} & ={ }^{\circ} \mathrm{C}+273 \tag{1.12}
\end{align*}
$$

It is customary to drop the degree symbol for absolute temperature in Kelvin.
Ideal gases also obey Boyle's law and Charles's law. Boyle's law relates the pressure and volume of a given quantity of gas when the temperature is kept constant. Constant temperature is also called isothermal condition. Boyle's law is stated as follows:

$$
\begin{equation*}
\frac{P_{1}}{P_{2}}=\frac{V_{2}}{V_{1}} \quad \text { or } \quad P_{1} V_{1}=P_{2} V_{2} \tag{1.13}
\end{equation*}
$$

where $P_{1}$ and $V_{1}$ are the pressure and volume at condition 1 and $P_{2}$ and $V_{2}$ are the corresponding values at some other condition 2 where the temperature is not changed.

Therefore, according to Boyle's law, a given quantity of gas under isothermal conditions will double in volume if its pressure is halved and vice versa. In other words, the pressure is inversely proportional to the volume, provided the temperature is maintained constant. Since density and volume are inversely related, Boyle's law also means that the pressure is directly proportional to the density at constant temperature. Thus, a given quantity of gas at a fixed temperature will double its density when the pressure is doubled. Similarly, a $10 \%$ reduction in pressure will cause the density to decrease by the same amount.

Charles's law states that for constant pressure, the gas volume is directly proportional to its temperature. Similarly, if volume is kept constant, the pressure varies directly as the temperature. Therefore, we can state the following:

$$
\begin{align*}
& \frac{V_{1}}{V_{2}}=\frac{T_{1}}{T_{2}} \quad \text { at constant pressure }  \tag{1.14}\\
& \frac{P_{1}}{P_{2}}=\frac{T_{1}}{T_{2}} \quad \text { at constant volume } \tag{1.15}
\end{align*}
$$

Therefore, according to Charles's law, for an ideal gas at constant pressure, the volume will change in the same proportion as its temperature. Thus, a $20 \%$ increase in temperature will cause a $20 \%$ increase in volume as long as the pressure does not change. Similarly, if volume is kept constant, a $20 \%$ increase in temperature will result in the same percentage increase in gas pressure. Constant pressure is also known as isobaric condition.

## Example 2

A certain mass of gas occupies a volume of $1000 \mathrm{ft}^{3}$ at 60 psig . If temperature is constant (isothermal) and the pressure increases to 120 psig , what is the final volume of the gas? The atmospheric pressure is 14.7 psi .

## Solution

Boyle's law can be applied because the temperature is constant. Using Equation 1.13, we can write

$$
V_{2}=\frac{P_{1} V_{1}}{P_{2}}
$$

or

$$
V_{2}=\frac{(60+14.7) \times 1000}{120+14.7}=554.57 \mathrm{ft}^{3}
$$

Note that the pressures must be converted to absolute values before being used in Equation 1.13.

## Example 3

At 75 psig and $70^{\circ} \mathrm{F}$, a gas has a volume of $1000 \mathrm{ft}^{3}$. If the volume is kept constant and the gas temperature increases to $120^{\circ} \mathrm{F}$, what is the final pressure of the gas? For constant pressure at 75 psig , if the temperature increases to $120^{\circ} \mathrm{F}$, what is the final volume? Use 14.7 psi for base pressure.

## Solution

Since the volume is constant in the first part of the problem, Charles's law applies.
From Equation 1.15

$$
\frac{75+14.7}{P_{2}}=\frac{70+460}{120+460}
$$

Solving for $P_{2}$, we get

$$
P_{2}=98.16 \mathrm{psia} \text { or } 83.46 \mathrm{psig}
$$

For the second part, the pressure is constant and Charles's law can be applied.

From Equation 1.14, we get

$$
\frac{1000}{V_{2}}=\frac{70+460}{120+460}
$$

Solving for $V_{2}$ we get

$$
V_{2}=1094.34 \mathrm{ft}^{3}
$$

## Example 4

An ideal gas occupies a tank volume of $250 \mathrm{ft}^{3}$ at a pressure of 80 psig and temperature of $110^{\circ} \mathrm{F}$.
(1) What is the gas volume at standard conditions of 14.73 psia and $60^{\circ} \mathrm{F}$ ? Assume atmospheric pressure is 14.6 psia.
(2) If the gas is cooled to $90^{\circ} \mathrm{F}$, what is the gas pressure?

## Solution

(1) Initial conditions

$$
\begin{aligned}
& P_{1}=80+14.6=94.6 \mathrm{psia} \\
& V_{1}=250 \mathrm{ft}^{3} \\
& T_{1}=110+460=570^{\circ} \mathrm{R}
\end{aligned}
$$

Final conditions

$$
P_{2}=14.73 \mathrm{psia}
$$

$V_{2}$ is to be calculated

$$
T_{2}=60+460=520^{\circ} \mathrm{R}
$$

Using the ideal gas Equation 1.8, we can state that

$$
\begin{aligned}
\frac{94.6 \times 250}{570} & =\frac{14.73 V_{2}}{520} \\
V_{2} & =1464.73 \mathrm{ft}^{3}
\end{aligned}
$$

(2) When the gas is cooled to $90^{\circ} \mathrm{F}$, the final conditions are

$$
\begin{aligned}
& T_{2}=90+460=550^{\circ} \mathrm{R} \\
& V_{2}=250 \mathrm{ft}^{3}
\end{aligned}
$$

$P_{2}$ is to be calculated

The initial conditions are

$$
\begin{aligned}
& P_{1}=80+14.6=94.6 \mathrm{psia} \\
& V_{1}=250 \mathrm{ft}^{3} \\
& T_{1}=110+460=570^{\circ} \mathrm{R}
\end{aligned}
$$

It can be seen that the volume of gas is constant (tank volume) and the temperature reduces from $110^{\circ} \mathrm{F}$ to $90^{\circ} \mathrm{F}$. Therefore, using Charles's law and Equation 1.15, we can calculate the final pressure as follows.

$$
\frac{94.6}{P_{2}}=\frac{570}{550}
$$

Solving for $P_{2}$, we get

$$
P_{2}=91.28 \mathrm{psia}=91.28-14.6=76.68 \mathrm{psig}
$$

### 1.7 REAL GASES

When dealing with real gases, we can apply the ideal gas equation discussed in the preceding sections and get reasonably accurate results only when the pressures are close to the atmospheric pressure. When pressures are higher, the ideal gas equation will not be accurate for most real gases. The error in calculations at high pressures using the ideal gas equation may be as high as $500 \%$ in some instances. This compares with errors of 2 to $3 \%$ at low pressures. At higher temperatures and pressures, the "equation of state" that relates pressure, volume, and temperature is used to calculate the properties of gases. Many of these correlations require a computer program to get accurate results in a reasonable amount of time. However, we can modify the ideal gas equation and obtain reasonably accurate results fairly quickly using manual calculations.

Two terms called critical temperature and critical pressure need to be defined. The critical temperature of a pure gas is defined as the temperature above which a gas cannot be compressed to form a liquid, regardless of the pressure. The critical pressure is defined as the minimum pressure that is required at the critical temperature to compress a gas into a liquid.

Real gases can be considered to follow a modified form of the ideal gas law discussed in Section 1.6. The modifying factor is included in the gas property known as the compressibility factor $Z$. This is also called the gas deviation factor. It can be defined as the ratio of the gas volume at a given temperature and pressure to the volume the gas would occupy if it were an ideal gas at the same temperature and pressure. $Z$ is a dimensionless number less than 1.0 and it varies with temperature, pressure, and composition of the gas.

Using the compressibility factor $Z$, the ideal gas equation is modified for real gas as follows:

$$
\begin{equation*}
P V=Z n R T \quad \text { (USCS units) } \tag{1.16}
\end{equation*}
$$

where
$P=$ absolute pressure of gas, psia
$V=$ volume of gas, $\mathrm{ft}^{3}$
$Z=$ gas compressibility factor, dimensionless
$T=$ absolute temperature of gas, ${ }^{\circ} \mathrm{R}$
$n=$ number of lb moles as defined in Equation 1.7
$R=$ universal gas constant, $10.73 \mathrm{psia}^{3}{ }^{3} / \mathrm{lb} \mathrm{mole}^{\circ} \mathrm{R}$
The theorem known as corresponding states says that the extent of deviation of a real gas from the ideal gas equation is the same for all real gases when the gases are at the same corresponding state. The corresponding state can be represented by the two parameters called reduced temperature and reduced pressure. The reduced temperature is the ratio of the temperature of the gas to its critical temperature. Similarly, the reduced pressure is the ratio of the gas pressure to its critical pressure as indicated in the following equations:

$$
\begin{align*}
T_{r} & =\frac{T}{T_{c}}  \tag{1.17}\\
P_{r} & =\frac{P}{P_{c}} \tag{1.18}
\end{align*}
$$

where
$P=$ absolute pressure of gas, psia
$T=$ absolute temperature of gas, ${ }^{\circ} \mathrm{R}$
$T_{r}=$ reduced temperature, dimensionless
$P_{r}=$ reduced pressure, dimensionless
$T_{c}=$ critical temperature, ${ }^{\circ} \mathrm{R}$
$P_{c}=$ critical pressure, psia
For example, suppose the critical temperature and critical pressure of methane are $343^{\circ} \mathrm{R}$ and 666 psia , respectively; the reduced temperature and pressure of the gas at $80^{\circ} \mathrm{F}$ and 1000 psia pressure are as follows:

$$
T_{r}=\frac{80+460}{343}=1.57
$$

and

$$
P_{r}=\frac{1000}{666}=1.50
$$

Therefore, according to the theorem of corresponding states, two gases, A and B, may be at different temperatures and pressures; however, if their reduced temperature and reduced pressure are the same, then their gas deviation factors $(Z)$ will be the same. Therefore, generalized plots showing the variation of $Z$ with reduced temperature and reduced pressure can be used for most gases for calculating the


Figure 1.2 Compressibility factor chart for natural gases. (From Gas Processors Suppliers Association, Eng. Data Book, Vol. II. With permission.)
compressibility factor. Such a plot is shown in Figure 1.2. The calculation of the compressibility factor $Z$ will be discussed in detail in Section 1.11 of this chapter.

### 1.8 NATURAL GAS MIXTURES

As mentioned earlier, the critical temperature of a pure gas is defined as the temperature above which it cannot be liquefied, whatever the pressure of the gas. Similarly, the critical pressure is defined as the pressure above which liquid and gas cannot
coexist, regardless of the temperature. When the gas consists of a mixture of different components, the critical temperature and critical pressure are called the pseudocritical temperature and pseudo-critical pressure, respectively. If we know the composition of the gas mixture, we can calculate these pseudo-critical values of the mixture, using the critical pressure and temperature values of the pure components that constitute the gas mixture.

The reduced temperature is defined as the ratio of the temperature of the gas to its critical temperature. Similarly, the reduced pressure is the ratio of gas pressure to its critical pressure. Both temperature and pressure are stated in absolute units. Similar to the pseudo-critical temperature and pseudo-critical pressure discussed above, for a gas mixture, we can define the pseudo-reduced temperature and the pseudo-reduced pressure.

Thus,

$$
\begin{align*}
T_{p r} & =\frac{T}{T_{p c}}  \tag{1.19}\\
P_{p r} & =\frac{P}{P_{p c}} \tag{1.20}
\end{align*}
$$

where
$P=$ absolute pressure of gas mixture, psia
$T=$ absolute temperature of gas mixture, ${ }^{\circ} \mathrm{R}$
$T_{p r}=$ pseudo-reduced temperature, dimensionless
$P_{p r}=$ pseudo-reduced pressure, dimensionless
$T_{p c}=$ pseudo-critical temperature, ${ }^{\circ} \mathrm{R}$
$P_{p c}=$ pseudo-critical pressure, psia
In hydrocarbon mixtures, frequently we refer to gas components as $\mathrm{C}_{1}, \mathrm{C}_{2}, \mathrm{C}_{3}$, etc. These are equivalent to $\mathrm{CH}_{4}$ (methane), $\mathrm{C}_{2} \mathrm{H}_{6}$ (ethane), $\mathrm{C}_{3} \mathrm{H}_{8}$ (propane), and so on. A natural gas mixture that consists of components such as $\mathrm{C}_{1}, \mathrm{C}_{2}, \mathrm{C}_{3}$, and so forth is said to have an apparent molecular weight as defined by the equation

$$
\begin{equation*}
M_{a}=\Sigma y_{i} M_{i} \tag{1.21}
\end{equation*}
$$

where
$M_{a}=$ apparent molecular weight of gas mixture
$y_{i}=$ mole fraction of gas component $i$
$M_{i}=$ molecular weight of gas component $i$
In a similar manner, from the given mole fractions of the gas components, we use Kay's rule to calculate the average pseudo-critical properties of the gas mixture.

$$
\begin{align*}
& T_{p c}=\Sigma y_{i} T_{c}  \tag{1.22}\\
& P_{p c}=\Sigma y_{i} P_{c} \tag{1.23}
\end{align*}
$$

where $T_{c}$ and $P_{c}$ are the critical temperature and pressure, respectively, of the pure component $\left(\mathrm{C}_{1}, \mathrm{C}_{2}\right.$, etc.) and $y_{i}$ refers to the mole fraction of the component. $T_{p c}$ and $P_{p c}$ are the average pseudo-critical temperature and pseudo-critical pressure, respectively, of the gas mixture.

## Example 5

Calculate the apparent molecular weight of a natural gas mixture that has $85 \%$ methane, $9 \%$ ethane, $4 \%$ propane, and $2 \%$ normal butane as shown below:

| Component | Percent | Molecular Weight |
| :---: | :---: | :---: |
| $\mathrm{C}_{1}$ | 85 | 16.01 |
| $\mathrm{C}_{2}$ | 9 | 30.10 |
| $\mathrm{C}_{3}$ | 4 | 44.10 |
| $\mathrm{n}-\mathrm{C}_{4}$ | 2 | 58.10 |
| Total | 100 |  |

## Solution

Using Equation 1.21, we get

$$
M_{a}=(0.85 \times 16.01)+(0.09 \times 30.1)+(0.04 \times 44.1)+(0.02 \times 58.1)=19.24
$$

Therefore, the apparent molecular weight of the gas mixture is 19.24 .

## Example 6

Calculate the pseudo-critical temperature and the pseudo-critical pressure of a natural gas mixture consisting of $83 \%$ methane, $12 \%$ ethane, and $5 \%$ propane.

The critical properties of $\mathrm{C}_{1}, \mathrm{C}_{2}$, and $\mathrm{C}_{3}$ components are as follows:

|  | Critical <br> Components | Critical <br> Pressure, <br> psia |
| :---: | :---: | :---: |
| $\mathrm{C}_{1}$ | 343 | 666 |
| $\mathrm{C}_{2}$ | 550 | 707 |
| $\mathrm{C}_{3}$ | 666 | 617 |

## Solution

Using the given data, from Equation 1.22 and Equation 1.23, we calculate the pseudocritical properties as follows:

$$
T_{p c}=(0.83 \times 343)+(0.12 \times 550)+(0.05 \times 666)=383.99^{\circ} \mathrm{R}
$$

and

$$
P_{p c}=(0.83 \times 666)+(0.12 \times 707)+(0.05 \times 617)=668.47 \mathrm{psia}
$$

Therefore, the pseudo-critical temperature of the gas mixture $=383.99^{\circ} \mathrm{R}$ and the pseudo-critical pressure of the gas mixture $=668.47 \mathrm{psia}$.

## Example 7

If the temperature of the gas in the previous example is $70^{\circ} \mathrm{F}$ and the average gas pressure is 1200 psig , what are the pseudo-reduced temperature and pseudo-reduced pressure of this gas? Use 14.7 psia for base pressure.

## Solutions

From Equation 1.19 and Equation 1.20, we get

$$
\text { The pseudo-reduced temperature } T_{p r}=\frac{70+460}{383.99}=1.38
$$

$$
\text { The pseudo-reduced pressure } P_{p r}=\frac{1200+14.7}{668.47}=1.82
$$

### 1.9 PSEUDO-CRITICAL PROPERTIES FROM GAS GRAVITY

If the percentages of the various components in the natural gas mixture are not available, we can calculate approximate values of the pseudo-critical properties of the gas mixture if we know the gas gravity. The pseudo-critical properties are calculated, approximately, from the following equations:

$$
\begin{align*}
& T_{p c}=170.491+307.344 G  \tag{1.24}\\
& P_{p c}=709.604-58.718 G \tag{1.25}
\end{align*}
$$

where
$G=$ gas gravity (air $=1.00$ )
$T_{p c}=$ pseudo-critical temperature, ${ }^{\circ} \mathrm{R}$
$P_{p c}=$ pseudo-critical pressure, psia

## Example 8

Calculate the gravity of a natural gas mixture consisting of $83 \%$ methane, $12 \%$ ethane, and $5 \%$ propane. From the gas gravity, calculate the pseudo-critical temperature and pseudo-critical pressure for this natural gas mixture.

Solution

Using Kay's rule for multicomponent mixtures and Equation 1.4 for gas gravity, we get

$$
G=\frac{(0.83 \times 16.04)+(0.12 \times 30.07)+(0.05 \times 44.10)}{29}=0.6595
$$

Therefore, the gas gravity is 0.6595 .

From Equation 1.24 and Equation 1.25, we calculate the pseudo-critical properties as follows:

$$
\begin{aligned}
& T_{p c}=170.491+307.344 \times(0.6595)=373.18^{\circ} \mathrm{R} \\
& P_{p c}=709.604-58.718 \times(0.6595)=670.88 \mathrm{psia}
\end{aligned}
$$

Comparing the above values with the values calculated using the more accurate method in the previous example, we find that the value of $T_{p c}$ is off by $2.8 \%$ and $P_{p c}$ is off by $0.4 \%$. These differences are small enough for most calculations related to natural gas pipeline hydraulics.

### 1.10 IMPACT OF SOUR GAS AND NON-HYDROCARBON COMPONENTS

The Standing-Katz chart used for determining the compressibility factor (discussed in Section 1.11) of a gas mixture is accurate only if the amount of non-hydrocarbon components is small. Since sour gases contain carbon dioxide and hydrogen sulfide, adjustments must be made to take into account these components in calculations of the pseudo-critical temperature and pseudo-critical pressure. This method is described below. Depending on the amounts of carbon dioxide and hydrogen sulfide present in the sour gas, we calculate an adjustment factor $\varepsilon$ from

$$
\begin{equation*}
\varepsilon=120\left(A^{0.9}-A^{1.6}\right)+15\left(B^{0.5}-B^{4.0}\right) \tag{1.26}
\end{equation*}
$$

where
$\varepsilon=$ adjustment factor, ${ }^{\circ} \mathrm{R}$
$A=$ sum of the mole fractions of $\mathrm{CO}_{2}$ and $\mathrm{H}_{2} \mathrm{~S}$
$B=$ mole fraction of $\mathrm{H}_{2} \mathrm{~S}$
The pseudo-critical temperature is modified to get the adjusted pseudo-critical temperature $T_{p c}^{\prime}$ from the following equation:

$$
\begin{equation*}
T_{p c}^{\prime}=T_{p c}-\varepsilon \tag{1.27}
\end{equation*}
$$

where
$T_{p c}^{\prime}=$ adjusted pseudo-critical temperature, ${ }^{\circ} \mathrm{R}$

Similarly, the pseudo-critical pressure is adjusted as follows:

$$
\begin{equation*}
P_{p c}^{\prime}=\frac{P_{p c} \times T_{p c}^{\prime}}{T_{p c}+B(1-B) \varepsilon} \tag{1.28}
\end{equation*}
$$

where
$P_{p c}^{\prime}=$ adjusted pseudo-critical pressure, psia.

## Example 9

The pseudo-critical temperature and the pseudo-critical pressure of a natural gas mixture were calculated as $370^{\circ} \mathrm{R}$ and 670 psia, respectively. If the $\mathrm{CO}_{2}$ content is $10 \%$ and $\mathrm{H}_{2} \mathrm{~S}$ is $20 \%$, calculate the adjustment factor $\varepsilon$ and the adjusted values of the pseudo-critical temperature and pressure.

Solution

$$
\begin{aligned}
& A=0.10+0.20=0.30 \\
& B=0.20
\end{aligned}
$$

The adjustment factor $\varepsilon$ from Equation 1.26 is

$$
\varepsilon=120\left(0.30^{0.9}-0.30^{1.6}\right)+15\left(0.20^{0.5}-0.20^{4}\right)=29.8082^{\circ} \mathrm{R}
$$

Therefore, the adjustment factor $\varepsilon$ is $29.81^{\circ} \mathrm{R}$.
The adjusted values of the pseudo-critical properties are found using Equation 1.27 and Equation 1.28 as follows:

$$
T_{p c}^{\prime}=370-29.81=340.19^{\circ} \mathrm{R}
$$

and

$$
P_{p c}^{\prime}=\frac{670 \times 340.19}{370+0.20(1-0.20) \times 29.8082}=608.18 \mathrm{psia}
$$

Therefore, the adjusted pseudo-critical temperature $=340.19^{\circ} \mathrm{R}$ and the adjusted pseudo-critical pressure $=608.18 \mathrm{psia}$.

### 1.11 COMPRESSIBILITY FACTOR

The compressibility factor, or gas deviation factor, was briefly mentioned in Section 1.6. It is a measure of how close a real gas is to an ideal gas. The compressibility factor is defined as the ratio of the gas volume at a given temperature and pressure to the volume the gas would occupy if it were an ideal gas at the same temperature and pressure. The compressibility factor is a dimensionless number close to 1.00 and is
a function of the gas gravity, gas temperature, gas pressure, and the critical properties of the gas. As an example, a particular natural gas mixture can have a compressibility factor equal to 0.87 at 1000 psia and $80^{\circ} \mathrm{F}$. Charts have been constructed that depict the variation of $Z$ with the reduced temperature and reduced pressure. Another term, the "supercompressibility factor," $F_{p v}$, which is related to the compressibility factor $Z$, is defined as follows:

$$
\begin{equation*}
F_{p v}=\frac{1}{\sqrt{Z}} \tag{1.29}
\end{equation*}
$$

or

$$
\begin{equation*}
Z=\frac{1}{\left(F_{p v}\right)^{2}} \tag{1.30}
\end{equation*}
$$

As an example, if the compressibility factor $Z=0.85$, using Equation 1.29, we calculate the supercompressibility factor, $F_{p v}$, as follows:

$$
F_{p v}=\frac{1}{\sqrt{0.85}}=1.0847
$$

There are several approaches to calculating the compressibility factor for a particular gas temperature $T$ and pressure $P$. One method uses the critical temperature and critical pressure of the gas mixture. First, the reduced temperature, $T r$, and reduced pressure, $P r$, are calculated from the given gas temperature and gas pressure and the critical temperature and critical pressure using Equation 1.17 and Equation 1.18.

Once we know the values of $\operatorname{Tr}$ and $\operatorname{Pr}$, the compressibility factor can be found from charts similar to the Standing-Katz chart. This will be illustrated using an example.

The following methods are available to calculate the compressibility factor:
a. Standing-Katz method
b. Dranchuk, Purvis, and Robinson method
c. AGA method
d. CNGA method

Although the Standing-Katz method is the most popular, we will discuss this as well as the AGA and CNGA methods.

### 1.11.1 Standing-Katz Method

The Standing-Katz method of calculating compressibility factor is based on the use of a graph that has been constructed for binary mixtures and saturated hydrocarbon vapor. This method is used generally for sweet natural gas mixtures containing various hydrocarbon components. When the natural gas mixture contains appreciable amounts of non-hydrocarbons such as nitrogen, hydrogen sulfide, and carbon dioxide, certain corrections must be applied for these components. These adjustments are applied to
the critical temperatures and pressures and were discussed in Section 1.10. The Standing-Katz chart for compressibility factors is shown in Figure 1.2. The use of this chart will be explained in an example problem.

### 1.11.2 Dranchuk, Purvis, and Robinson Method

In this method of calculating the compressibility factor, the Benedict-Webb-Rubin equation of state is used to correlate the Standing-Katz chart. The coefficients $A_{1}$, $A_{2}$, etc. are used in a polynomial function of the reduced density $\rho_{r}$ as follows:

$$
\begin{equation*}
Z=1+\left(A_{1}+\frac{A_{2}}{T_{p r}}+\frac{A_{3}}{T_{p r}^{3}}\right) \rho_{r}+\left(A_{4}+\frac{A_{5}}{T_{p r}}\right) \rho_{r}^{2}+\frac{A_{5} A_{6} \rho_{r}^{5}}{T_{p r}}+\frac{A_{7} \rho_{r}^{3}}{T_{p r}^{3}\left(1+A_{8} \rho_{r}^{2}\right) e^{\left(-A_{8} \rho_{r}^{2}\right)}} \tag{1.31}
\end{equation*}
$$

where

$$
\begin{equation*}
\rho_{r}=\frac{0.27 P_{p r}}{Z T_{p r}} \tag{1.32}
\end{equation*}
$$

and
$A_{1}=0.31506237$
$A_{2}=-1.04670990$
$A_{3}=-0.57832729$
$A_{4}=0.53530771$
$A_{5}=-0.61232032$
$A_{6}=-0.10488813$
$A_{7}=0.68157001$
$A_{8}=0.68446549$
$P_{p r}=$ pseudo-reduced pressure
$T_{p r}=$ pseudo-reduced temperature
Other symbols have been defined earlier.

### 1.11.3 American Gas Association (AGA) Method

The AGA method for the compressibility factor uses a complicated mathematical algorithm and, therefore, does not lend itself easily to manual calculations. Generally, a computer program is used to calculate the compressibility factor. Mathematically, the AGA method is represented by the following function:

$$
\begin{equation*}
Z=\text { Function (gas properties, pressure, temperature) } \tag{1.33}
\end{equation*}
$$

where gas properties include the critical temperature, critical pressure, and gas gravity.

The AGA-IGT Report No. 10 describes in detail this method of calculating $Z$. This approach is valid for gas temperatures in the range of 30 to $120^{\circ} \mathrm{F}$ and for pressures not exceeding 1380 psig. The compressibility factors calculated using this method are quite accurate and generally within $0.03 \%$ of those calculated using the Standing-Katz chart in this range of temperatures and pressures. When temperatures and pressures are higher than these values, the compressibility factor calculated using this method is within $0.07 \%$ of the value obtained from the Standing-Katz chart.

The reader may also refer to the AGA publication Report No. 8, Second Edition, November 1992, for more information on compressibility factor calculation methods.

### 1.11.4 California Natural Gas Association (CNGA) Method

This is a fairly simple equation for quickly calculating the compressibility factor when the gas gravity, temperature, and pressure are known. The following equation is used for calculating the compressibility factor $Z$ :

$$
\begin{equation*}
Z=\frac{1}{\left[1+\left(\frac{P_{\text {avg }} 344,400(10)^{1.785 G}}{T_{f}^{3.825}}\right)\right]} \tag{1.34}
\end{equation*}
$$

This formula for the compressibility factor is valid when the average gas pressure, $P_{\text {avg }}$, is more than 100 psig . For pressures less than $100 \mathrm{psig}, Z$ is approximately equal to 1.00
where
$P_{\text {avg }}=$ average gas pressure, psig
$T_{f}=$ average gas temperature, ${ }^{\circ} \mathrm{R}$
$G=$ gas gravity (air = 1.00)

Note that the pressure used in Equation 1.34 is the gauge pressure.
In a gas pipeline, the pressure varies along the length of the pipeline. The compressibility factor $Z$ also varies and must therefore be calculated for an average pressure at any location on the pipeline. If two points along the pipeline are at pressures $P_{1}$ and $P_{2}$, we could use an average pressure of $\frac{P_{1}+P_{2}}{2}$. However, the following formula is used for a more accurate value of the average pressure:

$$
\begin{equation*}
P_{\text {avg }}=\frac{2}{3}\left(P_{1}+P_{2}-\frac{P_{1} \times P_{2}}{P_{1}+P_{2}}\right) \tag{1.35}
\end{equation*}
$$

Another form of the average pressure in a pipe segment is

$$
\begin{equation*}
P_{\text {avg }}=\frac{2}{3}\left(\frac{P_{1}^{3}-P_{2}^{3}}{P_{1}^{2}-P_{2}^{2}}\right) \tag{1.36}
\end{equation*}
$$

## Example 10

Using the Standing-Katz compressibility chart, calculate the compressibility factor for the gas in Example 7 at $70^{\circ} \mathrm{F}$ and 1200 psig . Use the values of $T_{p c}$ and $P_{p c}$ calculated in Example 7.

## Solutions

From previous Example 7,
The pseudo-reduced temperature $T_{p r}=1.38$
The pseudo-reduced pressure $P_{p r}=1.82$
Using the Standing-Katz chart in Figure 1.2, we read the value of $Z$ as

$$
Z=0.770
$$

## Example 11

A natural gas mixture consists of the following components:

| Component | Mole Fraction $\boldsymbol{y}$ |
| :---: | :---: |
| $\mathrm{C}_{1}$ | 0.780 |
| $\mathrm{C}_{2}$ | 0.005 |
| $\mathrm{C}_{3}$ | 0.002 |
| $\mathrm{~N}_{2}$ | 0.013 |
| $\mathrm{CO}_{2}$ | 0.016 |
| $\mathrm{H}_{2} \mathrm{~S}$ | 0.184 |

(a) Calculate the apparent molecular weight of the gas, gas gravity, and the pseudocritical temperature and pseudo-critical pressure.
(b) Calculate the compressibility factor of the gas at $90^{\circ} \mathrm{F}$ and 1200 psia .

Solution
Using Table 1.1, we create the following table showing the molecular weight $(M)$, critical temperature ( $T c$ ), and critical pressure $(P c)$ for each of the component gases. The molecular weight of the mixture and the pseudo-critical temperature and pseudocritical pressure are then calculated using Equation 1.22 and Equation 1.23.

| Component | $\boldsymbol{y}$ | $\boldsymbol{M}$ | $\boldsymbol{y} \boldsymbol{M}$ | $\boldsymbol{T}_{\boldsymbol{c}}$ | $\boldsymbol{P}_{\boldsymbol{c}}$ | $\boldsymbol{y} \boldsymbol{T}_{\boldsymbol{c}}$ | $\boldsymbol{y} \boldsymbol{P}_{\boldsymbol{c}}$ |
| :---: | :---: | :---: | ---: | ---: | ---: | ---: | ---: |
| $\mathrm{C}_{1}$ | 0.780 | 16.04 | 12.5112 | 343.34 | 667.00 | 267.81 | 520.26 |
| $\mathrm{C}_{2}$ | 0.005 | 30.07 | 0.1504 | 550.07 | 707.80 | 2.75 | 3.54 |
| $\mathrm{C}_{3}$ | 0.002 | 44.10 | 0.0882 | 665.93 | 615.00 | 1.33 | 1.23 |
| $\mathrm{~N}_{2}$ | 0.013 | 28.01 | 0.3641 | 227.52 | 492.80 | 2.96 | 6.41 |
| $\mathrm{CO}_{2}$ | 0.016 | 44.01 | 0.7042 | 547.73 | 1070.00 | 8.76 | 17.12 |
| $\mathrm{H}_{2} \mathrm{~S}$ | 0.184 | 34.08 | 6.2707 | 672.40 | 1300.00 | 123.72 | 239.20 |
| Total | 1.000 |  | 20.0888 |  |  | 407.33 | 787.76 |

Therefore, the apparent molecular weight of the natural gas is

$$
M w=\Sigma y M=20.09
$$

The gas gravity, using Equation 1.4, is

$$
G=\frac{20.09}{29}=0.6928
$$

Next, we calculate the pseudo-critical values

$$
\begin{aligned}
\text { Pseudo-critical temperature } & =\Sigma y T_{c}=407.33^{\circ} \mathrm{R} \\
\text { Pseudo-critical pressure } & =\Sigma y P_{c}=787.76 \mathrm{psia}
\end{aligned}
$$

Since this sour gas contains more than 5\% non-hydrocarbons, we will adjust the pseudo-critical properties using Equation 1.26 through Equation 1.28.

We first calculate the temperature adjustment factor $\mathcal{\varepsilon}$, using Equation 1.26, as follows:

$$
A=(0.016+0.184)=0.20
$$

and

$$
B=0.184
$$

Therefore, the adjustment factor is

$$
\varepsilon=120\left[(0.2)^{0.9}-(0.2)^{1.6}\right]+15\left[(0.184)^{0.5}-(0.184)^{4.0}\right]=25.47^{\circ} \mathrm{R}
$$

The adjusted pseudo-critical temperature from Equation 1.27 is

$$
T_{p c}^{\prime}=407.33-25.47=381.86^{\circ} \mathrm{R}
$$

and the adjusted pseudo-critical pressure from Equation 1.28 is

$$
P_{p c}^{\prime}=\frac{787.76 \times 381.86}{407.33+0.184 \times(1-0.184) \times 25.47}=731.63 \mathrm{psia}
$$

Next, we calculate the compressibility factor at $90^{\circ} \mathrm{F}$ and 1200 psia pressure using these values as follows:

From Equation 1.19,

$$
\text { pseudo-reduced temperature }=\frac{90+460}{381.86}=1.44
$$

From Equation 1.20,

$$
\text { pseudo-reduced pressure }=\frac{1200}{731.63}=1.64
$$

Finally, from the Standing-Katz chart, we get the compressibility factor for the reduced temperature and reduced pressure as

$$
Z=0.825
$$

## Example 12

The gravity of a natural gas mixture is 0.60 . Calculate the compressibility factor of this gas at 1200 psig pressure and a temperature of $70^{\circ} \mathrm{F}$, using the CNGA method.

Solution

Gas temperature $T_{f}=70+460=530^{\circ} \mathrm{R}$

From Equation 1.34, the $Z$ factor can be written as

$$
\frac{1}{Z}=1+\frac{1200 \times 344,400 \times(10)^{1.785 \times 0.60}}{530^{3.825}}=1.1849
$$

Therefore, solving for $Z$, we get

$$
Z=0.8440
$$

### 1.12 HEATING VALUE

The heating value of a gas is defined as the thermal energy per unit volume of the gas. It is expressed in Btu/ft ${ }^{3}$. For natural gas, it is approximately in the range of 900 to $1200 \mathrm{Btu} / \mathrm{ft}^{3}$. There are two heating values used in the industry. These are the lower heating value (LHV) and higher heating value (HHV). For a gas mixture, the term gross heating value is used. It is calculated based upon the heating values of the component gases and their mole fractions using the following equation:

$$
\begin{equation*}
H_{m}=\Sigma\left(y_{i} H_{i}\right) \tag{1.37}
\end{equation*}
$$

where
$H_{m}=$ gross heating value of mixture, Btu/ft ${ }^{3}$
$y_{i}=$ mole fraction or percent of gas component $i$
$H_{i}=$ heating value of gas component, Btu/ $\mathrm{ft}^{3}$

For example, a natural gas mixture consisting of $80 \%$ of gas A (heating value $=$ $900 \mathrm{Btu} / \mathrm{ft}^{3}$ ) and $20 \%$ of gas B (heating value $=1000 \mathrm{Btu} / \mathrm{ft}^{3}$ ) will have a gross heating value of $H_{m}=(0.8 \times 900)+(0.2 \times 1000)=920 \mathrm{Btu} / \mathrm{ft}^{3}$.

### 1.13 SUMMARY

We discussed several gas properties that influence gas pipeline transportation. The ideal gas equation was introduced along with Boyle's law and Charles's law, and how they can be applied with modifications to real gases and real gas mixtures was explained. The gas deviation factor, or compressibility factor, which modifies ideal gas behavior, was introduced. Critical properties of hydrocarbon gases and mixtures and the reduced temperature and pressure that determine the state of a gas were
explained. Variation of the compressibility factor with pressure and temperature was explored, and calculation methodologies using analytical and graphical approaches were covered. The influence of non-hydrocarbon components in a natural gas mixture was also discussed, along with correction factors for $\mathrm{CO}_{2}$ and $\mathrm{H}_{2} \mathrm{~S}$ in sour gas.

## PROBLEMS

1. A natural gas mixture consists of three components, $\mathrm{C}_{1}, \mathrm{C}_{2}$, and $\mathrm{C}_{3}$. Their mole fractions and viscosities at a particular temperature are indicated below:

| Component | Mole Fraction $\boldsymbol{y}$ | Viscosity, $\mathbf{c P}$ |
| :---: | :---: | :---: |
| $\mathrm{C}_{1}$ | 0.9000 | 0.0130 |
| $\mathrm{C}_{2}$ | 0.0800 | 0.0112 |
| $\mathrm{C}_{3}$ | 0.0200 | 0.0098 |
| Total | 1.000 |  |

Calculate the viscosity of the gas mixture.
2. At 100 psig and $75^{\circ} \mathrm{F}$, a gas has a volume of $800 \mathrm{ft}^{3}$. If the volume is kept constant and the gas temperature increases to $100^{\circ} \mathrm{F}$, what is the final pressure of the gas? Keeping the pressure constant at 100 psig , if the temperature increases to $100^{\circ} \mathrm{F}$, what is the final volume? Use 14.73 psi for base pressure.
3. Calculate the apparent molecular weight of a natural gas mixture that has $89 \%$ methane, $8 \%$ ethane, $2 \%$ propane, and $1 \%$ normal butane as shown below.

| Component | Percent | Molecular Weight |
| :---: | :---: | :---: |
| $\mathrm{C}_{1}$ | 89 | 16.01 |
| $\mathrm{C}_{2}$ | 8 | 30.10 |
| $\mathrm{C}_{3}$ | 2 | 44.10 |
| $\mathrm{C}_{4}$ | 1 | 58.10 |
| Total | 100 |  |

4. Calculate the pseudo-critical temperature and the pseudo-critical pressure of a natural gas mixture consisting of $89 \%$ methane, $8 \%$ ethane, and $3 \%$ propane. The critical properties of $\mathrm{C}_{1}, \mathrm{C}_{2}$, and $\mathrm{C}_{3}$ components are as follows:

| Components | Critical Temperature, ${ }^{\circ} \mathbf{R}$ | Critical Pressure, psia |
| :---: | :---: | :---: |
| $\mathrm{C}_{1}$ | 343 | 667 |
| $\mathrm{C}_{2}$ | 550 | 708 |
| $\mathrm{C}_{3}$ | 666 | 615 |

5. If the temperature of the gas in the previous example is $80^{\circ} \mathrm{F}$ and the average gas pressure is 1000 psig , what are the pseudo-reduced temperature and pseudoreduced pressure of this gas? Use 14.7 psia for base pressure.
6. Calculate the gravity of a natural gas mixture consisting of $84 \%$ methane, $10 \%$ ethane, and $6 \%$ propane. From the gas gravity, calculate the pseudo-critical temperature and pseudo-critical pressure for this natural gas mixture.
7. The pseudo-critical temperature and pressure of a natural gas mixture were calculated as $380^{\circ} \mathrm{R}$ and 675 psia. If the $\mathrm{CO}_{2}$ content is $12 \%$ and $\mathrm{H}_{2} \mathrm{~S}$ is $22 \%$, calculate the adjustment factor $\varepsilon$ and the adjusted values of the pseudo-critical temperature and pseudo-critical pressure.
8. Using the Standing-Katz compressibility chart, calculate the compressibility factor for the gas in Problem 7 at $80^{\circ} \mathrm{F}$ and 1000 psig. Use the values of $T_{p c}$ and $P_{p c}$ calculated in Problem 7.
9. A natural gas mixture consists of the following components:

| Component | Mole Fraction $\boldsymbol{y}$ |
| :---: | :---: |
| $\mathrm{C}_{1}$ | 0.850 |
| $\mathrm{C}_{2}$ | 0.004 |
| $\mathrm{C}_{3}$ | 0.002 |
| $\mathrm{~N}_{2}$ | 0.014 |
| $\mathrm{CO}_{2}$ | 0.010 |
| $\mathrm{H}_{2} \mathrm{~S}$ | 0.120 |

(a) Calculate the apparent molecular weight of the gas, gravity, and the pseudocritical temperature and pseudo-critical pressure.
(b) Calculate the compressibility factor of the gas at $100^{\circ} \mathrm{F}$ and 1400 psia .
10. The gravity of a natural gas mixture is 0.62 . Calculate the compressibility factor of this gas at 1400 psig and a temperature of $80^{\circ} \mathrm{F}$, using the CNGA method.

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## CHAPTER 2

## Pressure Drop Due to Friction

In this chapter we will discuss the various methods of calculating the pressure drop due to friction in a gas pipeline. The pipeline throughput (flow rate) will depend upon the gas properties, pipe diameter and length, initial gas pressure and temperature, and the pressure drop due to friction. Commonly used formulas will be reviewed and illustrated using examples. The impact of internal conditions of the pipe on the pipe capacity will also be explored.

### 2.1 BERNOULLI'S EQUATION

As gas flows through a pipeline, the total energy of the gas at various points consists of energy due to pressure, energy due to velocity, and energy due to position or elevation above an established datum. Bernoulli's equation simply connects these components of the energy of the flowing fluid to form an energy conservation equation. Bernoulli's equation is stated as follows, considering two points, 1 and 2, as shown in Figure 2.1.

$$
\begin{equation*}
Z_{A}+\frac{P_{A}}{\gamma}+\frac{V_{A}^{2}}{2 g}+H_{p}=Z_{B}+\frac{P_{B}}{\gamma}+\frac{V_{B}^{2}}{2 g}+h_{f} \tag{2.1}
\end{equation*}
$$

where $H_{p}$ is the equivalent head added to the fluid by a compressor at A and $h_{f}$ represents the total frictional pressure loss between points A and B .

Starting with the basic energy Equation 2.1, applying gas laws, and after some simplification, various formulas were developed over the years to predict the performance of a pipeline transporting gas. These formulas are intended to show the relationship between the gas properties, such as gravity and compressibility factor, with the flow rate, pipe diameter and length, and the pressures along the pipeline.

Thus, for a given pipe size and length, we can predict the flow rate possible through a pipeline based upon an inlet pressure and an outlet pressure of a pipe segment. Simplifications are sometimes introduced, such as uniform gas temperature and no heat transfer between the gas and the surrounding soil in a buried pipeline, in order


Figure 2.1 Energy of flow of a fluid.
to adopt these equations for manual calculations. With the advent of microcomputers, we are able to introduce heat transfer effects and, therefore, more accurately model gas pipelines, taking into consideration gas flow temperatures, soil temperatures, and thermal conductivities of pipe material, insulation, and soil. In this chapter we will concentrate on steady-state isothermal flow of gas in pipelines. Appendix D includes an output report from a commercial gas pipeline simulation model that takes into account heat transfer. For most practical purposes, the assumption of isothermal flow is good enough, since in long transmission lines the gas temperature reaches constant values, anyway.

### 2.2 FLOW EQUATIONS

Several equations are available that relate the gas flow rate with gas properties, pipe diameter and length, and upstream and downstream pressures. These equations are listed as follows:

1. General Flow equation
2. Colebrook-White equation
3. Modified Colebrook-White equation
4. AGA equation
5. Weymouth equation
6. Panhandle A equation
7. Panhandle B equation
8. IGT equation
9. Spitzglass equation
10. Mueller equation
11. Fritzsche equation

We will discuss each of these equations, their limitations, and their applicability to compressible fluids, such as natural gas, using illustrated examples. A comparison of these equations will also be discussed using an example pipeline.

### 2.3 GENERAL FLOW EQUATION

The General Flow equation, also called the Fundamental Flow equation, for the steady-state isothermal flow in a gas pipeline is the basic equation for relating the pressure drop with flow rate. The most common form of this equation in the U.S. Customary System (USCS) of units is given in terms of the pipe diameter, gas properties, pressures, temperatures, and flow rate as follows. Refer to Figure 2.2 for an explanation of symbols used.

$$
\begin{equation*}
Q=77.54\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-P_{2}^{2}}{G T_{f} L Z f}\right)^{0.5} D^{2.5} \quad \text { (USCS units) } \tag{2.2}
\end{equation*}
$$

where
$Q=$ gas flow rate, measured at standard conditions, $\mathrm{ft}^{3} /$ day (SCFD)
$f=$ friction factor, dimensionless
$P_{b}=$ base pressure, psia
$T_{b}=$ base temperature, ${ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)$
$P_{1}=$ upstream pressure, psia
$P_{2}=$ downstream pressure, psia
$G=$ gas gravity (air $=1.00$ )
$T_{f}=$ average gas flowing temperature, ${ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)$
$L=$ pipe segment length, mi
$Z=$ gas compressibility factor at the flowing temperature, dimensionless
$D=$ pipe inside diameter, in.

It must be noted that for the pipe segment from section 1 to section 2, the gas temperature $T_{f}$ is assumed to be constant (isothermal flow).

In SI units, the General Flow equation is stated as follows:

$$
\begin{equation*}
Q=1.1494 \times 10^{-3}\left(\frac{T_{b}}{P_{b}}\right)\left[\frac{\left(P_{1}^{2}-P_{2}^{2}\right)}{G T_{f} L Z f}\right]^{0.5} D^{2.5} \quad \text { (SI units) } \tag{2.3}
\end{equation*}
$$



Figure 2.2 Steady flow in gas pipeline.
where
$Q=$ gas flow rate, measured at standard conditions, $\mathrm{m}^{3} /$ day
$f=$ friction factor, dimensionless
$P_{b}=$ base pressure, kPa
$T_{b}=$ base temperature, $\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$P_{1}=$ upstream pressure, kPa
$P_{2}=$ downstream pressure, kPa
$G=$ gas gravity (air $=1.00$ )
$T_{f}=$ average gas flowing temperature, $\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$L=$ pipe segment length, km
$Z=$ gas compressibility factor at the flowing temperature, dimensionless
$D=$ pipe inside diameter, mm

Due to the nature of Equation 2.3, the pressures can also be in MPa or Bar, as long as the same consistent unit is used.

Equation 2.2 relates the capacity (flow rate or throughput) of a pipe segment of length $L$, based on an upstream pressure of $P_{1}$ and a downstream pressure of $P_{2}$ as shown in Figure 2.2. It is assumed that there is no elevation difference between the upstream and downstream points; therefore, the pipe segment is horizontal.

Upon examining the General Flow Equation 2.2, we see that for a pipe segment of length $L$ and diameter $D$, the gas flow rate $Q$ (at standard conditions) depends on several factors. $Q$ depends on gas properties represented by the gravity $G$ and the compressibility factor $Z$. If the gas gravity is increased (heavier gas), the flow rate will decrease. Similarly, as the compressibility factor $Z$ increases, the flow rate will decrease. Also, as the gas flowing temperature $T_{f}$ increases, throughput will decrease. Thus, the hotter the gas, the lower the flow rate will be. Therefore, to increase the flow rate, it helps to keep the gas temperature low. The impact of pipe length and inside diameter is also clear. As the pipe segment length increases for given pressure $P_{1}$ and $P_{2}$, the flow rate will decrease. On the other hand, the larger the diameter, the larger the flow rate will be. The term $P_{1}{ }^{2}-P_{2}{ }^{2}$ represents the driving force that causes the flow rate from the upstream end to the downstream end. As the downstream pressure $P_{2}$ is reduced, keeping the upstream pressure $P_{1}$ constant, the flow rate will increase. It is obvious that when there is no flow rate, $P_{1}$ is equal to $P_{2}$. It is due to friction between the gas and pipe walls that the pressure drop $\left(P_{1}-P_{2}\right)$ occurs from the upstream point 1 to downstream point 2 . The friction factor $f$ depends on the internal condition of the pipe as well as the type of flow (laminar or turbulent) and will be discussed in detail beginning in Section 2.8.

Sometimes the General Flow equation is represented in terms of the transmission factor $F$ instead of the friction factor $f$. This form of the equation is as follows.

$$
\begin{equation*}
Q=38.77 F\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-P_{2}^{2}}{G T_{f} L Z}\right)^{0.5} D^{2.5} \quad \text { (USCS units) } \tag{2.4}
\end{equation*}
$$

where the transmission factor $F$ and friction factor $f$ are related by

$$
\begin{equation*}
F=\frac{2}{\sqrt{f}} \tag{2.5}
\end{equation*}
$$

and in SI units

$$
\begin{equation*}
Q=5.747 \times 10^{-4} F\left(\frac{T_{b}}{P_{b}}\right)\left[\frac{\left(P_{1}^{2}-P_{2}^{2}\right)}{G T_{f} L Z}\right]^{0.5} D^{2.5} \quad \text { (SI units) } \tag{2.6}
\end{equation*}
$$

We will discuss several aspects of the General Flow equation before moving on to the other formulas for pressure drop calculation.

### 2.4 EFFECT OF PIPE ELEVATIONS

When elevation difference between the ends of a pipe segment is included, the General Flow equation is modified as follows:

$$
\begin{equation*}
Q=38.77 F\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G T_{f} L_{e} Z}\right)^{0.5} D^{2.5} \quad \text { (USCS units) } \tag{2.7}
\end{equation*}
$$

and in SI units

$$
\begin{equation*}
Q=5.747 \times 10^{-4} F\left(\frac{T_{b}}{P_{b}}\right)\left[\frac{\left(P_{1}^{2}-e^{s} P_{2}^{2}\right)}{G T_{f} L_{e} Z}\right]^{0.5} D^{2.5} \quad \text { (SI units) } \tag{2.8}
\end{equation*}
$$

where

$$
\begin{equation*}
L_{e}=\frac{L\left(e^{s}-1\right)}{s} \tag{2.9}
\end{equation*}
$$

The equivalent length, $L_{e}$, and the term $e^{s}$ take into account the elevation difference between the upstream and downstream ends of the pipe segment. The parameter $s$ depends upon the gas gravity, gas compressibility factor, the flowing temperature, and the elevation difference. It is defined as follows in USCS units:

$$
\begin{equation*}
s=0.0375 G\left(\frac{H_{2}-H_{1}}{T_{f} Z}\right) \quad \text { (USCS units) } \tag{2.10}
\end{equation*}
$$

where
$s$ = elevation adjustment parameter, dimensionless
$H_{1}=$ upstream elevation, ft
$H_{2}=$ downstream elevation, ft
$e=$ base of natural logarithms $(e=2.718 \ldots)$
Other symbols are as defined earlier.
In SI units, the elevation adjustment parameter $s$ is defined as follows:

$$
\begin{equation*}
s=0.0684 G\left(\frac{H_{2}-H_{1}}{T_{f} Z}\right) \quad \text { (SI units) } \tag{2.11}
\end{equation*}
$$

where
$H_{1}=$ upstream elevation, m
$H_{2}=$ downstream elevation, m
Other symbols are as defined earlier.
In the calculation of $L_{e}$ in Equation 2.9, we have assumed that there is a single slope between the upstream point 1 and the downstream point 2 in Figure 2.2. If, however, the pipe segment of length $L$ has a series of slopes, then we introduce a parameter $j$ as follows for each individual pipe subsegment that constitutes the pipe length from point 1 to point 2 .

$$
\begin{equation*}
j=\frac{e^{s}-1}{s} \tag{2.12}
\end{equation*}
$$

The parameter $j$ is calculated for each slope of each pipe subsegment of length $L_{1}$, $L_{2}$, etc. that make up the total length $L$. The equivalent length term $L_{e}$ in Equation 2.7 and Equation 2.8 is calculated by summing the individual slopes as defined below.

$$
\begin{equation*}
L_{e}=j_{1} L_{1}+j_{2} L_{2} e^{s 1}+j_{3} L_{3} e^{s 2}+\cdots \tag{2.13}
\end{equation*}
$$

The terms $j_{1}, j_{2}$, etc. for each rise or fall in the elevations of individual pipe subsegments are calculated for the parameters $s_{1}, s_{2}$, etc. for each segment in accordance with Equation 2.12, from the pipeline inlet to the end of each segment.

In the subsequent sections of this chapter, we will discuss how the friction factor and transmission factor are calculated using various equations such as ColebrookWhite and AGA. It is important to note that the General Flow equation is the most commonly used equation to calculate the flow rate and pressure in a gas pipeline. In order to apply it correctly, we must use the correct friction factor or transmission factor. The Colebrook equation, AGA equation, and other empirical equations are used to calculate the friction factor to be used in the General Flow equation. Several other equations, such as Panhandle A, Panhandle B, and Weymouth, calculate the flow rate for a given pressure without using a friction factor or transmission factor. However, an equivalent friction factor (or transmission factor) can be calculated using these methods as well.

### 2.5 AVERAGE PIPE SEGMENT PRESSURE

In the General Flow equation, the compressibility factor $Z$ is used. This must be calculated at the gas flowing temperature and average pressure in the pipe segment. Therefore, it is important to first calculate the average pressure in a pipe segment, described in Figure 2.2.

Consider a pipe segment with upstream pressure $P_{1}$ and downstream pressure $P_{2}$, as in Figure 2.2. An average pressure for this segment must be used to calculate the compressibility factor of gas at the average gas temperature $T_{f}$. As a first approximation, we may use an arithmetic average of $\left(P_{1}+P_{2}\right) / 2$. However, it has been found that a more accurate value of the average gas pressure in a pipe segment is

$$
\begin{equation*}
P_{\mathrm{avg}}=\frac{2}{3}\left(P_{1}+P_{2}-\frac{P_{1} P_{2}}{P_{1}+P_{2}}\right) \tag{2.14}
\end{equation*}
$$

Another form of the average pressure in a pipe segment is

$$
\begin{equation*}
P_{\mathrm{avg}}=\frac{2}{3}\left(\frac{P_{1}^{3}-P_{2}^{3}}{P_{1}^{2}-P_{2}^{2}}\right) \tag{2.15}
\end{equation*}
$$

It must be noted that the pressures used in the General Flow equation are all in absolute units. Therefore, gauge pressure units should be converted to absolute pressure by adding the base pressure.

For example, the upstream and downstream pressures are 1000 psia and 900 psia , respectively. From Equation 2.14, the average pressure is

$$
P_{\text {avg }}=\frac{2}{3}\left(1000+900-\frac{1000 \times 900}{1900}\right)=950.88 \mathrm{psia}
$$

Compare this to the arithmetic average of

$$
P_{\text {avg }}=\frac{1}{2}(1000+900)=950 \mathrm{psia}
$$

### 2.6 VELOCITY OF GAS IN A PIPELINE

The velocity of gas flow in a pipeline represents the speed at which the gas molecules move from one point to another. Unlike a liquid pipeline, due to compressibility, the gas velocity depends upon the pressure and, hence, will vary along the pipeline even if the pipe diameter is constant. The highest velocity will be at the downstream end, where the pressure is the least. Correspondingly, the least velocity will be at the upstream end, where the pressure is higher.

Consider a pipe transporting gas from point A to point B as shown in Figure 2.2. Under steady state flow, at A, the mass flow rate of gas is designated as $M$ and will be the same as the mass flow rate at point B , if between A and B there is no injection or delivery of gas. The mass being the product of volume and density, we can write the following relationship for point A :

$$
\begin{equation*}
M=Q \rho \tag{2.16}
\end{equation*}
$$

The volume rate $Q$ can be expressed in terms of the flow velocity $u$ and pipe cross sectional area $A$ as follows:

$$
\begin{equation*}
Q=u A \tag{2.17}
\end{equation*}
$$

Therefore, combining Equation 2.16 and Equation 2.17 and applying the conservation of mass to points A and B , we get

$$
\begin{equation*}
M_{1}=u_{1} A_{1} \rho_{1}=M_{2}=u_{2} A_{2} \rho_{2} \tag{2.18}
\end{equation*}
$$

where subscripts 1 and 2 refer to points $A$ and $B$, respectively. If the pipe is of uniform cross section between A and B , then $A_{1}=A_{2}=A$.

Therefore, the area term in Equation 2.18 can be dropped, and the velocities at A and B are related by the following equation:

$$
\begin{equation*}
u_{1} \rho_{1}=u_{2} \rho_{2} \tag{2.19}
\end{equation*}
$$

Since the flow of gas in a pipe can result in variation of temperature from point $A$ to point B , the gas density will also vary with temperature and pressure. If the density and velocity at one point are known, the corresponding velocity at the other point can be calculated using Equation 2.19.

If inlet conditions are represented by point A and the volume flow rate $Q$ at standard conditions of $60^{\circ} \mathrm{F}$ and 14.7 psia are known, we can calculate the velocity at any point along the pipeline at which the pressure and temperature of the gas are $P$ and $T$, respectively.

The velocity of gas at section 1 is related to the flow rate $Q_{1}$ at section 1 and pipe cross-sectional area $A$ as follows from Equation 2.17:

$$
Q_{1}=u_{1} A
$$

The mass flow rate $M$ at section 1 and 2 is the same for steady-state flow. Therefore,

$$
\begin{equation*}
M=Q_{1} \rho_{1}=Q_{2} \rho_{2}=Q_{b} \rho_{b} \tag{2.20}
\end{equation*}
$$

where $Q_{b}$ is the gas flow rate at standard conditions and $\rho_{b}$ is the corresponding gas density.

Therefore, simplifying Equation 2.20,

$$
\begin{equation*}
Q_{1}=Q_{b}\left(\frac{\rho_{b}}{\rho_{1}}\right) \tag{2.21}
\end{equation*}
$$

Applying the gas law Equation 1.9, we get

$$
\frac{P_{1}}{\rho_{1}}=Z_{1} R T_{1}
$$

or

$$
\begin{equation*}
\rho_{1}=\frac{P_{1}}{Z_{1} R T_{1}} \tag{2.22}
\end{equation*}
$$

where $P_{1}$ and $T_{1}$ are the pressure and temperature at pipe section 1.
Similarly, at standard conditions,

$$
\begin{equation*}
\rho_{b}=\frac{P_{b}}{Z_{b} R T_{b}} \tag{2.23}
\end{equation*}
$$

From Equation 2.21, Equation 2.22, and Equation 2.23, we get

$$
\begin{equation*}
Q_{1}=Q_{b}\left(\frac{P_{b}}{T_{b}}\right)\left(\frac{T_{1}}{P_{1}}\right)\left(\frac{Z_{1}}{Z_{b}}\right) \tag{2.24}
\end{equation*}
$$

Since $Z_{b}=1.00$, approximately, we can simplify this to

$$
\begin{equation*}
Q_{1}=Q_{b}\left(\frac{P_{b}}{T_{b}}\right)\left(\frac{T_{1}}{P_{1}}\right) Z_{1} \tag{2.25}
\end{equation*}
$$

Therefore, the gas velocity at section 1 is, using Equation 2.17 and Equation 2.25,

$$
u_{1}=\frac{Q_{b} Z_{1}}{A}\left(\frac{P_{b}}{T_{b}}\right)\left(\frac{T_{1}}{P_{1}}\right)=\frac{4 \times 144}{\pi D^{2}} Q_{b} Z_{1}\left(\frac{P_{b}}{T_{b}}\right)\left(\frac{T_{1}}{P_{1}}\right)
$$

or

$$
\begin{equation*}
u_{1}=0.002122\left(\frac{Q_{b}}{D^{2}}\right)\left(\frac{P_{b}}{T_{b}}\right)\left(\frac{Z_{1} T_{1}}{P_{1}}\right) \quad \text { (USCS units) } \tag{2.26}
\end{equation*}
$$

where

$$
u_{1}=\text { upstream gas velocity, } \mathrm{ft} / \mathrm{s}
$$

$Q_{b}=$ gas flow rate, measured at standard conditions, $\mathrm{ft}^{3} /$ day (SCFD)
$D=$ pipe inside diameter, in.
$P_{b}=$ base pressure, psia
$T_{b}=$ base temperature, ${ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)$
$P_{1}=$ upstream pressure, psia
$T_{1}=$ upstream gas temperature, ${ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)$
$Z_{1}=$ gas compressibility factor at upstream conditions, dimensionless

Similarly, the gas velocity at section 2 is given by

$$
\begin{equation*}
u_{2}=0.002122\left(\frac{Q_{b}}{D^{2}}\right)\left(\frac{P_{b}}{T_{b}}\right)\left(\frac{Z_{2} T_{2}}{P_{2}}\right) \quad \text { (USCS units) } \tag{2.27}
\end{equation*}
$$

In general, the gas velocity at any point in a pipeline is given by

$$
\begin{equation*}
u=0.002122\left(\frac{Q_{b}}{D^{2}}\right)\left(\frac{P_{b}}{T_{b}}\right)\left(\frac{Z T}{P}\right) \tag{2.28}
\end{equation*}
$$

In SI units, the gas velocity at any point in a gas pipeline is given by

$$
\begin{equation*}
u=14.7349\left(\frac{Q_{b}}{D^{2}}\right)\left(\frac{P_{b}}{T_{b}}\right)\left(\frac{Z T}{P}\right) \quad \text { (SI units) } \tag{2.29}
\end{equation*}
$$

where
$u=$ gas velocity, $\mathrm{m} / \mathrm{s}$
$Q_{b}=$ gas flow rate, measured at standard conditions, $\mathrm{m}^{3} /$ day
$D=$ pipe inside diameter, mm
$P_{b}=$ base pressure, kPa
$T_{b}=$ base temperature, $\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$P=$ pressure, kPa
$T=$ average gas flowing temperature, $\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$Z=$ gas compressibility factor at the flowing temperature, dimensionless
Since the right-hand side of Equation 2.29 contains ratios of pressures, any consistent unit can be used, such as $\mathrm{kPa}, \mathrm{MPa}$, or Bar.

### 2.7 EROSIONAL VELOCITY

We have seen from the preceding section that the gas velocity is directly related to the flow rate. As flow rate increases, so does the gas velocity. How high can the gas velocity be in a pipeline? As the velocity increases, vibration and noise are evident. In addition, higher velocities will cause erosion of the pipe interior over a long period of time. The upper limit of the gas velocity is usually calculated approximately from the following equation:

$$
\begin{equation*}
u_{\max }=\frac{100}{\sqrt{\rho}} \tag{2.30}
\end{equation*}
$$

where
$u_{\text {max }}=$ maximum or erosional velocity, $\mathrm{ft} / \mathrm{s}$
$\rho=$ gas density at flowing temperature, $\mathrm{lb} / \mathrm{ft}^{3}$

Since the gas density $\rho$ may be expressed in terms of pressure and temperature, using the gas law Equation 1.8, the maximum velocity Equation 2.30 can be rewritten as

$$
\begin{equation*}
u_{\max }=100 \sqrt{\frac{Z R T}{29 G P}} \quad \text { (USCS units) } \tag{2.31}
\end{equation*}
$$

where
$Z=$ compressibility factor of gas, dimensionless
$R=$ gas constant $=10.73 \mathrm{ft}^{3} \mathrm{psia} / \mathrm{lb}-\mathrm{mole} \mathrm{R}$
$T=$ gas temperature, ${ }^{\circ} \mathrm{R}$
$G=$ gas gravity ( air $=1.00$ )
$P=$ gas pressure, psia
Usually, an acceptable operational velocity is $50 \%$ of the above.

## Example 1

A gas pipeline, NPS 20 with 0.500 in . wall thickness, transports natural gas (specific gravity $=0.6$ ) at a flow rate of 250 MMSCFD at an inlet temperature of $60^{\circ} \mathrm{F}$. Assuming isothermal flow, calculate the velocity of gas at the inlet and outlet of the pipe if the inlet pressure is 1000 psig and the outlet pressure is 850 psig . The base pressure and base temperature are 14.7 psia and $60^{\circ} \mathrm{F}$, respectively. Assume compressibility factor $Z=1.00$. What is the erosional velocity for this pipeline based on the above data and a compressibility factor $Z=0.90$ ?

## Solution

If we assume compressibility factor $Z=1.00$, then using Equation 2.26 , the velocity of gas at the inlet pressure of 1000 psig is

$$
u_{1}=0.002122\left(\frac{250 \times 10^{6}}{19.0^{2}}\right)\left(\frac{14.7}{60+460}\right)\left(\frac{60+460}{1014.7}\right)=21.29 \mathrm{ft} / \mathrm{s}
$$

and the gas velocity at the outlet is by proportions

$$
u_{2}=21.29 \times \frac{1014.7}{864.7}=24.98 \mathrm{ft} / \mathrm{s}
$$

The erosional velocity is found for $Z=0.90$, using Equation 2.31,

$$
u_{\max }=100 \sqrt{\frac{0.9 \times 10.73 \times 520}{29 \times 0.6 \times 1014.7}}=53.33 \mathrm{ft} / \mathrm{s}
$$

## Example 2

A gas pipeline, DN 500 with 12 mm wall thickness, transports natural gas (specific gravity $=0.6$ ) at a flow rate of $7.5 \mathrm{Mm}^{3} /$ day at an inlet temperature of $15^{\circ} \mathrm{C}$. Assuming
isothermal flow, calculate the velocity of gas at the inlet and outlet of the pipe if the inlet pressure is 7 MPa and the outlet pressure is 6 MPa . The base pressure and base temperature are 0.1 MPa and $15^{\circ} \mathrm{C}$.

Assume compressibility factor $Z=0.95$.

## Solution

Inside diameter of pipe $D=500-(2 \times 12)=476 \mathrm{~mm}$.
Flow rate at standard conditions $Q_{b}=7.5 \times 10^{6} \mathrm{~m}^{3} / \mathrm{day}$.

Using Equation 2.29, the velocity of gas at the inlet pressure of 7 MPa is

$$
u_{1}=14.7349\left(\frac{7.5 \times 10^{6}}{476^{2}}\right)\left(\frac{0.1}{15+273}\right)\left(\frac{0.95 \times 288}{7.0}\right)=6.62 \mathrm{~m} / \mathrm{s}
$$

and the gas velocity at the outlet is by proportions

$$
u_{2}=6.62 \times \frac{7.0}{6.0}=7.72 \mathrm{~m} / \mathrm{s}
$$

In the preceding Examples 1 and 2, we have assumed the value of compressibility factor $Z$ to the constant. A more accurate solution will be to calculate the value of $Z$ using one of the methods outlined in Chapter 1, such as the CNGA or Standing-Katz method.

For example, if we used the CNGA Equation 1.34, the compressibility factor in Example 1 will be

$$
\begin{aligned}
Z_{1} & =\frac{1}{\left[1+\frac{1000 \times 344400 \times(10)^{1.785 \times 0.6}}{520^{3.825}}\right]} \\
& =0.8578 \text { at an inlet pressure of } 1000 \mathrm{psig} .
\end{aligned}
$$

and

$$
\begin{aligned}
Z_{2} & =\frac{1}{\left[1+\frac{850 \times 344400 \times(10)^{1.785 \times 0.6}}{520^{.825}}\right]} \\
& =0.8765 \text { at an outlet pressure of } 850 \mathrm{psig} .
\end{aligned}
$$

The inlet and outlet gas velocities then will be modified as follows:
Inlet velocity $u_{1}=0.8578 \times 21.29=18.26 \mathrm{ft} / \mathrm{s}$
Outlet velocity $u_{2}=0.8765 \times 24.98=21.90 \mathrm{ft} / \mathrm{s}$

### 2.8 REYNOLDS NUMBER OF FLOW

An important parameter in flow of fluids in a pipe is the nondimensional term Reynolds number. The Reynolds number is used to characterize the type of flow in a pipe, such as laminar, turbulent, or critical flow. It is also used to calculate the friction factor in pipe flow. We will first outline the calculation of the Reynolds number based upon the properties of the gas and pipe diameter and then discuss the range of Reynolds number for the various types of flow and how to calculate the friction factor. The Reynolds number is a function of the gas flow rate, pipe inside diameter, and the gas density and viscosity and is calculated from the following equation:

$$
\begin{equation*}
R e=\frac{u D \rho}{\mu} \quad \text { (USCS units) } \tag{2.32}
\end{equation*}
$$

where
$R e=$ Reynolds number, dimensionless
$u=$ average velocity of gas in pipe, $\mathrm{ft} / \mathrm{s}$
$D=$ inside diameter of pipe, ft
$\rho=$ gas density, $\mathrm{lb} / \mathrm{ft}^{3}$
$\mu=$ gas viscosity, lb/ft-s
The above equation for the Reynolds number is in USCS units. The corresponding equation for the Reynolds number in SI units is as follows:

$$
\begin{equation*}
R e=\frac{u D \rho}{\mu} \quad \text { (SI units) } \tag{2.33}
\end{equation*}
$$

where
$R e=$ Reynolds number, dimensionless
$u=$ average velocity of gas in pipe, $\mathrm{m} / \mathrm{s}$
$D=$ inside diameter of pipe, m
$\rho=$ gas density, $\mathrm{kg} / \mathrm{m}^{3}$
$\mu=$ gas viscosity, $\mathrm{kg} / \mathrm{m}-\mathrm{s}$
In gas pipeline hydraulics, using customary units, a more suitable equation for the Reynolds number is as follows:

$$
\begin{equation*}
R e=0.0004778\left(\frac{P_{b}}{T_{b}}\right)\left(\frac{G Q}{\mu D}\right) \quad \text { (USCS units) } \tag{2.34}
\end{equation*}
$$

where
$P_{b}=$ base pressure, psia
$T_{b}=$ base temperature, ${ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)$
$G=$ specific gravity of gas (air $=1.0$ )
$Q=$ gas flow rate, standard $\mathrm{ft}^{3} /$ day (SCFD)
$D=$ pipe inside diameter, in.
$\mu=$ viscosity of gas, lb/ft-s

In SI units, the Reynolds number is

$$
\begin{equation*}
R e=0.5134\left(\frac{P_{b}}{T_{b}}\right)\left(\frac{G Q}{\mu D}\right) \quad \text { (SI units) } \tag{2.35}
\end{equation*}
$$

where
$P_{b}=$ base pressure, kPa
$T_{b}=$ base temperature, ${ }^{\circ} \mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$G=$ specific gravity of gas (air $=1.0$ )
$Q=$ gas flow rate, $\mathrm{m}^{3} /$ day (standard conditions)
$D=$ pipe inside diameter, mm
$\mu=$ viscosity of gas, Poise
Laminar flow occurs in a pipeline when the Reynolds number is below a value of approximately 2000. Turbulent flow occurs when the Reynolds number is greater than 4000 . For Reynolds numbers between 2000 and 4000, the flow is undefined and is referred to as critical flow.

Thus,
For laminar flow, $R e \leq 2000$
For turbulent flow, $R e>4000$
For critical flow, $R e>2000$ and $R e \leq 4000$
Most natural gas pipelines operate in the turbulent flow region. Therefore, the Reynolds number is greater than 4000. Turbulent flow is further divided into three regions known as smooth pipe flow, fully rough pipe flow, and transition flow. We will discuss these flow regions in more detail in the subsequent sections of this chapter.

## Example 3

A natural gas pipeline, NPS 20 with 0.500 in . wall thickness, transports 100 MMSCFD. The specific gravity of gas is 0.6 and viscosity is $0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{s}$. Calculate the value of the Reynolds number of flow. Assume the base temperature and base pressure are $60^{\circ} \mathrm{F}$ and 14.7 psia, respectively.

## Solution

Pipe inside diameter $=20-2 \times 0.5=19.0$ in.
The base temperature $=60+460=520^{\circ} \mathrm{R}$

Using Equation 2.34, we get

$$
R e=0.0004778\left(\frac{14.7}{520}\right)\left(\frac{0.6 \times 100 \times 10^{6}}{0.000008 \times 19}\right)=5,331,726
$$

Since $R e$ is greater than 4000, the flow is in the turbulent region.

## Example 4

A natural gas pipeline, DN 500 with 12 mm wall thickness, transports $3 \mathrm{Mm}^{3} /$ day . The specific gravity of gas is 0.6 and viscosity is 0.00012 Poise. Calculate the value of the Reynolds number. Assume the base temperature and base pressure are $15^{\circ} \mathrm{C}$ and 101 kPa , respectively.

Solution
Pipe inside diameter $=500-2 \times 12=476 \mathrm{~mm}$
The base temperature $=15+273=288 \mathrm{~K}$
Using Equation 2.35, we get

$$
R e=0.5134\left(\frac{101}{15+273}\right)\left(\frac{0.6 \times 3 \times 10^{6}}{0.00012 \times 476}\right)=5,673,735
$$

Since $R e$ is greater than 4000 , the flow is in the turbulent region.

### 2.9 FRICTION FACTOR

In order to calculate the pressure drop in a pipeline at a given flow rate, we must first understand the concept of friction factor. The term friction factor is a dimensionless parameter that depends upon the Reynolds number of flow. In engineering literature, we find two different friction factors mentioned. The Darcy friction factor is more common and will be used throughout this book. Another friction factor known as the Fanning friction factor is preferred by some engineers. The Fanning friction factor is numerically equal to one-fourth the Darcy friction factor as below.

$$
\begin{equation*}
f_{f}=\frac{f_{d}}{4} \tag{2.36}
\end{equation*}
$$

where
$f_{f}=$ Fanning friction factor
$f_{d}=$ Darcy friction factor
To avoid confusion, in subsequent discussions, the Darcy friction factor is used and will be represented by the symbol $f$. For laminar flow, the friction factor is inversely proportional to the Reynolds number, as indicated below.

$$
\begin{equation*}
f=\frac{64}{R e} \tag{2.37}
\end{equation*}
$$

For turbulent flow, the friction factor is a function of the Reynolds number, pipe inside diameter, and internal roughness of the pipe. Many empirical relationships for


Figure 2.3 Moody diagram.
calculating $f$ have been put forth by researchers. The more popular correlations include the Colebrook-White and AGA equations.

Before we discuss the equations for calculating the friction factor in turbulent flow, it is appropriate to analyze the turbulent flow regime. Turbulent flow in pipes ( $R e>4000$ ) is subdivided into three separate regions as follows:

1. Turbulent flow in smooth pipes
2. Turbulent flow in fully rough pipes
3. Transition flow between smooth pipes and rough pipes

For turbulent flow in smooth pipes, the friction factor $f$ depends only on the Reynolds number. For fully rough pipes, $f$ depends more on the pipe internal roughness and less on the Reynolds number. In the transition zone between smooth pipe flow and flow in fully rough pipes, $f$ depends on the pipe roughness, pipe inside diameter, and the Reynolds number. The various flow regimes are depicted in the Moody diagram, shown in Figure 2.3.

The Moody diagram is a graphic plot of the variation of the friction factor with the Reynolds number for various values of relative pipe roughness. The latter term is simply a dimensionless parameter obtained by dividing the absolute (or internal) pipe roughness by the pipe inside diameter as follows:

$$
\begin{equation*}
\text { Relative roughness }=\frac{e}{D} \tag{2.38}
\end{equation*}
$$

where
$e=$ absolute or internal roughness of pipe, in.
$D=$ pipe inside diameter, in.
The terms absolute pipe roughness and internal pipe roughness are equivalent.
Generally, the internal pipe roughness is expressed in microinches (one-millionth of an inch). For example, an internal roughness of 0.0006 in . is referred to as 600 microinches or $600 \mu \mathrm{in}$. If the pipe inside diameter is 15.5 in ., the relative roughness is, in this case,

$$
\text { Relative roughness }=\frac{0.0006}{15.5}=0.0000387=3.87 \times 10^{-5}
$$

For example, from the Moody diagram in Figure 2.3, for $R e=10$ million and $e / D=0.0001$, we find that $f=0.012$.

### 2.10 COLEBROOK-WHITE EQUATION

The Colebrook-White equation, sometimes referred to simply as the Colebrook equation, is a relationship between the friction factor and the Reynolds number, pipe roughness, and inside diameter of pipe. The following form of the Colebrook
equation is used to calculate the friction factor in gas pipelines in turbulent flow.

$$
\begin{equation*}
\frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{e}{3.7 D}+\frac{2.51}{\operatorname{Re} \sqrt{f}}\right) \quad \text { for } R e>4000 \tag{2.39}
\end{equation*}
$$

where
$f=$ friction factor, dimensionless
$D=$ pipe inside diameter, in.
$e=$ absolute pipe roughness, in.
$R e=$ Reynolds number of flow, dimensionless
Since $R e$ and $f$ are dimensionless, as long as consistent units are used for both $e$ and $D$, the Colebrook equation is the same regardless of the units employed. Therefore, in SI units, Equation 2.39 is used with $e$ and $D$ expressed in mm.

It can be seen from Equation 2.39 that in order to calculate the friction factor $f$, we must use a trial-and-error approach. It is an implicit equation in $f$, since $f$ appears on both sides of the equation. We first assume a value of $f$ (such as 0.01 ) and substitute it in the right-hand side of the equation. This will yield a second approximation for $f$, which can then be used to calculate a better value of $f$, and so on. Generally 3 to 4 iterations are sufficient to converge on a reasonably good value of the friction factor.

It can be seen from the Colebrook Equation 2.39, for turbulent flow in smooth pipes, the first term within the square brackets is negligible compared to the second term, since pipe roughness $e$ is very small.

Therefore, for smooth pipe flow, the friction factor equation reduces to

$$
\begin{equation*}
\frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{2.51}{\operatorname{Re} \sqrt{f}}\right), \quad \text { for turbulent flow in smooth pipes } \tag{2.40}
\end{equation*}
$$

Similarly, for turbulent flow in fully rough pipes, with $R e$ being a large number, $f$ depends mostly on the roughness $e$ and, therefore, the friction factor equation reduces to

$$
\begin{equation*}
\frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{e}{3.7 D}\right), \quad \text { for turbulent flow in fully rough pipes } \tag{2.41}
\end{equation*}
$$

Table 2.1 lists typical values of pipe internal roughness used to calculate the friction factor.

As an example, if $R e=100$ million or larger and $e / D=0.0002$, the friction factor from Equation 2.41 is

$$
\frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{0.0002}{3.7}\right)
$$

or $f=0.0137$, which correlates well with the friction factor obtained from the Moody diagram in Figure 2.3.

Table 2.1 Pipe Internal Roughness

| Pipe Material | Roughness, <br> in. | Roughness, <br> mm |
| :--- | :---: | :---: |
| Riveted steel | 0.0354 to 0.354 | 0.9 to 9.0 |
| Commercial steel/welded steel | 0.0018 | 0.045 |
| Cast iron | 0.0102 | 0.26 |
| Galvanized iron | 0.0059 | 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 to 0.118 | 0.3 to 3.0 |

## Example 5

A natural gas pipeline, NPS 20 with 0.500 in . wall thickness, transports 200 MMSCFD. The specific gravity of gas is 0.6 and viscosity is $0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{s}$. Calculate the friction factor using the Colebrook equation. Assume absolute pipe roughness $=600 \mu \mathrm{in}$.

The base temperature and base pressure are $60^{\circ} \mathrm{F}$ and 14.7 psia , respectively.

## Solution

Pipe inside diameter $=20-2 \times 0.5=19.0 \mathrm{in}$.
Absolute pipe roughness $=600 \mu$ in. $=0.0006$ in.
First, we calculate the Reynolds number using Equation 2.34:

$$
R e=0.0004778\left(\frac{14.7}{60+460}\right)\left(\frac{0.6 \times 200 \times 10^{6}}{0.000008 \times 19}\right)=10,663,452
$$

Using Equation 2.39,

$$
\frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{0.0006}{3.7 \times 19}+\frac{2.51}{10,663,452 \sqrt{f}}\right)
$$

This equation will be solved by successive iteration.
Assume $f=0.01$ initially; substituting above, we get a better approximation as $f=$ 0.0101 . Repeating the iteration, we get the final value as $f=0.0101$. Therefore, the friction factor is 0.0101 .

## Example 6

A natural gas pipeline, DN 500 with 12 mm wall thickness, transports $6 \mathrm{Mm}^{3} /$ day. The specific gravity of gas is 0.6 and viscosity is 0.00012 Poise. Calculate the
friction factor using the Colebrook equation. Assume absolute pipe roughness $=$ 0.03 mm and assume the base temperature and base pressure are $15^{\circ} \mathrm{C}$ and 101 kPa , respectively.

Solution
Pipe inside diameter $=500-2 \times 12=476 \mathrm{~mm}$
First, we calculate the Reynolds number using Equation 2.35:

$$
R e=0.5134\left(\frac{101}{15+273}\right)\left(\frac{0.6 \times 6 \times 10^{6}}{0.00012 \times 476}\right)=11,347,470
$$

Using Equation 2.39, the friction factor is

$$
\frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{0.030}{3.7 \times 476}+\frac{2.51}{11,347,470 \sqrt{f}}\right)
$$

This equation will be solved by successive iteration.
Assume $f=0.01$ initially; substituting above, we get a better approximation as $f=0.0112$. Repeating the iteration, we get the final value as $f=0.0112$. Therefore, the friction factor is 0.0112 .

### 2.11 TRANSMISSION FACTOR

The transmission factor $F$ is considered the opposite of the friction factor $f$. Whereas the friction factor indicates how difficult it is to move a certain quantity of gas through a pipeline, the transmission factor is a direct measure of how much gas can be transported through the pipeline. As the friction factor increases, the transmission factor decreases and, therefore, the gas flow rate also decreases. Conversely, the higher the transmission factor, the lower the friction factor and, therefore, the higher the flow rate will be.

The transmission factor $F$ is related to the friction factor $f$ as follows:

$$
\begin{equation*}
F=\frac{2}{\sqrt{f}} \tag{2.42}
\end{equation*}
$$

Therefore,

$$
\begin{equation*}
f=\frac{4}{F^{2}} \tag{2.43}
\end{equation*}
$$

where
$f=$ friction factor
$F=$ transmission factor
It must be noted that the friction factor $f$ in the above equation is the Darcy friction factor. Since some engineers prefer to use the Fanning friction factor, the relationship between the transmission factor $F$ and the Fanning friction factor is given below for reference.

$$
\begin{equation*}
F=\frac{1}{\sqrt{f_{f}}} \tag{2.44}
\end{equation*}
$$

where $f_{f}$ is the Fanning friction factor.
For example, if the Darcy friction factor is 0.025 , the transmission factor is, using Equation 2.42,

$$
F=\frac{2}{\sqrt{0.025}}=12.65
$$

The Fanning friction factor in this case will be $\frac{0.025}{4}=0.00625$. Therefore, the transmission factor using Equation 2.44 is $F=\frac{1}{\sqrt{0.00625}}=12.65$, which is the same as calculated using the Darcy friction factor.

Thus, it must be noted that there is only one transmission factor, whereas there are two different friction factors.

Having defined a transmission factor, we can rewrite the Colebrook Equation 2.39 in terms of the transmission factor using Equation 2.42 as follows:

$$
\begin{equation*}
F=-4 \log _{10}\left(\frac{e}{3.7 D}+\frac{1.255 F}{R e}\right) \tag{2.45}
\end{equation*}
$$

Since $R e$ and $F$ are dimensionless, as long as consistent units are used for both $e$ and $D$, the transmission factor equation is the same regardless of the units employed. Therefore, in SI units, Equation 2.45 is used with $e$ and $D$ expressed in mm.

Similar to the calculation of the friction factor $f$ from Equation 2.39, to calculate the transmission factor $F$ from Equation 2.45, an iterative approach must be used. This will be illustrated using an example.

## Example 7

For a gas pipeline, flowing 100 MMSCFD gas of specific gravity 0.6 and viscosity of $0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{s}$, calculate the friction factor and transmission factor considering an NPS 20 pipeline, $0.500-\mathrm{in}$. wall thickness, and an internal roughness of 600 microinches. Assume the base temperature and base pressure are $60^{\circ} \mathrm{F}$ and 14.7 psia , respectively. If the flow rate increases by $50 \%$, what is the impact on the friction factor and transmission factor?

Solution

The base temperature $=60+460=520^{\circ} \mathrm{R}$
Pipe inside diameter $=20-2 \times 0.500=19.0$ in.

Using Equation 2.34, we calculate the Reynolds number as

$$
\operatorname{Re}=0.0004778\left(\frac{14.7}{520}\right)\left(\frac{0.6 \times 100 \times 10^{6}}{0.000008 \times 19}\right)=5,331,726
$$

The relative roughness $=\frac{600 \times 10^{-6}}{19}=0.0000316$
Using Equation 2.39, the friction factor is

$$
\frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{0.0000316}{3.7}+\frac{2.51}{5,331,726 \sqrt{f}}\right)
$$

Solving by successive iteration, we get

$$
f=0.0105
$$

Therefore, the transmission factor $F$ is found from Equation 2.42 as follows:

$$
F=\frac{2}{\sqrt{0.0105}}=19.53
$$

It must be noted that the friction factor calculated above is the Darcy friction factor. The corresponding Fanning friction factor will be one-fourth the calculated value.

When flow rate is increased by $50 \%$, the Reynolds number becomes, by proportion,

$$
\operatorname{Re}=1.5 \times 5,331,726=7,997,589
$$

The new friction factor from Equation 2.39 is

$$
\frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{0.0000316}{3.7}+\frac{2.51}{7,997,589 \sqrt{f}}\right)
$$

Solving for $f$ by successive iteration, we get

$$
f=0.0103
$$

The corresponding transmission factor is

$$
F=\frac{2}{\sqrt{0.0103}}=19.74
$$

Compared to the previous values of 0.0105 for the friction factor and 19.53 for the transmission factor, we see the following changes:

$$
\begin{gathered}
\text { Decrease in friction factor }=\frac{0.0105-0.0103}{0.0105}=0.019 \text { or } 1.9 \% \\
\text { Increase in transmission factor }=\frac{19.74-19.53}{19.53}=0.0108 \text { or } 1.08 \%
\end{gathered}
$$

Thus, increasing the flow rate by $50 \%$ reduces the friction factor by $1.9 \%$ and increases the transmission factor by $1.08 \%$.

## Example 8

For a gas pipeline, flowing $3 \mathrm{Mm}^{3} /$ day gas of specific gravity 0.6 and viscosity of 0.000119 Poise, calculate the friction factor and transmission factor considering a DN 400 pipeline, 10 mm wall thickness, and an internal roughness of 0.02 mm . The base temperature and base pressure are $15^{\circ} \mathrm{C}$ and 101 kPa , respectively. If the flow rate is doubled, what is the impact on the friction factor and transmission factor?

## Solution

The base temperature $=15+273=288 \mathrm{~K}$

Pipe inside diameter $=400-2 \times 10=380 \mathrm{~mm}$
Using Equation 2.35, we calculate the Reynolds number as

$$
R e=0.5134\left(\frac{101}{288}\right)\left(\frac{0.6 \times 3 \times 10^{6}}{0.000119 \times 380}\right)=7,166,823
$$

The relative roughness $=\frac{0.02}{380}=0.0000526$
Using Equation 2.39, the friction factor is

$$
\frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{0.0000526}{3.7}+\frac{2.51}{7,166,823 \sqrt{f}}\right)
$$

Solving by iteration, we get

$$
f=0.0111
$$

Therefore, the transmission factor $F$ is found from Equation 2.42 as follows:

$$
F=\frac{2}{\sqrt{0.0111}}=18.98
$$

It must be noted that the friction factor calculated above is the Darcy friction factor. The corresponding Fanning friction factor will be one-fourth the calculated value.

When the flow rate is doubled, the Reynolds number becomes

$$
R e=2 \times 7,166,823=14,333,646
$$

The new value of the friction factor from Equation 2.39 is

$$
\frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{0.0000526}{3.7}+\frac{2.51}{14,333,646 \sqrt{f}}\right)
$$

Solving for $f$ by successive iteration, we get

$$
f=0.0109
$$

and the transmission factor is

$$
F=\frac{2}{\sqrt{0.0109}}=19.16
$$

Therefore, doubling the flow rate increases the transmission factor and decreases the friction factor as follows:

$$
\begin{gathered}
\text { Decrease in friction factor }=\frac{0.0111-0.0109}{0.0111}=0.018 \text { or } 1.8 \% \\
\text { Increase in transmission factor }=\frac{19.16-18.98}{18.98}=0.0095 \text { or } 0.95 \%
\end{gathered}
$$

### 2.12 MODIFIED COLEBROOK-WHITE EQUATION

The Colebrook-White equation discussed in the preceding section has been in use for many years in both liquid flow and gas flow. The U.S. Bureau of Mines, in 1956, published a report that introduced a modified form of the Colebrook-White equation. The modification results in a higher friction factor and, hence, a smaller value of the transmission factor. Because of this, a conservative value of flow rate is obtained due to the higher friction and pressure drop. The modified version of the ColebrookWhite equation for turbulent flow is as follows:

$$
\begin{equation*}
\frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{e}{3.7 D}+\frac{2.825}{\operatorname{Re} \sqrt{f}}\right) \tag{2.46}
\end{equation*}
$$

Rewriting Equation 2.46 in terms of the transmission factor, we get the following version of the modified Colebrook-White equation:

$$
\begin{equation*}
F=-4 \log _{10}\left(\frac{e}{3.7 D}+\frac{1.4125 F}{R e}\right) \quad \text { (USCS and SI units) } \tag{2.47}
\end{equation*}
$$

Since $R e, f$, and $F$ are dimensionless, as long as consistent units are used for both $e$ and $D$, the modified Colebrook equation is the same, regardless of the units employed. Therefore, in SI units, Equation 2.46 and Equation 2.47 are used with $e$ and $D$ expressed in mm.

Upon comparing Equation 2.39 with Equation 2.46, it is seen that the difference between the Colebrook equation and the modified Colebrook equation lies in the second constant term within the square brackets. The constant 2.51 in Equation 2.39 is replaced with the constant 2.825 in Equation 2.46. Similarly, in the transmission factor equations, the modified equation has 1.4125 instead of 1.255 in the original Colebrook-White equation. Many commercial hydraulic simulation programs list both Colebrook-White equations. Some use only the original Colebrook-White equation.

## Example 9

For a gas pipeline, flowing 100 MMSCFD gas of specific gravity 0.6 and viscosity of $0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{s}$, calculate, using the modified Colebrook-White equation, the friction factor and transmission factor assuming an NPS 20 pipeline, 0.500 in . wall thickness, and an internal roughness of $600 \mu \mathrm{in}$. The base temperature and base pressure are $60^{\circ} \mathrm{F}$ and 14.7 psia, respectively.

How do these numbers compare with those calculated, using the original Colebrook equation?

Solution

The base temperature $=60+460=520^{\circ} \mathrm{R}$
Pipe inside diameter $=20-2 \times 0.500=19.0 \mathrm{in}$.
Using Equation 2.34, we calculate the Reynolds number as

$$
R e=0.0004778\left(\frac{14.7}{520}\right)\left(\frac{0.6 \times 100 \times 10^{6}}{0.000008 \times 19}\right)=5,331,726
$$

The relative roughness is

$$
\frac{e}{D}=\frac{600 \times 10^{6}}{19}=3.16 \times 10^{-5}
$$

From Equation 2.46, the friction factor using the modified Colebrook equation is

$$
\frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{0.0000316}{3.7}+\frac{2.825}{5,331,726 \sqrt{f}}\right)
$$

Solving by successive iteration, we get

$$
f=0.0106
$$

Therefore, the transmission factor $F$ is found from Equation 2.42 as follows:

$$
F=\frac{2}{\sqrt{0.0106}}=19.43
$$

By comparing these results with the friction factor and the transmission factor calculated in Example 7 using the unmodified Colebrook equation, it can be seen that the modified friction factor is approximately $0.95 \%$ higher than that calculated using the original Colebrook-White equation, whereas the transmission factor is approximately $0.51 \%$ lower than that calculated using the original Colebrook-White equation.

## Example 10

A gas pipeline, NPS 20 with 0.500 in. wall thickness, flows 200 MMSCFD gas of specific gravity 0.6 and viscosity of $0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{s}$. Using the modified ColebrookWhite equation, calculate the pressure drop in a 50 mi segment of pipe, based on an upstream pressure of 1000 psig. Assume an internal pipe roughness of $600 \mu \mathrm{in}$. and the base temperature and base pressure of $60^{\circ} \mathrm{F}$ and 14.73 psia , respectively. Neglect elevation effects and use $60^{\circ} \mathrm{F}$ for gas flowing temperature and compressibility factor $Z=0.88$.

## Solution

Inside diameter of pipe $=20-2 \times 0.5=19.0 \mathrm{in}$.

The base temperature $=60+460=520^{\circ} \mathrm{R}$

Gas flow temperature $=60+460=520^{\circ} \mathrm{R}$

First, we calculate the Reynolds number using Equation 2.34.

$$
R e=0.0004778\left(\frac{14.73}{520}\right)\left(\frac{0.6 \times 200 \times 10^{6}}{0.000008 \times 19}\right)=10,685,214
$$

The transmission factor $F$ is calculated from Equation 2.47 as follows:

$$
F=-4 \log _{10}\left(\frac{600 \times 10^{-6}}{3.7 \times 19}+\frac{1.4125 F}{10,685,214}\right)
$$

Solving for $F$ by successive iteration,

$$
F=19.81
$$

Next, using General Flow Equation 2.4, we calculate the downstream pressure $P_{2}$ as follows:

$$
200 \times 10^{6}=38.77 \times 19.81\left(\frac{60+460}{14.73}\right)\left[\frac{1014.73^{2}-P_{2}^{2}}{0.6 \times 520 \times 50 \times 0.88}\right]^{0.5} \times 19^{2.5}
$$

Solving for $P_{2}$, we get

$$
P_{2}=853.23 \mathrm{psia}=838.5 \mathrm{psig}
$$

Therefore, the pressure drop $=1014.73-853.23=161.5 \mathrm{psi}$.

### 2.13 AMERICAN GAS ASSOCIATION (AGA) EQUATION

In 1964 and 1965, the American Gas Association (AGA) published a report on how to calculate the transmission factor for gas pipelines to be used in the General Flow equation. This is sometimes referred to as the AGA NB-13 method. Using the method outlined in this report, the transmission factor $F$ is calculated using two different equations. First, $F$ is calculated for the rough pipe law (referred to as the fully turbulent zone). Next, $F$ is calculated based on the smooth pipe law (referred to as the partially turbulent zone). Finally, the smaller of the two values of the transmission factor is used in the General Flow Equation 2.4 for pressure drop calculation. Even though the AGA method uses the transmission factor $F$ instead of the friction factor $f$, we can still calculate the friction factor using the relationship shown in Equation 2.42.

For the fully turbulent zone, AGA recommends using the following formula for $F$, based on relative roughness $e / D$ and independent of the Reynolds number:

$$
\begin{equation*}
F=4 \log _{10}\left(\frac{3.7 D}{e}\right) \tag{2.48}
\end{equation*}
$$

Equation 2.48 is also known as the Von Karman rough pipe flow equation.
For the partially turbulent zone, $F$ is calculated from the following equations using the Reynolds number, a parameter $D_{f}$ known as the pipe drag factor, and the Von Karman smooth pipe transmission factor $F_{t}$ :

$$
\begin{equation*}
F=4 D_{f} \log _{10}\left(\frac{R e}{1.4125 F_{t}}\right) \tag{2.49}
\end{equation*}
$$

and

$$
\begin{equation*}
F_{t}=4 \log _{10}\left(\frac{R e}{F_{t}}\right)-0.6 \tag{2.50}
\end{equation*}
$$

where
$F_{t}=$ Von Karman smooth pipe transmission factor
$D_{f}=$ pipe drag factor that depends on the Bend Index (BI) of the pipe
The pipe drag factor $D_{f}$ is a parameter that takes into account the number of bends, fittings, etc. Its value ranges from 0.90 to 0.99 . The Bend index is the sum of all the angles and bends in the pipe segment, divided by the total length of the pipe section under consideration.

$$
\begin{equation*}
B I=\frac{\text { total degrees of all bends in pipe section }}{\text { total length of pipe section }} \tag{2.51}
\end{equation*}
$$

Table 2.2 Bend Index and Drag Factor

|  | Bend Index |  |  |
| :--- | :---: | :---: | :---: |
|  | Extremely Low <br> $\mathbf{5}^{\circ}$ to $\mathbf{1 0}$ | Average <br> $\mathbf{6 0}$ |  |
|  | to $\mathbf{8 0}$ |  |  | | Extremely High |
| :---: |
| $\mathbf{2 0 0}^{\circ}$ to $\mathbf{3 0 0}$ |

Note: The drag factors above are based on 40 -ft joints of pipelines and mainline valves at 10-mile spacing.

The value of $D_{f}$ is generally chosen from Table 2.2.
For further discussion on the bend index and drag factor, the reader is referred to Steady Flow in Gas Pipelines listed in the Reference section.

## Example 11

Using the AGA method, calculate the transmission factor and friction factor for gas flow in an NPS 20 pipeline with 0.500 in. wall thickness. The flow rate is 200 MMSCFD, gas gravity $=0.6$, and viscosity $=0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{sec}$. The absolute pipe roughness is $700 \mu \mathrm{in}$. Assume a bend index of $60^{\circ}$, base pressure of 14.73 psia , and base temperature of $60^{\circ} \mathrm{F}$.

## Solution

Inside diameter of pipe $=20-2 \times 0.5=19.0 \mathrm{in}$.
The base temperature $=60+460=520^{\circ} \mathrm{R}$
We will first calculate the Reynolds number using Equation 2.34.

$$
\operatorname{Re}=\frac{0.0004778 \times 200 \times 10^{6} \times 0.6 \times 14.73}{19 \times 0.000008 \times 520}=10,685,214
$$

Next, calculate the two transmission factors.

The fully turbulent transmission factor, using Equation 2.48, is

$$
F=4 \log _{10}\left(\frac{3.7 \times 19}{0.0007}\right)=20.01
$$

For the smooth pipe zone, using Equation 2.50, the Von Karman transmission factor is

$$
F_{t}=4 \log _{10}\left(\frac{10,685,214}{F_{t}}\right)-0.6
$$

Solving this equation by trial and error, we get $F_{t}=22.13$.

From Table 2.2, for a bend index of $60^{\circ}$, the drag factor $D_{f}$ is 0.96 .

Therefore, for the partially turbulent flow zone, using Equation 2.49, the transmission factor is

$$
F=4 \times 0.96 \log _{10}\left(\frac{10,685,214}{1.4125 \times 22.13}\right)=21.25
$$

From the above two values of $F$, using the smaller number, we get the AGA transmission factor as

$$
F=20.01
$$

Therefore, the corresponding friction factor $f$ is found from Equation 2.42 as

$$
\frac{2}{\sqrt{f}}=20.01
$$

or

$$
f=0.0100
$$

## Example 12

Using the AGA method, calculate the transmission factor and friction factor for gas flow in a DN 500 pipeline with 12 mm wall thickness. The flow rate is $6 \mathrm{Mm}^{3} / \mathrm{day}$, gas gravity $=0.6$, and viscosity $=0.00012$ Poise. The absolute pipe roughness is 0.02 mm . Assume a bend index of $60^{\circ}$, base pressure of 101 kPa , and base temperature of $15^{\circ} \mathrm{C}$. For a 60 km pipe length, calculate the upstream pressure needed to hold a downstream pressure of 5 MPa (absolute). Assume flow temperature $=20^{\circ} \mathrm{C}$ and compressibility factor $Z=0.85$. Neglect elevation effects.

## Solution

Inside diameter of pipe $=500-2 \times 12=476 \mathrm{~mm}$

The base temperature $=15+273=288 \mathrm{~K}$

Gas flowing temperature $=20+273=293 \mathrm{~K}$

We first calculate the Reynolds number from Equation 2.35.

$$
R e=0.5134\left(\frac{101}{288}\right)\left(\frac{0.6 \times 6 \times 10^{6}}{0.00012 \times 476}\right)=11,347,470
$$

Next, calculate the two transmission factors as follows:

The fully turbulent transmission factor, using Equation 2.48, is

$$
F=4 \log _{10}\left(\frac{3.7 \times 476}{0.02}\right)=19.78
$$

For the smooth pipe zone, using Equation 2.50, the Von Karman transmission factor is

$$
F_{t}=4 \log _{10}\left(\frac{11,347,470}{F_{t}}\right)-0.6
$$

Solving by successive iteration, we get

$$
F_{t}=22.23
$$

From Table 2.2, for a bend index of $60^{\circ}$, the drag factor is 0.96 .
Therefore, for the partially turbulent flow zone, using Equation 2.49, the transmission factor is

$$
F=4 \times 0.96 \log _{10}\left(\frac{11,347,470}{1.4125 \times 22.23}\right)=21.34
$$

Using the smaller of the two values of $F$, the AGA transmission factor is

$$
F=19.78
$$

Therefore, the corresponding friction factor is found from Equation 2.42 as

$$
\frac{2}{\sqrt{f}}=19.78
$$

or

$$
f=0.0102
$$

Using the General Flow Equation 2.8, we calculate the upstream pressure $P_{1}$ as follows:

$$
6 \times 10^{6}=5.747 \times 10^{-4} \times 19.78 \times\left(\frac{288}{101}\right)\left[\frac{P_{1}^{2}-5000^{2}}{0.6 \times 293 \times 60 \times 0.85}\right]^{0.5} \times 476^{2.5}
$$

Solving for $P_{1}$, we get

$$
P_{1}=6130 \mathrm{kPa}=6.13 \mathrm{MPa}
$$

### 2.14 WEYMOUTH EQUATION

The Weymouth equation is used for high pressure, high flow rate, and large diameter gas gathering systems. This formula directly calculates the flow rate through a pipeline for given values of gas gravity, compressibility, inlet and outlet pressures, pipe diameter, and length. In USCS units, the Weymouth equation is stated as follows:

$$
\begin{equation*}
Q=433.5 E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G T_{f} L_{e} Z}\right)^{0.5} D^{2.667} \tag{2.52}
\end{equation*}
$$

where
$Q=$ volume flow rate, standard $\mathrm{ft}^{3} /$ day (SCFD)
$E=$ pipeline efficiency, a decimal value less than or equal to 1.0
$P_{b}=$ base pressure, psia
$T_{b}=$ base temperature, ${ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)$
$P_{1}=$ upstream pressure, psia
$P_{2}=$ downstream pressure, psia
$G=$ gas gravity (air = 1.00)
$T_{f}=$ average gas flow temperature, ${ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)$
$L_{e}=$ equivalent length of pipe segment, mi
$Z=$ gas compressibility factor, dimensionless
$D=$ pipe inside diameter, in.
where the equivalent length $L_{e}$ and $s$ were defined earlier in Equation 2.9 and Equation 2.10.

By comparing the Weymouth equation with the General Flow equation, we can isolate an equivalent transmission factor as follows:

The Weymouth transmission factor in USCS units is

$$
\begin{equation*}
F=11.18(D)^{1 / 6} \tag{2.53}
\end{equation*}
$$

In SI units, the Weymouth equation is as follows:

$$
\begin{equation*}
Q=3.7435 \times 10^{-3} E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G T_{f} L_{e} Z}\right)^{0.5} D^{2.667} \tag{2.54}
\end{equation*}
$$

where
$Q=$ gas flow rate, standard $\mathrm{m}^{3} /$ day
$T_{b}=$ base temperature, $\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$P_{b}=$ base pressure, kPa
$T_{f}=$ average gas flow temperature, $\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$P_{1}=$ upstream pressure, kPa
$P_{2}=$ downstream pressure, kPa
$L_{e}=$ equivalent length of pipe segment, km

Other symbols are as defined previously.

The Weymouth transmission factor in SI units is

$$
\begin{equation*}
F=6.521(D)^{1 / 6} \tag{2.53a}
\end{equation*}
$$

You will notice that a pipeline efficiency factor, $E$, is used in the Weymouth equation so we can compare the throughput performance of a pipeline using the General Flow equation that does not include an efficiency factor.

## Example 13

Calculate the flow rate using the Weymouth equation in a gas pipeline system, 15 miles long, NPS 12 pipe with 0.250 in . wall thickness, at an efficiency of 0.95 . The upstream pressure is 1200 psia , and the delivery pressure required at the end of the pipe segment is 750 psia . Use gas gravity $=0.59$ and viscosity $=0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{sec}$. The flowing temperature of gas $=75^{\circ} \mathrm{F}$, base pressure $=14.7 \mathrm{psia}$, and base temperature $=60^{\circ} \mathrm{F}$. Assume compressibility factor to be 0.94 .

Neglect elevation difference along the pipe. How does this compare with the flow rate calculated using the General Flow equation with the Colebrook friction factor? Assume a pipe roughness of $700 \mu \mathrm{in}$.

## Solution

Using Equation 2.52, we get the flow rate for the Weymouth equation as follows:

$$
\begin{gathered}
Q=433.5 \times 0.95\left(\frac{60+460}{14.7}\right)\left[\frac{1200^{2}-750^{2}}{0.59 \times(75+460) \times 15 \times 0.94}\right]^{0.5} \times 12.25^{2.667} \\
Q=163,255,858 \mathrm{SCFD}
\end{gathered}
$$

or

$$
Q=163.26 \text { MMSCFD }
$$

Next, we will calculate the Reynolds number using Equation 2.34.

$$
R e=\frac{0.0004778 \times Q \times 0.59 \times 14.7}{12.25 \times 0.000008 \times 520}
$$

where $Q$ is the flow rate in SCFD.
Simplifying, we get $R e=0.0813 Q$.
Since $Q$ is unknown, we will first assume a transmission factor $F=20$ and calculate the flow rate from the General Flow Equation 2.4.

$$
Q=38.77 \times 20\left(\frac{520}{14.7}\right)\left[\frac{1200^{2}-750^{2}}{0.59 \times 535 \times 15 \times 0.94}\right]^{0.5} \times 12.25^{2.5}=202,284,747 \mathrm{SCFD}
$$

or

$$
Q=202.28 \mathrm{MMSCFD}
$$

Next, we will calculate the Reynolds number and the transmission factor based on this flow rate as

$$
R e=0.0813 \times 202,284,747=16.45 \text { million }
$$

and, using Equation 2.45,

$$
F=-4 \log _{10}\left(\frac{700 \times 10^{-6}}{3.7 \times 12.25}+\frac{1.255 F}{16.45 \times 10^{6}}\right)
$$

Solving for $F$, we get

$$
F=19.09
$$

Using this value, the revised flow rate is found by proportion as

$$
Q=202.28 \times \frac{19.09}{20}=193.08 \mathrm{MMSCFD}
$$

Repeating the calculation of $R e$ and $F$, we get

$$
R e=16.45 \times \frac{193.08}{202.28}=15.7 \text { million }
$$

and

$$
F=-4 \log _{10}\left(\frac{700 \times 10^{-6}}{3.7 \times 12.25}+\frac{1.255 F}{15.7 \times 10^{6}}\right)
$$

Therefore, $F=19.08$.

This is fairly close to the previous value of $F=19.09$; therefore, we will use this value and calculate the flow rate as

$$
Q=202.28 \times \frac{19.08}{20}=192.98 \mathrm{MMSCFD}
$$

Comparing this result using the General Flow equation with that calculated using the Weymouth equation, we see that the latter equation is quite conservative.

## Example 14

A natural gas transmission line transports 30 million $\mathrm{m}^{3} /$ day of gas from a processing plant to a compressor station site 100 km away. The pipeline can be assumed to be along a flat terrain. Calculate the minimum pipe diameter required such that the maximum pipe operating pressure is limited to 8500 kPa . The delivery pressure desired at the end of the pipeline is a minimum of 5500 kPa . Assume a pipeline efficiency of 0.95 . The gas gravity is 0.65 , and the gas temperature is $18^{\circ} \mathrm{C}$. Use the Weymouth equation, considering a base temperature $=15^{\circ} \mathrm{C}$ and base pressure 101 kPa . The gas compressibility factor $Z=0.92$.

Solution

The base temperature $=15+273=288 \mathrm{~K}$
The gas flowing temperature $=18+273=291 \mathrm{~K}$

We will assume that given pressures are absolute values.
Upstream pressure $=8500 \mathrm{kPa}$ (absolute)

Downstream pressure $=5500 \mathrm{kPa}$ (absolute)
Using the Weymouth Equation 2.52 and substituting given values, we get

$$
30 \times 10^{6}=3.7435 \times 10^{-3} \times 0.95 \times\left(\frac{288}{101}\right)\left[\frac{8500^{2}-5500^{2}}{0.65 \times 291 \times 100 \times 0.92}\right]^{0.5} \times D^{2.667}
$$

Solving for diameter, $D$, we get

$$
D=826.1 \mathrm{~mm}
$$

Therefore, the minimum diameter required will be DN 850 with 10 mm wall thickness.

### 2.15 PANHANDLE A EQUATION

The Panhandle A Equation was developed for use in natural gas pipelines, incorporating an efficiency factor for Reynolds numbers in the range of 5 to 11 million. In this equation, the pipe roughness is not used. The general form of the Panhandle A equation is expressed in USCS units as follows:

$$
\begin{equation*}
Q=435.87 E\left(\frac{T_{b}}{P_{b}}\right)^{1.0788}\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G^{0.8539} T_{f} L_{e} Z}\right)^{0.5394} D^{2.6182} \quad \text { (USCS units) } \tag{2.55}
\end{equation*}
$$

where
$Q=$ volume flow rate, standard $\mathrm{ft}^{3} /$ day (SCFD)
$E=$ pipeline efficiency, a decimal value less than 1.0
$P_{b}=$ base pressure, psia
$T_{b}=$ base temperature, ${ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)$
$P_{1}=$ upstream pressure, psia
$P_{2}=$ downstream pressure, psia
$G=$ gas gravity (air = 1.00)
$T_{f}=$ average gas flow temperature, ${ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)$
$L_{e}=$ equivalent length of pipe segment, mi
$Z=$ gas compressibility factor, dimensionless
$D=$ pipe inside diameter, in.
Other symbols are as defined previously.
In SI units, the Panhandle A equation is

$$
\begin{equation*}
Q=4.5965 \times 10^{-3} E\left(\frac{T_{b}}{P_{b}}\right)^{1.0788}\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G^{0.8539} T_{f} L_{e} Z}\right)^{0.5394} \quad D^{2.6182} \quad \text { (SI units) } \tag{2.56}
\end{equation*}
$$

where
$Q=$ gas flow rate, standard $\mathrm{m}^{3} /$ day
$E=$ pipeline efficiency, a decimal value less than 1.0
$T_{b}=$ base temperature, $\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$P_{b}=$ base pressure, kPa
$T_{f}=$ average gas flow temperature, $\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$P_{1}=$ upstream pressure, kPa (absolute)
$P_{2}=$ downstream pressure, kPa (absolute)
$L_{e}=$ equivalent length of pipe segment, km
Other symbols are as defined previously.
Due to the exponents involved in this equation, all pressures must be in kPa .
By comparing the Panhandle A equation with the General Flow equation, we can calculate an equivalent transmission factor in USCS units as follows:

$$
\begin{equation*}
F=7.2111 E\left(\frac{Q G}{D}\right)^{0.07305} \quad(\mathrm{USCS}) \tag{2.57}
\end{equation*}
$$

and in SI units, it is

$$
\begin{equation*}
F=11.85 E\left(\frac{Q G}{D}\right)^{0.07305} \tag{SI}
\end{equation*}
$$

Sometimes the transmission factor is used to compare the results of calculations using the General Flow equation and the Panhandle A equation.

## Example 15

Using the Panhandle A equation, calculate the outlet pressure in a natural gas pipeline, NPS 16 with 0.250 in. wall thickness, 15 miles long. The gas flow rate is 100 MMSCFD at an inlet pressure of 1000 psia. The gas gravity $=0.6$ and viscosity $=0.000008 \mathrm{lb} / \mathrm{ft}$-sec. The average gas temperature is $80^{\circ} \mathrm{F}$. Assume base pressure $=14.73 \mathrm{psia}$ and base temperature $=60^{\circ}$. For compressibility factor $Z$, use the CNGA method. Assume pipeline efficiency of 0.92 .

## Solution

The average pressure, $P_{\text {avg }}$, needs to be calculated before the compressibility factor $Z$ can be determined. Since the inlet pressure $P_{1}=1,000 \mathrm{psia}$, and the outlet pressure $P_{2}$ is unknown, we will have to assume a value of $P_{2}$ (such as 800 psia ) and calculate $P_{\text {avg }}$ and then calculate the value of $Z$. Once $Z$ is known, from the Panhandle A equation we can calculate the outlet pressure $P_{2}$. Using this value of $P_{2}$, a better approximation for $Z$ is calculated from a new $P_{\text {avg. }}$. This process is repeated until successive values of $P_{2}$ are within allowable limits, such as 0.5 psia .

Assume $P_{2}=800$ psia. The average pressure from Equation 2.14 is

$$
P_{\text {avg }}=\frac{2}{3}\left(1000+800-\frac{1000 \times 800}{1000+800}\right)=903.7 \mathrm{psia}
$$

Next, we calculate the compressibility factor $Z$ using the CNGA method.
From Equation 1.34,

$$
Z=\frac{1}{1+\frac{(903.7-14.73) \times 3.444 \times 10^{5} \times(10)^{1.785 \times 0.6}}{(80+460)^{3.825}}}
$$

or

$$
Z=0.8869
$$

From Panhandle A Equation 2.55, substituting given values, neglecting elevations, we get
$100 \times 10^{6}=435.87 \times 0.92\left(\frac{60+460}{14.73}\right)^{1.0788}\left(\frac{1000^{2}-P_{2}^{2}}{(0.6)^{0.8539}(540 \times 15 \times 0.8869)}\right)^{0.5334}(15.5)^{2.6182}$
Solving for $P_{2}$, we get

$$
P_{2}=968.02 \mathrm{psia}
$$

Since this is different from the assumed value of $P_{2}=800$, we recalculate the average pressure and $Z$ using

$$
P_{2}=968.02 \mathrm{psia}
$$

The revised average pressure is

$$
P_{\text {avg }}=\frac{2}{3}\left(1000+968.02-\frac{1000 \times 968.02}{1000+968.02}\right)=984.10 \mathrm{psia}
$$

Using this value of $P_{\text {avg }}$, we recalculate $Z$ as

$$
Z=\frac{1}{1+\frac{(984.10-14.73) \times 3.444 \times 10^{5} \times(10)^{1.785 \times 0.6}}{(80+460)^{3.825}}}
$$

or

$$
Z=0.8780
$$

Recalculating $P_{2}$ from the Panhandle A Equation 2.55, we get
$100 \times 10^{6}=435.87 \times 0.92\left(\frac{60+460}{14.73}\right)^{1.0788}\left(\frac{1000^{2}-P_{2}^{2}}{(0.6)^{0.8539}(540 \times 15 \times 0.8780)}\right)^{0.5394}(15.5)^{2.6182}$

Solving for $P_{2}$, we get

$$
P_{2}=968.35 \mathrm{psia}
$$

This is within 0.5 psi of the previously calculated value. Hence, we will not continue the iteration any further.

Therefore, the outlet pressure is 968.35 psia .

## Example 16

Using the Panhandle A equation, calculate the inlet pressure required in a natural gas pipeline, DN 300 with 6 mm wall thickness, 24 km long, for a gas flow rate of $3.5 \mathrm{Mm}^{3} /$ day. The gas gravity $=0.6$ and viscosity $=0.000119$ Poise. The average gas temperature is $20^{\circ} \mathrm{C}$. The delivery pressure is 6000 kPa (absolute). Assume base pressure $=101 \mathrm{kPa}$, base temperature $=15^{\circ} \mathrm{C}$, and compressibility factor $Z=0.90$, with a pipeline efficiency of 0.92 .

## Solution

Pipe inside diameter $D=300-2 \times 6=288 \mathrm{~mm}$

Gas flow temperature $=20+273=293 \mathrm{~K}$
Using Panhandle A Equation 2.56 and neglecting elevation effect, we substitute

$$
3.5 \times 10^{6}=4.5965 \times 10^{-3} \times 0.92\left(\frac{15+273}{101}\right)^{1.0788}\left(\frac{P_{1}^{2}-6000^{2}}{(0.6)^{0.8539}(293 \times 24 \times 0.9)}\right)^{0.5394}(288)^{2.6182}
$$

Solving for inlet pressure, we get

$$
P_{1}^{2}-(6000)^{2}=19,812,783
$$

or

$$
P_{1}=7471 \mathrm{kPa} \text { (absolute) }
$$

### 2.16 PANHANDLE B EQUATION

The Panhandle B equation, also known as the revised Panhandle equation, is used in large diameter, high pressure transmission lines. In fully turbulent flow, it is found to be accurate for values of Reynolds number in the range of 4 to 40 million. This equation in USCS units is as follows:

$$
\begin{equation*}
Q=737 E\left(\frac{T_{b}}{P_{b}}\right)^{1.02}\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G^{0.961} T_{f} L_{e} Z}\right)^{0.51} D^{2.53} \quad \text { (USCS units) } \tag{2.59}
\end{equation*}
$$

where
$Q=$ volume flow rate, standard $\mathrm{ft}^{3} /$ day (SCFD)
$E=$ pipeline efficiency, a decimal value less than 1.0
$P_{b}=$ base pressure, psia
$T_{b}=$ base temperature, ${ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)$
$P_{1}=$ upstream pressure, psia
$P_{2}=$ downstream pressure, psia
$G=$ gas gravity (air = 1.00)
$T_{f}=$ average gas flow temperature, ${ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)$
$L_{e}=$ equivalent length of pipe segment, mi
$Z=$ gas compressibility factor, dimensionless
$D=$ pipe inside diameter, in.

Other symbols are as defined previously.
In SI units, the Panhandle B equation is

$$
\begin{equation*}
Q=1.002 \times 10^{-2} E\left(\frac{T_{b}}{P_{b}}\right)^{1.02}\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G^{0.961} T_{f} L_{e} Z}\right)^{0.51} D^{2.53} \quad \text { (SI units) } \tag{2.60}
\end{equation*}
$$

where
$Q=$ gas flow rate, standard $\mathrm{m}^{3} /$ day
$E=$ pipeline efficiency, a decimal value less than 1.0
$T_{b}=$ base temperature, $\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$P_{b}=$ base pressure, kPa
$T_{f}=$ average gas flow temperature, $\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$P_{1}=$ upstream pressure, kPa (absolute)
$P_{2}=$ downstream pressure, kPa (absolute)
$L_{e}=$ equivalent length of pipe segment, km
$Z=$ gas compressibility factor at the flowing temperature, dimensionless
Other symbols are as defined previously.
The equivalent transmission factor for the Panhandle B equation in USCS is given by

$$
\begin{equation*}
F=16.7 E\left(\frac{Q G}{D}\right)^{0.01961} \quad \text { (USCS units) } \tag{2.61}
\end{equation*}
$$

In SI units, it is

$$
\begin{equation*}
F=19.08 E\left(\frac{Q G}{D}\right)^{0.01961} \quad \text { (SI units) } \tag{2.62}
\end{equation*}
$$

## Example 17

Using the Panhandle B equation, calculate the outlet pressure in a natural gas pipeline, NPS 16 with 0.250 in . wall thickness, 15 miles long. The gas flow rate is 100 MMSCFD at 1000 psia inlet pressure. The gas gravity $=0.6$ and viscosity $=$ $0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{sec}$. The average gas temperature is $80^{\circ} \mathrm{F}$. Assume base pressure $=$ 14.73 psia and base temperature $=60^{\circ} \mathrm{F}$. The compressibility factor $Z=0.90$ and pipeline efficiency is 0.92 .

Solution
Inside diameter of pipe $=16-2 \times 0.25=15.5 \mathrm{in}$.
Gas flow temperature $=80+460=540^{\circ} \mathrm{R}$
Using Panhandle B Equation 2.59, substituting the given values, we get

$$
100 \times 10^{6}=737 \times 0.92\left(\frac{60+460}{14.73}\right)^{1.02}\left(\frac{1000^{2}-P_{2}^{2}}{(0.6)^{0.961}(540 \times 15 \times 0.90)}\right)^{0.51} 15.5^{2.53}
$$

Solving for $P_{2}$, we get

$$
\begin{gathered}
1000^{2}-P_{2}{ }^{2}=60,778 \\
P_{2}=969.13 \mathrm{psia}
\end{gathered}
$$

Compare this with the results of Panhandle A equation in Example 15, where the outlet pressure $P_{2}=968.35 \mathrm{psia}$. Therefore, the Panhandle B equation gives a slightly lower pressure drop compared to that from the Panhandle A equation. In other words, Panhandle A is more conservative and will give a lower flow rate for the
same pressures compared to Panhandle B. In this example, we use the constant value of $Z=0.9$, whereas in example $15, Z$ was calculated using the CNGA equation as $Z=0.8780$. If we factor this in, the result for the outlet pressure in this example will be 969.9 psia, which is not too different from the calculated value of 969.13 psia.

## Example 18

Using the Panhandle B equation, calculate the inlet pressure in a natural gas pipeline, DN 300 with 6 mm wall thickness, 24 km long. The gas flow rate is $3.5 \mathrm{Mm}^{3} /$ day, gas gravity $=0.6$, and viscosity $=0.000119$ Poise. The average gas temperature is $20^{\circ} \mathrm{C}$, and the delivery pressure is $6,000 \mathrm{kPa}$ (absolute). Assume base pressure $=$ 101 kPa , base temperature $=15^{\circ} \mathrm{C}$, and compressibility factor $Z=0.90$. The pipeline efficiency is 0.92 .

Solution

Inside diameter of pipe $=300-2 \times 6=288 \mathrm{~mm}$
Gas flow temperature $=20+273=293 \mathrm{~K}$

Neglecting elevations, using Panhandle B Equation 2.60, we get

$$
3.5 \times 10^{6}=1.002 \times 10^{-2} \times 0.92\left(\frac{15+273}{101}\right)^{1.02}\left(\frac{P_{1}^{2}-6000^{2}}{(0.6)^{0.961}(293 \times 24 \times 0.9)}\right)^{0.51} 288^{2.53}
$$

Solving for the inlet pressure $P_{1}$, we get

$$
\begin{gathered}
P_{1}^{2}-(6000)^{2}=19,945,469 \\
P_{1}=7480 \mathrm{kPa} \text { (absolute) }
\end{gathered}
$$

Compare this with the results of the Panhandle A equation in Example 16, where the inlet pressure $P_{1}=7471 \mathrm{kPa}$ (absolute). Again, we see that the Panhandle B equation gives a slightly lower pressure drop compared to that obtained from the Panhandle A equation.

### 2.17 INSTITUTE OF GAS TECHNOLOGY (IGT) EQUATION

The IGT equation proposed by the Institute of Gas Technology is also known as the IGT distribution equation and is stated as follows for USCS units:

$$
\begin{equation*}
Q=136.9 E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G^{0.8} T_{f} L_{e} \mu^{0.2}}\right)^{0.555} D^{2.667} \quad \text { (USCS units) } \tag{2.63}
\end{equation*}
$$

where
$Q=$ volume flow rate, standard $\mathrm{ft}^{3} /$ day (SCFD)
$E=$ pipeline efficiency, a decimal value less than 1.0

```
\(P_{b}=\) base pressure, psia
\(T_{b}=\) base temperature, \({ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)\)
\(P_{1}=\) upstream pressure, psia
\(P_{2}=\) downstream pressure, psia
\(G=\) gas gravity (air = 1.00)
\(T_{f}=\) average gas flow temperature, \({ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)\)
\(L_{e}=\) equivalent length of pipe segment, mi
\(Z=\) gas compressibility factor, dimensionless
\(D=\) pipe inside diameter, in.
\(\mu=\) gas viscosity, lb/ft-s
```

Other symbols are as defined previously.
In SI units, the IGT equation is expressed as follows:

$$
\begin{equation*}
Q=1.2822 \times 10^{-3} E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G^{0.8} T_{f} L_{e} \mu^{0.2}}\right)^{0.555} D^{2.667} \quad \text { (SI units) } \tag{2.64}
\end{equation*}
$$

where
$Q=$ gas flow rate, standard $\mathrm{m}^{3} / \mathrm{day}$
$E=$ pipeline efficiency, a decimal value less than 1.0
$T_{b}=$ base temperature, $\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$P_{b}=$ base pressure, kPa
$T_{f}=$ average gas flow temperature, $\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$P_{1}=$ upstream pressure, kPa (absolute)
$P_{2}=$ downstream pressure, kPa (absolute)
$L_{e}=$ equivalent length of pipe segment, km
$\mu=$ gas viscosity, Poise

Other symbols are as defined previously.

## Example 19

Using the IGT equation, calculate the flow rate in a natural gas pipeline, NPS 16 with 0.250 in . wall thickness, 15 mi long. The inlet and outlet pressure are 1000 psig and 800 psig , respectively. The gas gravity $=0.6$ and viscosity $=0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{s}$. The average gas temperature is $80^{\circ} \mathrm{F}$, base pressure $=14.7 \mathrm{psia}$, and base temperature $=60^{\circ} \mathrm{F}$. The compressibility factor $Z=0.90$, and the pipeline efficiency is 0.95 .

## Solution

Inside diameter of pipe $=16-2 \times 0.25=15.5 \mathrm{in}$.
The pressures given are in psig, and they must be converted to absolute pressures.

Therefore,

$$
\begin{aligned}
& P_{1}=1000+14.7=1014.7 \mathrm{psia} \\
& P_{2}=800+14.7=814.7 \mathrm{psia} \\
& T_{b}=60+460=520^{\circ} \mathrm{R} \\
& T_{f}=80+460=540^{\circ} \mathrm{R}
\end{aligned}
$$

Substituting in IGT Equation 2.63, we get

$$
\begin{gathered}
Q=136.9 \times 0.95\left(\frac{520}{14.7}\right)\left(\frac{1014.7^{2}-814.7^{2}}{(0.6)^{0.8} \times 540 \times 15 \times\left(8 \times 10^{-6}\right)^{0.2}}\right)^{0.555} 15.5^{2.667} \\
Q=263.1 \times 10^{6} \mathrm{ft}^{3} / \text { day }=263.1 \mathrm{MMSCFD}
\end{gathered}
$$

Therefore, the flow rate is 263.1 MMSCFD.

## Example 20

A natural gas pipeline, DN 400 with 6 mm wall thickness, 24 km long, is used to transport gas at an inlet pressure of 7000 kPa (gauge) and an outlet pressure of 5500 kPa (gauge). The gas gravity $=0.6$ and viscosity $=0.000119$ Poise. The average gas temperature is $20^{\circ} \mathrm{C}$. Assume base pressure $=101 \mathrm{kPa}$ and base temperature $=$ $15^{\circ} \mathrm{C}$. The compressibility factor $Z=0.90$ and pipeline efficiency is 0.95 .
a) Calculate the flow rate using the IGT equation.
b) What are the gas velocities at inlet and outlet?
c) If the velocity must be limited to $10 \mathrm{~m} / \mathrm{s}$, what should the minimum pipe size be, assuming the flow rate and inlet pressure remain constant?

Solution

Inside diameter of pipe $D=400-2 \times 6=388 \mathrm{~mm}$

All pressures are given in gauge values and must be converted to absolute values.

Inlet pressure $P_{1}=7000+101=7101 \mathrm{kPa}$ (absolute)
Outlet pressure $P_{2}=5500+101=5601 \mathrm{kPa}$ (absolute)
Base temperature $T_{b}=15+273=288 \mathrm{~K}$

Flowing temperature $T_{f}=20+273=293 \mathrm{~K}$
From IGT Equation 2.64, we get the flow rate in $\mathrm{m}^{3} /$ day as

$$
Q=1.2822 \times 10^{-3} \times 0.95\left(\frac{288}{101}\right)\left(\frac{7101^{2}-5601^{2}}{(0.6)^{0.8} \times 293 \times 24 \times\left(1.19 \times 10^{-4}\right)^{0.2}}\right)^{0.555}(388)^{2.667}
$$

or

$$
Q=7,665,328 \mathrm{~m}^{3} / \mathrm{day}=7.67 \mathrm{Mm}^{3} / \mathrm{day}
$$

(a) Therefore, the flow rate is $7.67 \mathrm{Mm}^{3} / \mathrm{day}$.
(b) Using Equation 2.29, we calculate the average velocity of the gas at the inlet pressure as

$$
\text { Inlet velocity } u_{1}=14.7349\left(\frac{7.67 \times 10^{6}}{388^{2}}\right)\left(\frac{101}{288}\right)\left(\frac{0.9 \times 293}{7101}\right)=9.78 \mathrm{~m} / \mathrm{s}
$$

In the preceding, we assumed a constant compressibility factor, $Z=0.9$.
Similarly, at the outlet pressure, the average gas velocity is

$$
\text { Outlet velocity } u_{2}=14.7349\left(\frac{7.67 \times 10^{6}}{388^{2}}\right)\left(\frac{101}{288}\right)\left(\frac{0.9 \times 293}{5601}\right)=12.4 \mathrm{~m} / \mathrm{s}
$$

(c) Since the velocity must be limited to $10 \mathrm{~m} / \mathrm{s}$, the pipe diameter must be increased. Increasing the pipe diameter will also increase the outlet pressure if we keep both the flow rate and inlet pressure the same as before. The increased outlet pressure will also reduce the gas velocity as can be seen from Equation 2.29. We will try a DN 450 pipe with 10 mm wall thickness.

$$
\text { Inside diameter of pipe } D=450-2 \times 10=430 \mathrm{~mm}
$$

Assuming $P_{1}$ and $Q$ are the same as before, we calculate the new outlet pressure $P_{2}$ from IGT Equation 2.64 as
$7.67 \times 10^{6}=1.2822 \times 10^{-3} \times 0.95\left(\frac{288}{101}\right)\left(\frac{7101^{2}-P_{2}^{2}}{(0.6)^{0.8} \times 293 \times 24 \times\left(1.19 \times 10^{-4}\right)^{0.2}}\right)^{0.555}(430)^{2.667}$
Solving for $P_{2}$, we get

$$
P_{2}=6228 \mathrm{kPa}
$$

The new velocity at the outlet will be

$$
u_{2}=14.7349\left(\frac{7.67 \times 10^{6}}{430^{2}}\right)\left(\frac{101}{288}\right)\left(\frac{0.9 \times 293}{6228}\right)=9.08 \mathrm{~m} / \mathrm{s}
$$

Since this is less than the $10 \mathrm{~m} / \mathrm{s}$ specified, the DN 450 pipe is satisfactory.
In the preceding calculations we assumed the same compressibility factor for both inlet and outlet pressures. Actually, a more nearly correct solution would be to calculate $Z$ using the CNGA equation at both inlet and outlet conditions and using these values in the calculation of gas velocities. This is left as an exercise for the reader.

### 2.18 SPITZGLASS EQUATION

The Spitzglass equation has been around for many years and originally was used in fuel gas piping calculations. There are two versions of the Spitzglass equation. One equation is for low pressure (less than or equal to 1 psig ) and another is for high pressure (more than 1 psig ). These equations have been modified to include a pipeline efficiency and compressibility factor.

The low-pressure (less than or equal to 1 psig ) version of the Spitzglass equation in USCS units is

$$
\begin{equation*}
Q=3.839 \times 10^{3} E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}-P_{2}}{G T_{f} L_{e} Z\left(1+\frac{3.6}{D}+0.03 D\right)}\right)^{0.5} D^{2.5} \quad \text { (USCS units) } \tag{2.65}
\end{equation*}
$$

where
$Q=$ volume flow rate, standard $\mathrm{ft}^{3} /$ day (SCFD)
$E=$ pipeline efficiency, a decimal value less than 1.0
$P_{b}=$ base pressure, psia
$T_{b}=$ base temperature, ${ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)$
$P_{1}=$ upstream pressure, psia
$P_{2}=$ downstream pressure, psia
$G=$ gas gravity (air = 1.00)
$T_{f}=$ average gas flow temperature, ${ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)$
$L_{e}=$ equivalent length of pipe segment, mi
$D=$ pipe inside diameter, in.
$Z=$ gas compressibility factor, dimensionless

Other symbols are as defined previously.
The low-pressure (less than 6.9 kPa ) version of the Spitzglass equation in SI units is

$$
\begin{equation*}
Q=5.69 \times 10^{-2} E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}-P_{2}}{G T_{f} L_{e} Z\left(1+\frac{91.44}{D}+0.0012 D\right)}\right)^{0.5} D^{2.5} \quad \text { (SI units) } \tag{2.66}
\end{equation*}
$$

where
$Q=$ gas flow rate, standard $\mathrm{m}^{3} /$ day
$E=$ pipeline efficiency, a decimal value less than 1.0
$T_{b}=$ base temperature, $\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$P_{b}=$ base pressure, kPa
$P_{1}=$ upstream pressure, kPa (absolute)
$P_{2}=$ downstream pressure, kPa (absolute)
$G=$ gas gravity (air = 1.00)
$T_{f}=$ average gas flow temperature, $\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$L_{e}=$ equivalent length of pipe segment, km
$Z=$ gas compressibility factor, dimensionless
Other symbols are as defined previously.
The high-pressure (more than 1 psig ) version in USCS units is as follows.

$$
\begin{equation*}
Q=729.6087 E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G T_{f} L_{e} Z\left(1+\frac{3.6}{D}+0.03 D\right)}\right)^{0.5} D^{2.5} \quad \text { (USCS units) } \tag{2.67}
\end{equation*}
$$

where
$Q=$ volume flow rate, standard $\mathrm{ft}^{3} /$ day (SCFD)
$E=$ pipeline efficiency, a decimal value less than 1.0
$P_{b}=$ base pressure, psia
$T_{b}=$ base temperature, ${ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)$
$P_{1}=$ upstream pressure, psia
$P_{2}=$ downstream pressure, psia
$G=$ gas gravity (air = 1.00)
$T_{f}=$ average gas flow temperature, ${ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)$
$L_{e}=$ equivalent length of pipe segment, mi
$D=$ pipe inside diameter, in.
$Z=$ gas compressibility factor, dimensionless
Other symbols are as defined previously.
In SI units, the high-pressure (more than 6.9 kPa ) version of the Spitzglass equation is

$$
\begin{equation*}
Q=1.0815 \times 10^{-2} E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G T_{f} L_{e} Z\left(1+\frac{91.44}{D}+0.0012 D\right)}\right)^{0.5} D^{2.5} \quad \text { (SI units) } \tag{2.68}
\end{equation*}
$$

where
$Q=$ gas flow rate, standard $\mathrm{m}^{3} /$ day
$E=$ pipeline efficiency, a decimal value less than 1.0
$T_{b}=$ base temperature, $\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$P_{b}=$ base pressure, kPa
$P_{1}=$ upstream pressure, kPa (absolute)
$P_{2}=$ downstream pressure, kPa (absolute)
$G=$ gas gravity (air = 1.00)
$T_{f}=$ average gas flow temperature, $\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$L_{e}=$ equivalent length of pipe segment, km
$Z=$ gas compressibility factor, dimensionless
Other symbols are as defined previously.

## Example 21

Calculate the fuel gas capacity of an NPS 6 pipe, with an inside diameter of 6.065 in. and a total equivalent length of 180 ft . The flowing temperature of fuel gas is $60^{\circ} \mathrm{F}$, and the inlet pressure is 1.0 psig . Consider a pressure drop of 0.7 in the water column and the specific gravity of gas $=0.6$. Assume pipeline efficiency $E=1.0$ and compressibility factor $Z=1.0$. The base pressure and base temperature are 14.7 psia and $60^{\circ} \mathrm{F}$, respectively.

Solution
Base temperature $=60+460=520^{\circ} \mathrm{R}$
Gas flowing temperature $=60+460=520^{\circ} \mathrm{R}$

$$
\text { Pressure drop }\left(P_{1}-P_{2}\right)=1.0-\frac{0.7}{12} \times 0.433=0.9747 \mathrm{psi}
$$

Since this is low pressure, using Spitzglass Equation 2.65, we get

$$
\begin{gathered}
Q=3.839 \times 10^{3} \times 1.0 \times\left(\frac{520}{14.7}\right)\left(\frac{0.9747}{0.6 \times 520 \times \frac{180}{5280} \times 1.0\left(1+\frac{3.6}{6.065}+0.03 \times 6.065\right)}\right)^{0.5} 6.065^{2.5} \\
Q=2,794,842 \mathrm{SCFD}=2.79 \mathrm{MMSCFD}
\end{gathered}
$$

Therefore, the fuel gas capacity is 2.79 MMSCFD.

### 2.19 MUELLER EQUATION

The Mueller equation is another form of the flow rate vs. pressure relationship in gas pipelines. In USCS units, it is expressed as follows:

$$
\begin{equation*}
Q=85.7368 E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G^{0.7391} T_{f} L_{e} \mu^{0.2609}}\right)^{0.575} D^{2.725} \quad \text { (USCS units) } \tag{2.69}
\end{equation*}
$$

where
$Q=$ volume flow rate, standard $\mathrm{ft}^{3} /$ day (SCFD)
$E=$ pipeline efficiency, a decimal value less than 1.0
$P_{b}=$ base pressure, psia
$T_{b}=$ base temperature, ${ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)$
$P_{1}=$ upstream pressure, psia
$P_{2}=$ downstream pressure, psia
$G=$ gas gravity (air = 1.00)
$T_{f}=$ average gas flow temperature, ${ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)$
$L_{e}=$ equivalent length of pipe segment, mi
$D=$ pipe inside diameter, in.
$\mu=$ gas viscosity, lb/ft-s
Other symbols are as defined previously.
In SI units, the Mueller equation is as follows:

$$
\begin{equation*}
Q=3.0398 \times 10^{-2} E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G^{0.7391} T_{f} L_{e} \mu^{0.2609}}\right)^{0.575} D^{2.725} \quad \text { (SI units) } \tag{2.70}
\end{equation*}
$$

where
$Q=$ gas flow rate, standard $\mathrm{m}^{3} /$ day
$E=$ pipeline efficiency, a decimal value less than 1.0
$T_{b}=$ base temperature, $\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$P_{b}=$ base pressure, kPa
$P_{1}=$ upstream pressure, kPa (absolute)
$P_{2}=$ downstream pressure, kPa (absolute)
$G=$ gas gravity (air = 1.00)
$T_{f}=$ average gas flow temperature, $\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)$
$L_{e}=$ equivalent length of pipe segment, km
$\mu=$ gas viscosity, cP
Other symbols are as defined previously.

### 2.20 FRITZSCHE EQUATION

The Fritzsche formula, developed in Germany in 1908, has found extensive use in compressed air and gas piping. In USCS units, it is expressed as follows:

$$
\begin{equation*}
Q=410.1688 E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-P_{2}^{2}}{G^{0.8587} T_{f} L_{e}}\right)^{0.538} D^{2.69} \quad \text { (USCS units) } \tag{2.71}
\end{equation*}
$$

All symbols are as defined before.
In SI units,

$$
\begin{equation*}
Q=2.827 E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G^{0.8587} T_{f} L_{e}}\right)^{0.538} D^{2.69} \quad \text { (SI units) } \tag{2.72}
\end{equation*}
$$

All symbols are as defined before.

### 2.21 EFFECT OF PIPE ROUGHNESS

In the preceding sections, we used the pipe roughness as a parameter in the friction factor and transmission factor calculations. Both the AGA and Colebrook-White equations use the pipe roughness, whereas the Panhandle and Weymouth equations do not use the pipe roughness directly in the calculations. Instead, these equations use a pipeline efficiency to compensate for the internal conditions and age of the pipe. Therefore, when comparing the predicted flow rates or pressures using the AGA or Colebrook-White equations with the Panhandle or Weymouth equations, we can adjust the pipeline efficiency to correlate with the pipe roughness used in the former equations.

Since most gas pipelines operate in the turbulent zone, the laminar flow friction factor, which is independent of pipe roughness, is of little interest to us. Concentrating, therefore, on turbulent flow, we see that Colebrook-White Equation 2.45 is affected by variation in pipe internal roughness. For example, suppose we want to compare an internally coated pipeline with an uncoated pipeline. The internal roughness of the coated pipe might be in the range of 100 to $200 \mu \mathrm{in}$., whereas the uncoated pipe might have a roughness of 600 to $800 \mu \mathrm{in}$. or more. If the pipe is NPS 20 with a 0.500 in . wall thickness, the relative roughness using the lower roughness value is as follows:

For coated pipe,

$$
\frac{e}{D}=\frac{100 \times 10^{-6}}{19}=5.263 \times 10^{-6}
$$

and
For uncoated pipe,

$$
\frac{e}{D}=\frac{600 \times 10^{-6}}{19}=3.1579 \times 10^{-5}
$$

Substituting these values of relative roughness in Equation 2.45 and using a Reynolds number of 10 million, we calculate the following transmission factors:

$$
F=21.54 \text { for coated pipe }
$$

and

$$
F=19.83 \text { for uncoated pipe }
$$

Since the flow rate is directly proportional to the transmission factor $F$, from General Flow Equation 2.4 we see that the coated pipe will be able to transport $\frac{21.54-19.83}{19.83}=0.086=8.6 \%$ more flow rate than the uncoated pipe, if all other parameters remain the same.

This is true in the fully turbulent zone where the Reynolds number has little effect on the friction factor $f$ and the transmission factor $F$. However, in the smooth
pipe zone, pipe roughness has less effect on the friction factor and the transmission factor. This is evident from the Moody diagram in Figure 2.3.

Using a Reynolds number of $10^{6}$, we find from the Moody diagram in Figure 2.3, for coated pipe, that

$$
f=0.0118 \quad \text { and } \quad F=18.41
$$

and, for the uncoated pipe,

$$
f=0.0122 \text { and } F=18.10
$$

Therefore, the increase in flow rate in this case will be

$$
\frac{18.41-18.10}{18.10}=0.017=1.7 \%
$$

Thus, the impact of pipe roughness is less in the smooth pipe zone or for a lower Reynolds number. A similar comparison can be made using the AGA equation.

Figure 2.4 shows the effect of pipe roughness on the pipeline flow rate considering the AGA and Colebrook-White equations. The graph is based on NPS 20 pipe, 0.500 in. wall thickness, 120 miles long, with 1200 psig upstream pressure and 800 psig downstream pressure. The flowing temperature of gas is $70^{\circ} \mathrm{F}$.

It can be seen that as the pipe roughness is increased from 200 to $800 \mu \mathrm{in}$., the flow rate decreases from 224 MMSCFD to 206 MMSCFD for the Colebrook-White equation and from 220 MMSCFD to 196 MMSCFD for the AGA equation.


Figure 2.4 Effect of pipe roughness.

We can therefore conclude that decreasing the pipe roughness directly results in a throughput increase in a pipeline. However, the cost of internally coating a pipe to reduce the pipe roughness must be weighed against the revenue increase due to enhanced flow rate. We will revisit this issue in Chapter 10, when we discuss pipeline economics.

### 2.22 COMPARISON OF FLOW EQUATIONS

In the preceding sections, we calculated the flow rates and pressures in gas pipelines using the various flow equations. Each equation is slightly different from the other, and some equations consider the pipeline efficiency while others use an internal pipe roughness value. How do these equations compare when predicting flow rates through a given pipe size when the upstream or downstream pressure is held constant? Obviously, some equations will predict higher flow rates for the same pressures than others. Similarly, if we start with a fixed upstream pressure in a pipe segment at a given flow rate, these equations will predict different downstream pressures. This indicates that some equations calculate higher pressure drops for the same flow rate than others.

Figure 2.5 and Figure 2.6 show some of these comparisons when using the AGA, Colebrook-White, Panhandle, and Weymouth equations.

In Figure 2.5, we consider a pipeline 100 mi long, NPS 16 with 0.250 in . wall thickness, operating at a flow rate of 100 MMSCFD. The gas flowing temperature is $80^{\circ} \mathrm{F}$. With the upstream pressure fixed at 1400 psig , the downstream pressure was calculated using the different flow equations. By examining Figure 2.5, it is clear that the highest pressure drop is predicted by the Weymouth equation and the lowest pressure drop is predicted by the Panhandle B equation. It must be noted that we


Figure 2.5 Comparison of flow equations.


Figure 2.6 Upstream pressures for various flow equations.
used a pipe roughness of $700 \mu \mathrm{in}$. for both the AGA and Colebrook equations, whereas a pipeline efficiency of 0.95 was used in the Panhandle and Weymouth equations.

Figure 2.6 shows a comparison of the flow equations from a different perspective. In this case, we calculated the upstream pressure required for an NPS 30 pipeline, 100 miles long, holding the delivery pressure constant at 800 psig. The upstream pressure required for various flow rates, ranging from 200 to 600 MMSCFD , was calculated using the five flow equations. Again it can be seen that the Weymouth equation predicts the highest upstream pressure at any flow rate, whereas the Panhandle A equation calculates the least pressure. We therefore conclude that the most conservative flow equation that predicts the highest pressure drop is the Weymouth equation and the least conservative flow equation is Panhandle A.

### 2.23 SUMMARY

In this chapter we introduced the various methods of calculating the pressure drop in a pipeline transporting gas and gas mixtures. The more commonly used equations for pressure drop vs. flow rate and pipe size were discussed and illustrated using example problems. The effect of elevation changes was explained, and the concepts of the Reynolds number, friction factor, and transmission factor were introduced. The importance of the Moody diagram and how to calculate the friction factor for laminar and turbulent flow were explained. We compared the more commonly used pressure drop equations, such as AGA, Colebrook-White, Weymouth, and Panhandle equations. The use of a pipeline efficiency in comparing various equations was illustrated using an example. The average velocity of gas flow was introduced, and the limiting value of erosional velocity was discussed.

## PROBLEMS

1. A gas pipeline, NPS 18 with 0.375 in . wall thickness, transports natural gas (specific gravity $=0.6$ ) at a flow rate of 160 MMSCFD at an inlet temperature of $60^{\circ} \mathrm{F}$. Assuming isothermal flow, calculate the velocity of gas at the inlet and outlet of the pipe if the inlet pressure is 1200 psig and the outlet pressure is 700 psig . The base pressure and base temperature are 14.7 psia and $60^{\circ} \mathrm{F}$, respectively. Assume the compressibility factor $Z=0.95$. What is the pipe length for these pressures, if elevations are neglected?
2. A natural gas pipeline, DN 400 with 10 mm wall thickness, transports $3.2 \mathrm{Mm}^{3} /$ day. The specific gravity of gas is 0.6 and viscosity is 0.00012 Poise. Calculate the value of the Reynolds number. Assume the base temperature and base pressure are 15 C and 101 kPa , respectively.
3. A natural gas pipeline, NPS 20 with 0.500 in. wall thickness, 50 miles long, transports 220 MMSCFD. The specific gravity of gas is 0.6 and viscosity is $0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{s}$. Calculate the friction factor using the Colebrook equation. Assume absolute pipe roughness $=750 \mu \mathrm{in}$. The base temperature and base pressure are $60^{\circ} \mathrm{F}$ and 14.7 psia, respectively. What is the upstream pressure for an outlet pressure of 800 psig?
4. For a gas pipeline flowing $3.5 \mathrm{Mm}^{3} /$ day gas of specific gravity 0.6 and viscosity of 0.000119 Poise, calculate the friction factor and transmission factor, assuming a DN 400 pipeline, 10 mm wall thickness, and internal roughness of 0.015 mm . The base temperature and base pressure are $15^{\circ} \mathrm{C}$ and 101 kPa , respectively. If the flow rate is increased by $50 \%$, what is the impact on the friction factor and transmission factor? If the pipe length is 48 km , what is the outlet pressure for an inlet pressure of 9000 kPa ?
5. A gas pipeline flows 110 MMSCFD gas of specific gravity 0.65 and viscosity of $0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{s}$. Calculate, using the modified Colebrook-White equation, the friction factor and transmission factor, assuming an NPS 20 pipeline, 0.375 in . wall thickness, and internal roughness of $700 \mu \mathrm{in}$. The base temperature and base pressure are $60^{\circ} \mathrm{F}$ and 14.7 psia , respectively.
6. Using the AGA method, calculate the transmission factor and friction factor for gas flow in an NPS 20 pipeline with 0.375 in. wall thickness. The flow rate is 250 MMSCFD, gas gravity $=0.6$, and viscosity $=0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{sec}$. The absolute pipe roughness is $600 \mu \mathrm{in}$. Assume a bend index of $60^{\circ}$, base pressure $=$ 14.73 psia , and base temperature $=60^{\circ} \mathrm{F}$. If the flow rate is doubled, what pipe size is needed to keep both inlet and outlet pressures the same as that at the original flow rate?
7. A natural gas transmission line transports 4 million $\mathrm{m}^{3} / \mathrm{day}$ of gas from a processing plant to a compressor station site 100 km away. The pipeline can be assumed to be along a flat terrain. Calculate the minimum pipe diameter required such that the maximum pipe operating pressure is limited to 8500 kPa . The delivery pressure desired at the end of the pipeline is a minimum of 5500 kPa . Assume a pipeline efficiency of 0.92 . The gas gravity is 0.60 , and the gas temperature is $18^{\circ} \mathrm{C}$. Use the Weymouth equation, considering a base temperature $=15^{\circ} \mathrm{C}$ and base pressure $=101 \mathrm{kPa}$. The gas compressibility factor $Z=0.90$.
8. Using the Panhandle B equation, calculate the outlet pressure in a natural gas pipeline, NPS 16 with 0.250 in . wall thickness, 25 miles long. The gas flow rate is 120 MMSCFD at 1200 psia inlet pressure. The gas gravity $=0.6$ and viscosity $=$ $0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{sec}$. The average gas temperature is $80^{\circ} \mathrm{F}$. Assume the base pressure $=$ 14.73 psia and base temperature $=60^{\circ} \mathrm{F}$. The compressibility factor $Z=0.90$ and pipeline efficiency is 0.95 .

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## CHAPTER 3

## Pressure Required to Transport

In this chapter we will extend the use of the concepts of pressure drop calculations developed in Chapter 2 to determine the total pressure required for transporting gas in a pipeline under various configurations, such as series and parallel pipelines. We will identify the various components that make up this total pressure and analyze their impact on gas pipeline pressures. The effect of intermediate delivery volumes and injection rates along a gas pipeline, the impact of contract delivery pressures, and the necessity of regulating pressures using a control valve or pressure regulators will also be analyzed. Thermal effects due to heat transfer between the gas and the surrounding soil in a buried pipeline, soil temperatures and thermal conductivities, and the JouleThompson effect will be introduced with reference to commercial hydraulic simulation models. Equivalent lengths in series piping and equivalent diameters in parallel piping will be explained. We will compare different pipe looping scenarios to improve pipeline throughput and review the concept of the hydraulic pressure gradient. Calculation methodology for line pack in a gas pipeline will also be discussed.

### 3.1 TOTAL PRESSURE DROP REQUIRED

In the flow of incompressible fluids such as water, the pressure required to transport a specified volume of fluid from point $A$ to point $B$ will consist of the following components:

1. Frictional component
2. Elevation component
3. Pipe delivery pressure

In addition, in some cases where the pipeline elevation differences are drastic, we must also take into account the minimum pressure in a pipeline such that vaporization of liquid does not occur. The latter results in two-phase flow in the pipeline, which causes higher pressure drop and, therefore, more pumping power requirement in addition to possible damage to pumping equipment. Thus, single-phase incompressible
fluids must be pumped such that the pressure at any point in the pipeline does not drop below the vapor pressure of the liquid.

When pumping gases, which are compressible fluids, the three components listed in the preceding section also contribute to the total pressure required. Even though the relationship between the total pressure required and the pipeline elevation is not straightforward (as in liquid flow), the dependency still exists and will be demonstrated using an example problem.

Going back to the case of a liquid pipeline, suppose the total pressure required to pump a given volume is 1000 psig and it is composed of the following components:

1. Frictional component $=600 \mathrm{psig}$
2. Elevation component $=300 \mathrm{psig}$
3. Delivery pressure $=100 \mathrm{psig}$

We will now discuss each of these components that make up the total pressure required by comparing the situation between a liquid pipeline and a gas pipeline.

### 3.2 FRICTIONAL EFFECT

The frictional effect results from the fluid viscosity and pipe roughness. It is similar in liquid and gas flow. The effect of friction was discussed in Chapter 2, where we introduced the internal roughness of pipe and how the friction factor and transmission factor were calculated using the Colebrook-White and AGA equations. We also discussed how the Weymouth and Panhandle equations took into account the internal conditions and age of the pipe by utilizing a pipeline efficiency factor rather than a friction factor. The magnitude of the pressure drop due to friction in a gas pipeline is generally held to smaller values in comparison with liquid pipelines. This is because efficient gas pipeline transportation requires keeping the average gas pressure as high as possible. As pressure drops due to expansion of gas, there is loss in efficiency. The lower the pressure at the downstream end, the higher will be the compression ratio required (hence, the higher the HP ) to boost the pressure for shipment downstream to the next compressor station in a long-distance gas pipeline. In this chapter we will continue to calculate the pressure drop due to friction in various pipe configurations that include flow injection, deliveries, and series and parallel piping.

### 3.3 EFFECT OF PIPELINE ELEVATION

The elevation component referred to in Section 3.1 is due to the difference in elevation along the pipeline that necessitates additional pressure for raising the fluid in the pipeline from one point to another. Of course, a drop in elevation will have the opposite effect of a rise in elevation.

The elevation component of 300 psig in the preceding example depends upon the static elevation difference between the beginning of the pipeline, A, and the delivery point, B, and the liquid specific gravity. In the case of a gas pipeline, the
elevation component will depend upon the static elevation differences between A and B, as well as the gas gravity. However, the relationship between these parameters is more complex in a gas pipeline compared to a liquid pipeline. The rise and fall in elevations between the origin A and the terminus B have to be accounted for separately and summed up according to Section 2.4 in Chapter 2. Further, compared to a liquid, the gas gravity is several orders of magnitude lower and, hence, the influence of elevation is smaller in a pipeline that transports gas. Generally, if we were to break down the total pressure required in a gas pipeline into the three components discussed earlier, we would find that the elevation component is very small. Let us illustrate this using an example.

## Example 1

A gas pipeline, NPS 16 with 0.250 in . wall thickness, 50 mi long, transports natural gas (specific gravity $=0.6$ and viscosity $=0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{s}$ ) at a flow rate of 100 MMSCFD at an inlet temperature of $60^{\circ} \mathrm{F}$. Assuming isothermal flow, calculate the inlet pressure required if the required delivery pressure at the pipeline terminus is 870 psig . The base pressure and base temperature are 14.7 psig and $60^{\circ} \mathrm{F}$, respectively. Use the Colebrook equation with pipe roughness of 0.0007 in.

Case A-Consider no elevation changes along the pipeline length.
Case B-Consider elevation changes as follows: inlet elevation of 100 ft and elevation at delivery point of 450 ft , with elevation at the midpoint of 250 ft .

## Solution

Inside diameter of pipe $\mathrm{D}=16-2 \times 0.250=15.5 \mathrm{in}$.

First, we calculate the Reynolds number from Equation 2.34:

$$
R=0.0004778\left(\frac{14.7}{60+460}\right)\left(\frac{0.6 \times 100 \times 10^{6}}{0.000008 \times 15.5}\right)=6,535,664
$$

Next, using Colebrook Equation 2.39, we calculate the friction factor as

$$
\frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{0.0007}{3.7 \times 15.5}+\frac{2.51}{6535664 \sqrt{f}}\right)
$$

Solving by trial and error, we get

$$
f=0.0109
$$

Therefore, the transmission factor is, using Equation 2.42,

$$
F=\frac{2}{\sqrt{0.0109}}=19.1954
$$

To calculate the compressibility factor $Z$, the average pressure is required. Since the inlet pressure is unknown, we will calculate an approximate value of $Z$ using a value of $110 \%$ of the delivery pressure for the average pressure.

The average pressure is

$$
P_{\mathrm{avg}}=1.1 \times(870+14.7)=973.17 \mathrm{psia}
$$

Using CNGA Equation 1.34, we calculate the value of the compressibility factor as

$$
Z=\frac{1}{\left[1+\left(\frac{(973.17-14.7) \times 344400(10)^{1.785 \times 0.6}}{520^{3.825}}\right)\right]}=0.8629
$$

Case A

Since there is no elevation difference between the beginning of the pipeline and the end of the pipeline, the elevation component in Equation 2.7 can be neglected, and $e^{s}=1$.

The outlet pressure is

$$
P_{2}=870+14.7=884.7 \mathrm{psia}
$$

From General Flow Equation 2.4, substituting the given values, we get

$$
100 \times 10^{6}=38.77 \times 19.1954\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-884.7^{2}}{0.6 \times 520 \times 50 \times 0.8629}\right)^{0.5}(15.5)^{2.5}
$$

Therefore, the upstream pressure is

$$
P_{1}=999.90 \mathrm{psia}=985.20 \mathrm{psig}
$$

Using this value of $P_{1}$, we calculate the new average pressure using Equation 2.14:

$$
P_{\mathrm{avg}}=\frac{2}{3}\left(999.9+884.7-\frac{999.9 \times 884.7}{999.9+884.7}\right)=943.47 \mathrm{psia}
$$

compared to 973.17 we used for calculating $Z$. Recalculating $Z$ using the new value of $P_{\text {avg }}$, we get

$$
Z=\frac{1}{\left[1+\left(\frac{(943.47-14.7) \times 344400(10)^{1.785 \times 0.6}}{520^{3.825}}\right)\right]}=0.8666
$$

This compares with 0.8629 we calculated earlier for $Z$. We will now recalculate the inlet pressure using this value of $Z$. From General Flow Equation 2.4, we get

$$
100 \times 10^{6}=38.77 \times 19.1954\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-884.7^{2}}{0.6 \times 520 \times 50 \times 0.8666}\right)^{0.5}(15.5)^{2.5}
$$

Solving for the upstream pressure, we get

$$
P_{1}=1000.36 \mathrm{psia}=985.66 \mathrm{psig}
$$

This is close enough to the previously calculated value 985.20 psig , and no further iteration is needed. Therefore, the pressure required at the beginning of the pipeline in case A is 985.66 psig when the elevation difference is zero.

We will now calculate the pressure required, taking into account the given elevations at the beginning, midpoint, and end of the pipeline.

## Case B

We will use $Z=0.8666$ throughout, as in case A .
Using Equation 2.10, the elevation adjustment factor is first calculated for each of the two segments.

For the first segment, from milepost 0.0 to milepost 25.0 , we get

$$
s_{1}=0.0375 \times 0.6\left(\frac{250-100}{520 \times 0.8666}\right)=0.0075
$$

Similarly, for the second segment, from milepost 25.0 to milepost 50.0 , we get

$$
s_{2}=0.0375 \times 0.6\left(\frac{450-100}{520 \times 0.8666}\right)=0.0175
$$

Therefore, the adjustment for elevation is, using Equation 2.12,

$$
j=\frac{e^{0.0075}-1}{0.0075}=1.0038 \text { for the first segment }
$$

and

$$
j=\frac{e^{0.0175}-1}{0.0175}=1.0088 \text { for the second segment }
$$

For the entire length,

$$
s_{2}=0.0375 \times 0.6\left(\frac{450-100}{520 \times 0.8666}\right)=0.0175
$$

The equivalent length from Equation 2.13 is then

$$
L_{e}=1.0038 \times 25+1.0088 \times 25 \times e^{0.0075}=50.5049 \mathrm{mi} .
$$

Therefore, we see that the effect of the elevation is taken into account partly by increasing the pipe length from 50 mi to 50.50 mi , approximately.

Substituting in Equation 2.7, we get

$$
100 \times 10^{6}=38.77 \times 19.1954\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-e^{0.0175} 884.7^{2}}{0.6 \times 520 \times 50.50 \times 0.8666}\right)^{0.5} 15.5^{2.5}
$$

Solving for the inlet pressure $P_{1}$,

$$
P_{1}=1008.34 \mathrm{psia}=993.64 \mathrm{psig}
$$

Thus, the pressure required at the beginning of the pipeline in case B is 993.64 psig, taking into account elevation difference along the pipeline. Compare this with 985.66 calculated ignoring the elevation differences.

For simplicity, we assume the same value of $Z$ in the preceding calculations as in the previous case. To be correct, we should recalculate $Z$ based on the average pressure and repeat calculations until the results are within 0.1 psi . This is left as an exercise for the reader.

It can be seen from the preceding calculations that, due to an elevation difference of $350 \mathrm{ft}(450 \mathrm{ft}-100 \mathrm{ft}$ ) between the delivery point and the beginning of the pipeline, the required pressure is approximately $8 \mathrm{psig}(993.64 \mathrm{psig}-985.66 \mathrm{psig})$ more. In a liquid line, the effect of elevation would have been more. The elevation difference of 350 ft in a water line would result in an increased pressure of

$$
350 \times 0.433=152 \mathrm{psi}, \text { approximately, at the upstream end. }
$$

### 3.4 EFFECT OF CHANGING PIPE DELIVERY PRESSURE

The delivery pressure component discussed in Section 3.1 is also similar to that between liquid and gas pipelines. The higher the pressure desired at the delivery end or terminus of the pipeline, the higher will be the total pressure required at the upstream end of the pipeline.

The impact of changing the delivery pressure is not linear in the case of a compressible fluid such as natural gas. For example, in a liquid pipeline, changing the delivery pressure from 100 to 200 psig will simply increase the required pressure at the pipe inlet by the same amount. Thus, suppose 1000 psig was the required inlet pressure in a liquid pipeline, at a certain flow rate and at a delivery pressure of 100 psig . When the delivery pressure required is increased to 200 psig , the inlet pressure will increase to exactly 1100 psig. We will now explore the effect of changing the contract delivery pressure at the end of a gas pipeline.

In liquid pipelines, an increase or decrease in the delivery pressure will proportionately increase or decrease the upstream pressure. In a gas pipeline, the increase and decrease in the upstream pressure will not be proportionate due to the nonlinear nature of the gas pressure drop. This will be explained in more detail in Section 3.9 in this chapter.

Suppose that in the preceding Example 1 (neglecting elevation change), the delivery pressure required increases from 870 to 950 psig. If pressure variations were linear, as in a liquid pipeline, we would expect the required inlet pressure to increase from 985.66 to $985.66+(950-870)=1066$ psig, approximately. However, this is incorrect because the pressure variation is not linear in gas pipelines. We will now calculate the required inlet pressure when the delivery pressure is increased from 870 to 950 psig.

All parameters in case A are the same except for the delivery pressure. The increased delivery pressure will cause the compressibility factor to change slightly due to the change in average pressure. However, for simplicity, we will assume $Z=0.8666$, as before.

The new delivery pressure is

$$
P_{2}=950+14.7=964.7 \mathrm{psia}
$$

Substituting in General Flow Equation 2.4, we get

$$
100 \times 10^{6}=38.77 \times 19.1954\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-964.7^{2}}{0.6 \times 520 \times 50 \times 0.8666}\right)^{0.5} 15.5^{2.5}
$$

Therefore,

$$
P_{1}=1071.77 \mathrm{psia}=1057.07 \mathrm{psig}
$$

Thus, the pressure required at the beginning of the pipeline is approximately 1057 psig. This compares with a value of 1066 psig we calculated if the pressure variation were linear. In general, for a gas pipeline, if the delivery pressure is increased by $\Delta P$, the inlet pressure will increase by less than $\Delta P$. Similarly, if the delivery pressure is decreased by $\Delta P$, the inlet pressure will decrease by less than $\Delta P$. We will illustrate this using the preceding example.

Suppose in case A, the delivery pressure was decreased from 870 to 800 psig. If pressure variation were linear, we would expect the pipe inlet pressure to decrease
by 70 psig to $(985.66-70)=916 \mathrm{psig}$, approximately. However, as indicated earlier, this is incorrect. We will now calculate the actual inlet pressure using the General Flow equation considering the reduced outlet pressure of 800 psig.

All parameters in case A are the same except for the delivery pressure. The decreased delivery pressure will cause the compressibility factor to change slightly due to the change in average pressure. However, for simplicity, we will assume $Z=0.8666$, as before.

$$
\text { New delivery pressure } P_{2}=800+14.7=814.7 \text { psia }
$$

Substituting in General Flow Equation 2.4, we get

$$
100 \times 10^{6}=38.77 \times 19.1954\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-814.7^{2}}{0.6 \times 520 \times 50 \times 0.8666}\right)^{0.5} 15.5^{2.5}
$$

Therefore,

$$
P_{1}=939.03 \mathrm{psia}=924.33 \mathrm{psig}
$$

Thus, the pressure required at the beginning of the pipeline is 924.33 psig . This compares with a value of 916 psig if the pressure variation were linear. Therefore, by decreasing the delivery pressure by 70 psig , the inlet pressure decreases by less than 70 psig.

We can perform a similar analysis by changing the inlet pressure by a fixed amount and calculating the effect on the pipe delivery pressure. As before, considering no elevation changes, an inlet pressure of 985.66 psig results in a delivery pressure of 870 psig. Suppose we decrease the inlet pressure to 900 psig (a reduction of 85.66 psig ); if pressures were linear, we would expect the delivery pressure to drop to $(870-85.66)=784.34$ psig. Actually, we will see that the delivery pressure would drop to a number lower than this. In other words, decreasing the inlet pressure by $\Delta P$ reduces the outlet pressure by more than $\Delta P$.

Following the previous methodology, we calculate the revised delivery pressure by assuming the same $Z=0.8666$ for simplicity.

$$
\text { New inlet pressure }=900+14.7=914.7 \mathrm{psia}
$$

Substituting in General Flow Equation 2.4, we calculate the outlet pressure as

$$
100 \times 10^{6}=38.77 \times 19.1954\left(\frac{520}{14.7}\right)\left(\frac{914.7^{2}-P_{2}^{2}}{0.6 \times 520 \times 50 \times 0.8666}\right)^{0.5} 15.5^{2.5}
$$

Therefore,

$$
P_{2}=786.54 \mathrm{psia}=771.84 \mathrm{psig}
$$

Thus, the delivery pressure is reduced by $(870-771.84)=98.16 \mathrm{psig}$, whereas the inlet pressure was reduced by only 85.66 psig.

In general, if the inlet pressure is decreased by $\Delta P$, the delivery pressure will decrease by more than $\Delta P$. On the other hand, if the inlet pressure is increased by $\Delta P$, the delivery pressure will increase by more than $\Delta P$.

Therefore, if the inlet pressure is increased from 985.66 to 1085.66 psig (an increase of 100 psig ), the delivery pressure will increase from 870 psig to a number larger than 970 psig.

### 3.5 PIPELINE WITH INTERMEDIATE INJECTIONS AND DELIVERIES

A pipeline in which gas enters at the beginning of the pipeline and the same volume exits at the end of the pipeline is a pipeline with no intermediate injection or deliveries. When portions of the inlet volume are delivered at various points along the pipeline and the remaining volume is delivered at the end of the pipeline, we call this system a pipeline with intermediate delivery points. A more complex case with gas flow into the pipeline (injection) at various points along its length combined with deliveries at other points is shown in Figure 3.1. In such a pipeline system, the pressure required at the beginning point A will be calculated by considering the pipeline broken into segments $\mathrm{AB}, \mathrm{BC}$, etc.

Another piping system can consist of gas flow at the inlet of the pipeline along with multiple pipe branches making deliveries of gas, as shown in Figure 3.2.

In this system, pipe AB has a certain volume, $Q_{1}$, flowing through it. At point B , another pipeline, CB , brings in additional volumes resulting in a volume of $\left(Q_{1}+Q_{2}\right)$ flowing through section BD . At D , a branch pipe, DE , delivers a volume of $Q_{3}$ to a customer location, E . The remaining volume $\left(Q_{1}+Q_{2}-Q_{3}\right)$ flows from D to F through pipe segment DF to a customer location at F .

In the subsequent sections, we will analyze pipelines with intermediate flow deliveries, injections, as well as branch pipes, as shown in Figure 3.1 and Figure 3.2. The objectives in all cases will be to calculate the pressures and flow rates through the various pipe sections and to determine pipe sizes required to limit pressure drop in certain pipe segments.


Figure 3.1 Pipeline with injection and deliveries.


Figure 3.2 Pipeline with branches.

## Example 2

A 150 mi long natural gas pipeline consists of several injections and deliveries as shown in Figure 3.3. The pipeline is NPS 20, has 0.500 in . wall thickness, and has an inlet volume of 250 MMSCFD. At points B (milepost 20) and C (milepost 80), 50 MMSCFD and 70 MMSCFD, respectively, are delivered. At D (milepost 100), gas enters the pipeline at 60 MMSCFD. All streams of gas may be assumed to have a specific gravity of 0.65 and a viscosity of $8.0 \times 10^{-6} \mathrm{lb} / \mathrm{ft}-\mathrm{s}$. The pipe is internally coated (to reduce friction), resulting in an absolute roughness of $150 \mu \mathrm{in}$. Assume a constant gas flow temperature of $60^{\circ} \mathrm{F}$ and base pressure and base temperature of 14.7 psia and $60^{\circ} \mathrm{F}$, respectively. Use a constant compressibility factor of 0.85 throughout. Neglect elevation differences along the pipeline.
a) Using the AGA equation, calculate the pressures along the pipeline at points A, B, C, and D for a minimum delivery pressure of 300 psig at the terminus E. Assume a drag factor $=0.96$.
b) What diameter pipe will be required for section DE if the required delivery pressure at $E$ is increased to 500 psig ? The inlet pressure at A remains the same as calculated above.


Figure 3.3 Example pipeline with injection and deliveries.

## Solution

We will start calculations beginning with the last segment DE.
Pipe inside diameter $D=20-2 \times 0.500=19.00 \mathrm{in}$.

The flow rate in pipe DE is 190 MMSCFD.
Using Equation 2.34, the Reynolds number is

$$
R=0.0004778\left(\frac{14.7}{520}\right)\left(\frac{0.65 \times 190 \times 10^{6}}{8 \times 10^{-6} \times 19}\right)=10,974,469
$$

Next, calculate the two transmission factors required per AGA.

1) The fully turbulent transmission factor, using Equation 2.48, is

$$
F=4 \log _{10}\left(\frac{3.7 \times 19}{150 \times 10^{-6}}\right)=22.68
$$

2) The smooth pipe zone Von Karman transmission factor, using Equation 2.50, is

$$
F_{t}=4 \log _{10}\left(\frac{10,974,469}{F_{t}}\right)-0.6
$$

Solving for $F_{t}$ by iteration, we get

$$
F_{t}=22.18
$$

Therefore, for a partly turbulent flow zone, the transmission factor, using Equation 2.49, is

$$
F=4 \times 0.96 \log _{10}\left(\frac{10,974,469}{1.4125 \times 22.18}\right)=21.29
$$

Using the smaller of the two values, the AGA transmission factor is

$$
F=21.29
$$

Next, we use General Flow Equation 2.4 to calculate the upstream pressure $P_{1}$ at D, based on a given downstream pressure of 300 psig at E .

$$
190 \times 10^{6}=38.77 \times 21.29\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-314.7^{2}}{0.65 \times 520 \times 50 \times 0.85}\right)^{0.5} 19^{2.5}
$$

Solving for $P_{1}$, we get the pressure at D as

$$
P_{1}=587.11 \mathrm{psia}=572.41 \mathrm{psig}
$$

Next, we consider the pipe segment CD, which has a flow rate of 130 MMSCFD. We calculate the pressure at C using the downstream pressure at D calculated above.

To simplify calculation, we will use the same AGA transmission factor we calculated for segment DE. A more nearly correct solution will be to calculate the Reynolds number and the two transmission factors as we did for the segment DE. However, for simplicity, we will use $F=21.29$ for all pipe segments.

Applying General Flow Equation 2.4, we calculate the pressure $P_{1}$ at C as follows:

$$
130 \times 10^{6}=38.77 \times 21.29\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-587.11^{2}}{0.65 \times 520 \times 20 \times 0.85}\right)^{0.5}(19.0)^{2.5}
$$

Solving for $P_{1}$, we get the pressure at C as

$$
P_{1}=625.06 \mathrm{psia}=610.36 \mathrm{psig}
$$

Similarly, we calculate the pressure at $B$, considering the pipe segment $B C$ that flows 200 MMSCFD.

$$
200 \times 10^{6}=38.77 \times 21.29\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-625.06^{2}}{0.65 \times 520 \times 60 \times 0.85}\right)^{0.5}(19.0)^{2.5}
$$

Solving for $P_{1}$, we get the pressure at B as

$$
P_{1}=846.95 \mathrm{psia}=832.25 \mathrm{psig}
$$

Finally, for pipe segment AB that flows 250 MMSCFD , we calculate the pressure $P_{1}$ at A as follows:

$$
250 \times 10^{6}=38.77 \times 21.29\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-846.95^{2}}{0.65 \times 520 \times 20 \times 0.85}\right)^{0.5}(19.0)^{2.5}
$$

Solving for $P_{1}$, we get the pressure at A as

$$
P_{1}=942.04 \mathrm{psia}=927.34 \mathrm{psig}
$$

If we maintain the same inlet pressure, 927.34 psig , at A and increase the delivery pressure at E to 500 psig , we can determine the pipe diameter required for section DE by considering the same upstream pressure of 572.41 psig at D , as we calculated before.

Therefore, for segment DE,
Upstream pressure $P_{1}=572.41+14.7=587.11 \mathrm{psia}$
Downstream pressure $P_{2}=500+14.7=514.7$ psia
Using General Flow Equation 2.4, with the same AGA transmission factor as before, we get

$$
190 \times 10^{6}=38.77 \times 21.29\left(\frac{520}{14.7}\right)\left(\frac{587.11^{2}-514.7^{2}}{0.65 \times 520 \times 50 \times 0.85}\right)^{0.5}(D)^{2.5}
$$

Solving for the inside diameter $D$ of pipe DE, we get

$$
D=23.79 \mathrm{in} .
$$

The nearest standard pipe size is NPS 26 with 0.500 in . wall thickness. This will give an inside diameter of 25 in ., which is slightly more than the required minimum of 23.79 in. calculated above.

The wall thickness required for this pipe diameter and pressure will be dictated by the pipe material and is the subject of Chapter 6.

## Example 3

A pipeline 100 mi long transports natural gas from Corona to Beaumont. The gas has a specific gravity of 0.60 and a viscosity of $8 \times 10^{-6} \mathrm{lb} / \mathrm{ft}-\mathrm{s}$. What is the minimum pipe diameter required to flow 100 MMSCFD from Corona to Beaumont for a delivery pressure of 800 psig at Beaumont and inlet pressure of 1400 psig at Corona? The gas can be assumed constant at $60^{\circ} \mathrm{F}$, and the base pressure and base temperature are 14.7 psia and $60^{\circ} \mathrm{F}$, respectively. Use a constant value of 0.90 for the compressibility factor and a pipe roughness of $700 \mu \mathrm{in}$. Compare results using the AGA, ColebrookWhite, Panhandle B, and Weymouth equations. Use $95 \%$ pipeline efficiency. Neglect elevation differences along the pipeline.

How will the result change if the elevation at Corona is 100 ft and at Beaumont is 500 ft ?

## Solution

We will first use the AGA equation to determine the pipe diameter. Since the transmission factor F depends on the Reynolds number, which depends on the unknown pipe diameter, we will first assume a value of $F=20$.

From General Flow Equation 2.4, we get

$$
100 \times 10^{6}=38.77 \times 20.0\left(\frac{520}{14.7}\right)\left[\frac{1414.7^{2}-814.7^{2}}{0.6 \times 520 \times 100 \times 0.9}\right]^{0.5} \times(D)^{2.5}
$$

Solving for diameter $D$,

$$
D=12.28 \mathrm{in} .
$$

or NPS 12 with a 0.250 in . wall thickness, approximately.
Next, we will recalculate the transmission factor using this pipe size. Using NPS 12 with a 0.250 in . wall thickness,

$$
\text { Inside pipe diameter } D=12.75-2 \times 0.250=12.25 \text { in. }
$$

Calculating the Reynolds number from Equation 2.34, we get

$$
R=0.0004778\left(\frac{14.7}{520}\right)\left(\frac{0.6 \times 100 \times 10^{6}}{8 \times 10^{-6} \times 12.25}\right)=8,269,615
$$

The fully turbulent transmission factor, using Equation 2.48, is

$$
F=4 \log _{10}\left(\frac{3.7 \times 12.25}{0.0007}\right)=19.25
$$

For the smooth pipe zone, using Equation 2.50, the Von Karman transmission factor is

$$
F_{t}=4 \log _{10}\left(\frac{8,269,615}{F_{t}}\right)-0.6
$$

Solving for $F_{t}$ by iteration, we get

$$
F_{t}=21.72
$$

Using a drag factor of 0.96 , for partly turbulent flow, the transmission factor is, from Equation 2.49,

$$
F=4 \times 0.96 \times \log _{10}\left(\frac{8,269,615}{1.4125 \times 21.72}\right)=20.85
$$

Using the lower of the two values, the AGA transmission factor is

$$
F=19.25
$$

Using this value of $F$, we recalculate the minimum pipe diameter from General Flow Equation 2.4 as follows:

$$
100 \times 10^{6}=38.77 \times 19.25\left(\frac{520}{14.7}\right)\left[\frac{1414.7^{2}-814.7^{2}}{0.6 \times 520 \times 100 \times 0.9}\right]^{0.5} \times(D)^{2.5}
$$

Solving for diameter $D$,

$$
D=12.47 \mathrm{in} .
$$

We will not continue iteration any further, since the new diameter will not change the value of $F$ appreciably.

Therefore, based on the AGA equation, the pipe inside diameter required is 12.47 in .
Next, we calculate the transmission factor based on the Colebrook-White equation, assuming an inside diameter of 12.25 in . and the Reynolds number $=8,269,615$, calculated earlier.

Using Colebrook-White Equation 2.45, we get

$$
F=-4 \log _{10}\left[\frac{0.0007}{3.7 \times 12.25}+\frac{1.255 F}{8,269,615}\right]
$$

Solving for $F$ by successive iteration, we get the Colebrook-White transmission factor as

$$
F=18.95
$$

Using the General Flow equation with this Colebrook-White transmission factor, we calculate the diameter as follows:

$$
100 \times 10^{6}=38.77 \times 18.95\left(\frac{520}{14.7}\right)\left[\frac{1414.7^{2}-814.7^{2}}{0.6 \times 520 \times 100 \times 0.9}\right]^{0.5} \times(D)^{2.5}
$$

Solving for diameter $D$,

$$
D=12.55 \mathrm{in} .
$$

Recalculating the Reynolds number and transmission factor using the pipe inside diameter of 12.55 in ., we get

$$
R=8,071,935 \text { and } F=18.94
$$

Therefore, the new diameter required is by proportions, using the General Flow equation,

$$
\left(\frac{D}{12.55}\right)^{2.5}=\left(\frac{18.95}{18.94}\right)
$$

or $D=12.55$, approximately. There is no appreciable change in the diameter required.
Therefore, based on the Colebrook-White equation, the pipe inside diameter required is $D=12.55 \mathrm{in}$.

Next, we determine the diameter required using Panhandle B Equation 2.59 and a pipeline efficiency of 0.95 :

$$
100 \times 10^{6}=737 \times 0.95\left(\frac{520}{14.7}\right)^{1.02}\left(\frac{1414.7^{2}-814.7^{2}}{0.6^{0.961} \times 520 \times 100 \times 0.9}\right)^{0.51} D^{2.53}
$$

Solving for diameter $D$, we get

$$
D=11.93 \mathrm{in} .
$$

Therefore, based on the Panhandle B equation, the pipe inside diameter required is 11.93 in.

Next, we calculate the diameter required, using Weymouth Equation 2.52 and a pipeline efficiency of 0.95 :

$$
100 \times 10^{6}=433.5 \times 0.95\left(\frac{520}{14.7}\right)\left(\frac{1414.7^{2}-814.7^{2}}{0.6 \times 520 \times 100 \times 0.9}\right)^{0.5} D^{2.667}
$$

Solving for diameter $D$, we get

$$
D=13.30 \mathrm{in} .
$$

Therefore, based on the Weymouth equation, the pipe inside diameter required is 13.30 in .

In summary, the minimum pipe inside diameter required based on the various flow equations is as follows:

AGA - D = 12.47 in.
Colebrook-White - $\mathrm{D}=12.55 \mathrm{in}$.
Panhandle $B-D=11.93$ in.
Weymouth equation $-\mathrm{D}=13.30 \mathrm{in}$.
It can be seen that the Weymouth equation is the most conservative equation. The AGA and Colebrook-White equations predict almost the same pipe size, while Panhandle B predicts the smallest pipe size. To further illustrate the comparison of various pressure drop equations, refer to the discussion in Chapter 2 and Figure 2.5, which shows how the delivery pressure varies for a fixed flow rate and inlet pressure. Table 3.1 also summarizes the various pressure drop equations used in the gas pipeline industry.

Considering elevation effects, with a single slope from Corona (100 ft) to Beaumont $(500 \mathrm{ft})$, the elevation adjustment parameter is, from Equation 2.10,

$$
s=0.0375 \times 0.6\left(\frac{500-100}{520 \times 0.9}\right)=0.0192
$$

Table 3.1 Summary of Pressure Drop Equations

| Equation | Application |
| :--- | :--- |
| General Flow | Fundamental flow equation using friction or transmission factor; <br> used with Colebrook-White friction factor or AGA transmission factor |
| Colebrook-White | Friction factor calculated for pipe roughness and Reynolds number; <br> most popular equation for general gas transmission pipelines |
| Modified | Modified equation based on U.S. Bureau of Mines experiments; gives <br> higher pressure drop compared to original Colebrook equation |
| Colebrook-White | Transmission factor calculated for partially turbulent and fully turbulent <br> flow considering roughness, bend index, and Reynolds number |
| AGA | Panhandle equations do not consider pipe roughness; instead, an <br> efficiency factor is used; less conservative than Colebrook or AGA |
| Panhandle A |  |
| Panhandle B | Does not consider pipe roughness; uses an efficiency factor <br> used for high-pressure gas gathering systems; most conservative <br> equation that gives highest pressure drop for given flow rate |
| Weymouth | Does not consider pipe roughness; uses an efficiency factor used on <br> gas distribution piping |

Therefore, the equivalent length from Equation 2.9 is

$$
L e=100 \times \frac{e^{0.0192}-1}{0.0192}=100.97 \mathrm{mi}
$$

We will apply the elevation correction factor for the extreme cases (Weymouth and Panhandle B equations) that produce the largest and the smallest diameter, respectively.

From Weymouth Equation 2.52, we see that, keeping all other items the same, the diameter and pipe length are related by the following equation:

$$
\begin{gathered}
\frac{D^{2.667}}{\sqrt{L}}=\text { Constant } \\
\left(\frac{D}{13.3}\right)^{2.667}=\left(\frac{100.97}{100}\right)^{0.5}
\end{gathered}
$$

Solving for the pipe inside diameter $D$, we get

$$
D=13.32 \mathrm{in} .
$$

This is not an appreciable change from the previous value of 13.30 in .

Similarly, from Panhandle B Equation 2.59, we see that the pipe diameter and length are related by

$$
\begin{gathered}
\frac{D^{2.53}}{L^{0.51}}=\text { Constant } \\
\left(\frac{D}{11.93}\right)^{2.53}=\left(\frac{100.97}{100}\right)^{0.51}
\end{gathered}
$$

Solving for pipe inside diameter $D$, we get

$$
D=11.95
$$

This is not an appreciable change from the previous value of 11.93 in .

Therefore, considering elevation difference between Corona and Beaumont, the minimum pipe sizes required are as follows:

Panhandle $B-D=11.95$ in.
Weymouth equation - $\mathrm{D}=13.32 \mathrm{in}$.

We thus see that even with a 400 ft elevation difference, the pipe diameter does not change appreciably.

## Example 4

A natural gas distribution piping system consists of NPS 12 with 0.250 in. wall thickness, 24 mi long, as shown in Figure 3.4. At Yale, an inlet flow rate of 65 MMSCFD of natural gas enters the pipeline at $60^{\circ} \mathrm{F}$. At the Compton terminus, gas must be supplied at a flow rate of 30 MMSCFD at a minimum pressure of 600 psig . There are intermediate deliveries of 15 MMSCFD at milepost 10 and 20 MMSCFD at milepost 18 . What is the required inlet pressure at Yale? Use a constant friction factor of 0.01 throughout. The compressibility factor can be assumed to be 0.94 . The gas gravity and viscosity are 0.6 and $7 \times 10^{-6} \mathrm{lb} / \mathrm{ft}-\mathrm{s}$, respectively. Assume isothermal flow at $60^{\circ} \mathrm{F}$. The base temperature and base pressure are $60^{\circ} \mathrm{F}$ and 14.7 psia , respectively. If the delivery volume at B is increased to 30 MMSCFD and other deliveries remain the same, what increased pressure is required at Yale to maintain the same flow rate and delivery pressure at Compton? Neglect elevation differences along the pipeline.

## Solution

For each section of piping, such as $A B$, we must calculate the pressure drop due to friction at the appropriate flow rate and then determine the total pressure drop for the entire pipeline.

$$
\begin{aligned}
& \text { Inside diameter of pipe }=12.75-2 \times 0.250=12.25 \mathrm{in} . \\
& \qquad \text { Friction factor } f=0.01
\end{aligned}
$$



Figure 3.4 Yale to Compton gas distribution pipeline.

Therefore, the transmission factor, using Equation 2.42, is

$$
F=\frac{2}{\sqrt{0.01}}=20.00
$$

Using General Flow Equation 2.7, for the last pipe segment from milepost 18 to milepost 24 , we get

$$
30 \times 10^{6}=38.77 \times 20.0\left(\frac{520}{14.7}\right)\left[\frac{P_{C}^{2}-614.7^{2}}{0.6 \times 520 \times 6 \times 0.94}\right]^{0.5} \times(12.25)^{2.5}
$$

Solving for the pressure at C,

$$
P_{C}=620.88 \mathrm{psia}
$$

Next we will use this pressure $P_{C}$ to calculate the pressure $P_{B}$ for the 8 mi section of pipe segment BC flowing 50 MMSCFD.

Using General Flow Equation 2.7,

$$
50 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left[\frac{P_{B}^{2}-620.88^{2}}{0.6 \times 520 \times 8 \times 0.94}\right]^{0.5} \times(12.25)^{2.5}
$$

Solving for $P_{B}$, we get

$$
P_{B}=643.24 \mathrm{psia}
$$

Finally, we calculate the pressure $P_{1}$ at Yale by considering the 10 mi pipe segment from Yale to point B that flows 65 MMSCFD.

$$
65 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left[\frac{P_{1}^{2}-643.24^{2}}{0.6 \times 520 \times 10 \times 0.94}\right]^{0.5} \times(12.25)^{2.5}
$$

Solving for the pressure at Yale, we get

$$
P_{1}=688.09 \mathrm{psia}=673.39 \mathrm{psig}
$$

Therefore, the required inlet pressure at Yale is 673.39 psig.
When the delivery volume at B is increased from 15 to 30 MMSCFD and all other delivery volumes remain the same, the inlet flow rate at Yale will increase to $65+$ $15=80$ MMSCFD. If the delivery pressure at Compton is to remain the same as before, the pressures at B and C will also be the same as calculated before, since the flow rate in BC and CD are the same as before. Therefore, we can recalculate the
inlet pressure for the pipe section from Yale to point B considering a flow rate of 80 MMSCFD that causes a pressure of 643.24 psia at B.

Using General Flow Equation 2.7, the pressure $P_{1}$ at Yale is

$$
80 \times 10^{6}=38.77 \times 20.0\left(\frac{520}{14.7}\right)\left[\frac{P_{1}^{2}-643.24^{2}}{0.6 \times 520 \times 10 \times 0.94}\right]^{0.5} \times(12.25)^{2.5}
$$

Solving for the pressure at Yale,

$$
P_{1}=710.07 \mathrm{psia}=695.37 \mathrm{psig}
$$

Therefore, increasing the delivery volume at B by 15 MMSCFD causes the pressure at Yale to increase by approximately 22 psig.

### 3.6 SERIES PIPING

In the preceding discussions we assumed the pipeline to have the same diameter throughout its length. There are situations where a gas pipeline can consist of different pipe diameters connected together in a series. This is especially true when the different pipe segments are required to transport different volumes of gas, as shown in Figure 3.5.

In Figure 3.5, section AB with a diameter of 16 in . is used to transport a volume of 100 MMSCFD, and after making a delivery of 20 MMSCFD at B , the remainder of 80 MMSCFD flows through the 14 in . diameter pipe BC. At C, a delivery of 30 MMSCFD is made, and the balance volume of 50 MMSCFD is delivered to the terminus D through a 12 in. pipeline CD.

It is clear that the pipe section AB flows the largest volume ( 100 MMSCFD ), whereas the pipe segment CD transports the least volume ( 50 MMSCFD ). Therefore, segments AB and CD , for reasons of economy, should be of different pipe diameters, as indicated in Figure 3.5. If we maintained the same pipe diameter of 16 in. from A to D , it would be a waste of pipe material and, therefore, cost. Constant diameter


Figure 3.5 Series piping.
is used only when the same flow that enters the pipeline is also delivered at the end of the pipeline, with no intermediate injections or deliveries.

However, in reality, there is no way of determining ahead what the future delivery volumes will be along the pipeline. Hence, it is difficult to determine initially the different pipe sizes for each segment. Therefore, in many cases you will find that the same-diameter pipe is used throughout the entire length of the pipeline even though there are intermediate deliveries. Even with the same nominal pipe diameter, different pipe sections can have different wall thicknesses. Therefore, we have different pipe inside diameters for each pipe segment. Such wall thickness changes are made to compensate for varying pressures along the pipeline. The subject of pipe strength and its relation to pipe diameter and wall thickness is discussed in Chapter 6.

The pressure required to transport gas in a series pipeline from point $A$ to point $D$ in Figure 3.5 is calculated by considering each pipe segment such as $A B$ and $B C$ and applying the appropriate flow equation, such as the General Flow equation, for each segment, as illustrated in Example 5.

Another approach to calculating the pressures in series piping systems is to use the equivalent length concept. This method can be applied when the same uniform flow exists throughout the pipeline, with no intermediate deliveries or injections. We will explain this method of calculation for a series piping system with the same flow rate $Q$ through all pipe segments. Suppose the first pipe segment has an inside diameter $D_{1}$ and length $L_{1}$, followed by the second segment of inside diameter $D_{2}$ and length $L_{2}$ and so on. We calculate the equivalent length of the second pipe segment based on the diameter $D_{1}$ such that the pressure drop in the equivalent length matches that in the original pipe segment of diameter $D_{2}$. The pressure drop in diameter $D_{2}$ and length $L_{2}$ equals the pressure drop in diameter $D_{1}$ and equivalent length $L e_{2}$.

Thus, the second segment can be replaced with a piece of pipe of length $L e_{2}$ and diameter $D_{1}$. Similarly, the third pipe segment with diameter $D_{3}$ and length $L_{3}$ will be replaced with a piece of pipe of $L e_{3}$ and diameter $D_{1}$. Thus, we have converted the three segments of pipe in terms of diameter $D_{1}$ as follows:

Segment 1 - diameter $D_{1}$ and length $L_{1}$
Segment 2 - diameter $D_{1}$ and length $L e_{2}$
Segment 3 - diameter $D_{1}$ and length $L e_{3}$
For convenience, we picked the diameter $D_{1}$ of segment 1 as the base diameter to use, to convert from the other pipe sizes. We now have the series piping system reduced to one constant-diameter $\left(D_{1}\right)$ pipe of total equivalent length given by

$$
\begin{equation*}
L e=L_{1}+L e_{2}+L e_{3} \tag{3.1}
\end{equation*}
$$

The pressure required at the inlet of this series piping system can then be calculated based on diameter $D_{1}$ and length $L e$. We will now explain how the equivalent length is calculated.

Upon examining General Flow Equation 2.7, we see that for the same flow rate and gas properties, neglecting elevation effects, the pressure difference $\left(P_{1}{ }^{2}-P_{2}{ }^{2}\right)$ is
inversely proportional to the fifth power of the pipe diameter and directly proportional to the pipe length. Therefore, we can state that, approximately,

$$
\begin{equation*}
\Delta P_{s q}=\frac{C L}{D^{5}} \tag{3.2}
\end{equation*}
$$

where
$\Delta P_{s q}=$ difference in the square of pressures $\left(P_{1}{ }^{2}-P_{2}{ }^{2}\right)$ for the pipe segment
$C=$ a constant
$L \quad=$ pipe length
$D$ = pipe inside diameter
Actually, $C$ depends on the flow rate, gas properties, gas temperature, base pressure, and base temperature. Therefore, $C$ will be the same for all pipe segments in a series pipeline with constant flow rate. Hence, we regard $C$ as a constant for all pipe segments.

From Equation 3.2 we conclude that the equivalent length for the same pressure drop is proportional to the fifth power of the diameter. Therefore, in the series piping discussed in the foregoing, the equivalent length of the second pipe segment of diameter $D_{2}$ and length $L_{2}$ is

$$
\begin{equation*}
\frac{C L_{2}}{D_{2}^{5}}=\frac{C L e_{2}}{D_{1}^{5}} \tag{3.3}
\end{equation*}
$$

or

$$
\begin{equation*}
L e_{2}=L_{2}\left(\frac{D_{1}}{D_{2}}\right)^{5} \tag{3.4}
\end{equation*}
$$

Similarly, for the third pipe segment of diameter $D_{3}$ and length $L_{3}$, the equivalent length is

$$
\begin{equation*}
L e_{3}=L_{3}\left(\frac{D_{1}}{D_{3}}\right)^{5} \tag{3.5}
\end{equation*}
$$

Therefore, the total equivalent length $L e$ for all three pipe segments in terms of diameter $D_{1}$ is

$$
\begin{equation*}
L e=L_{1}+L_{2}\left(\frac{D_{1}}{D_{2}}\right)^{5}+L_{3}\left(\frac{D_{1}}{D_{3}}\right)^{5} \tag{3.6}
\end{equation*}
$$

It can be seen from Equation 3.6 that if $D_{1}=D_{2}=D_{3}$, the total equivalent length reduces to $\left(L_{1}+L_{2}+L_{3}\right)$, as expected.

We can now calculate the pressure drop for the series piping system, considering a single pipe of length $L e$ and uniform diameter $D_{1}$ flowing a constant volume $Q$. An example will illustrate the use of the equivalent length method.


Figure 3.6 Example problem-series piping.

## Example 5

A series piping system, shown in Figure 3.6, consists of 12 mi of NPS 16, 0.375 in . wall thickness connected to 24 mi of NPS $14,0.250 \mathrm{in}$. wall thickness and 8 miles of NPS 12, 0.250 in . wall thickness pipes. Calculate the inlet pressure required at the origin A of this pipeline system for a gas flow rate of 100 MMSCFD. Gas is delivered to the terminus B at a delivery pressure of 500 psig . The gas gravity and viscosity are 0.6 and $0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{s}$, respectively. The gas temperature is assumed constant at $60^{\circ} \mathrm{F}$. Use a compressibility factor of 0.90 and the General Flow equation with Darcy friction factor $=0.02$. The base temperature and base pressure are $60^{\circ} \mathrm{F}$ and 14.7 psia, respectively.

Compare results using the equivalent length method and with the more detailed method of calculating pressure for each pipe segment separately.

## Solution

Inside diameter of first pipe segment $=16-2 \times 0.375=15.25 \mathrm{in}$.
Inside diameter of second pipe segment $=14-2 \times 0.250=13.50 \mathrm{in}$.
Inside diameter of third pipe segment $=12.75-2 \times 0.250=12.25 \mathrm{in}$.
Using Equation 3.6, we calculate the equivalent length of the pipeline, considering NPS 16 as the base diameter:

$$
L e=12+24 \times\left(\frac{15.25}{13.5}\right)^{5}+8 \times\left(\frac{15.25}{12.25}\right)^{5}
$$

or

$$
L e=12+44.15+23.92=80.07 \mathrm{mi}
$$

Therefore, we will calculate the inlet pressure $P_{1}$ considering a single pipe from A to $B$ having a length of 80.07 mi and inside diameter of 15.25 in .

$$
\text { Outlet pressure }=500+14.7=514.7 \mathrm{psia}
$$

Using General Flow Equation 2.2, neglecting elevation effects and substituting given values, we get

$$
100 \times 10^{6}=77.54\left(\frac{1}{\sqrt{0.02}}\right)\left(\frac{520}{14.7}\right)\left[\frac{\left(P_{1}^{2}-514.7^{2}\right)}{0.6 \times 520 \times 80.07 \times 0.9}\right]^{0.5} 15.25^{2.5}
$$

or

$$
P_{1}^{2}-514.7^{2}=724,642.99
$$

Solving for the inlet pressure $P_{1}$, we get

$$
P_{1}=994.77 \mathrm{psia}=980.07 \mathrm{psig}
$$

Next, we will compare the preceding result, using the equivalent length method, with the more detailed calculation of treating each pipe segment separately and adding the pressure drops.

Consider the 8 mi pipe segment 3 first, since we know the outlet pressure at B is 500 psig . Therefore, we can calculate the pressure at the beginning of segment 3 using General Flow Equation 2.2, as follows:

$$
100 \times 10^{6}=77.54\left(\frac{1}{\sqrt{0.02}}\right)\left(\frac{520}{14.7}\right)\left[\frac{\left(P_{1}^{2}-514.7^{2}\right)}{0.6 \times 520 \times 8 \times 0.9}\right]^{0.5} 12.25^{2.5}
$$

Solving for the pressure $P_{1}$, we get

$$
P_{1}=693.83 \mathrm{psia}=679.13 \mathrm{psig}
$$

This is the pressure at the beginning of the pipe segment 3 , which is also the end of pipe segment 2.

Next, consider pipe segment 2 ( 24 mi of NPS 14 pipe) and calculate the upstream pressure $P_{1}$ required for a downstream pressure of 679.13 psig, calculated in the preceding section. Using General Flow Equation 2.2 for pipe segment 2, we get

$$
100 \times 10^{6}=77.54\left(\frac{1}{\sqrt{0.02}}\right)\left(\frac{520}{14.7}\right)\left[\frac{\left(P_{1}^{2}-693.83^{2}\right)}{0.6 \times 520 \times 24 \times 0.9}\right]^{0.5} 13.5^{2.5}
$$

Solving for the pressure $P_{1}$, we get

$$
P_{1}=938.58 \mathrm{psia}=923.88 \mathrm{psig}
$$

This is the pressure at the beginning of pipe segment 2 , which is also the end of pipe segment 1 .

Next, we calculate the inlet pressure $P_{1}$ of pipe segment 1 ( 12 mi of NPS 16 pipe) for an outlet pressure of 923.88 psig , just calculated. Using the General Flow equation for pipe segment 1 , we get

$$
100 \times 10^{6}=77.54\left(\frac{1}{\sqrt{0.02}}\right)\left(\frac{520}{14.7}\right)\left[\frac{\left(P_{1}^{2}-938.58^{2}\right)}{0.6 \times 520 \times 12 \times 0.9}\right]^{0.5} 15.25^{2.5}
$$

Solving for pressure $P_{1}$, we get

$$
P_{1}=994.75 \mathrm{psia}=980.05 \mathrm{psig}
$$

This compares well with the pressure of 980.07 psig we calculated earlier using the equivalent length method.

## Example 6

A natural gas pipeline consists of three different pipe segments connected in series, pumping the same uniform flow rate of $3.0 \mathrm{Mm}^{3} / \mathrm{day}$ at $20^{\circ} \mathrm{C}$. The first segment, DN 500 with 12 mm wall thickness, is 20 km long. The second segment is DN $400,10 \mathrm{~mm}$ wall thickness, and 25 km long. The last segment is DN 300, 6 mm wall thickness, and 10 km long. The inlet pressure is 8500 kPa . Assuming flat terrain, calculate the delivery pressure, using the General Flow equation and the Colebrook friction factor of 0.02. The gas gravity $=0.65$ and viscosity $=0.000119$ Poise. The compressibility factor $Z=0.9$. The base temperature $=15^{\circ} \mathrm{C}$ and base pressure $=101 \mathrm{kPa}$. Compare results using the equivalent length method as well as the method using individual pipe segment pressure drops.

## Solution

Inside diameter of first pipe segment $=500-2 \times 12=476 \mathrm{~mm}$
Inside diameter of second pipe segment $=400-2 \times 10=380 \mathrm{~mm}$
Inside diameter of last pipe segment $=300-2 \times 6=288 \mathrm{~mm}$
Equivalent length method:
Using Equation 3.6, we calculate the total equivalent length of the pipeline system based on the first segment diameter DN 500 as follows:

$$
L e=20+25 \times\left(\frac{500-2 \times 12}{400-2 \times 10}\right)^{5}+10 \times\left(\frac{500-2 \times 12}{300-2 \times 6}\right)^{5}
$$

or

$$
L e=20+77.10+123.33=220.43 \mathrm{~km}
$$

Thus, the given pipeline system can be considered equivalent to a single pipe DN $500,12 \mathrm{~mm}$ wall thickness, 220.43 km long.

The outlet pressure $P_{2}$ is calculated using General Flow Equation 2.3 as follows:

$$
3 \times 10^{6}=1.1494 \times 10^{-3}\left(\frac{15+273}{101}\right)\left[\frac{\left(8500^{2}-P_{2}^{2}\right)}{0.65 \times 293 \times 0.9 \times 0.02 \times 220.43}\right]^{0.5}(476)^{2.5}
$$

Solving for $P_{2}$, we get

$$
8500^{2}-P_{2}^{2}=25,908,801
$$

or

$$
P_{2}=6807 \mathrm{kPa} \text { (absolute) }
$$

We have assumed that the given inlet pressure is in absolute value. Therefore, the delivery pressure is 6807 kPa (absolute).

Next, we calculate the delivery pressure considering the three pipe segments treated separately. For the first pipe segment 20 km long, we calculate the outlet pressure $P_{2}$ at the end of the first segment as follows. Using General Flow Equation 2.3, we get

$$
3 \times 10^{6}=1.1494 \times 10^{-3}\left(\frac{15+273}{101}\right)\left[\frac{\left(8500^{2}-P_{2}^{2}\right)}{0.65 \times 293 \times 0.9 \times 0.02 \times 20}\right]^{0.5}(476)^{2.5}
$$

Solving for $P_{2}$, we get

$$
P_{2}=8361 \mathrm{kPa} \text { (absolute) }
$$

Thus, the pressure at the end of the first pipe segment or the beginning of the second segment is 8361 kPa (absolute).

Next, we repeat the calculation for the second pipe segment DN 400, 25 km long, using $P_{1}=8361 \mathrm{kPa}$ (absolute), to calculate $P_{2}$ :

$$
3 \times 10^{6}=1.1494 \times 10^{-3}\left(\frac{15+273}{101}\right)\left[\frac{\left(8361^{2}-P_{2}^{2}\right)}{0.65 \times 293 \times 0.9 \times 0.02 \times 25}\right]^{0.5}(380)^{2.5}
$$

Solving for $P_{2}$, we get

$$
P_{2}=7800 \mathrm{kPa} \text { (absolute) }
$$

This is the pressure at the end of the second pipe segment, which is also the inlet pressure for the third pipe segment.

Finally, we calculate the outlet pressure of the last pipe segment (DN 300, 10 km ) using $P_{1}=7800 \mathrm{kPa}$ (absolute) as follows:

$$
3 \times 10^{6}=1.1494 \times 10^{-3}\left(\frac{15+273}{101}\right)\left[\frac{\left(7800^{2}-P_{2}^{2}\right)}{0.65 \times 293 \times 0.9 \times 0.02 \times 10}\right]^{0.5}(288)^{2.5}
$$

Solving for $P_{2}$, we get

$$
P_{2}=6808 \mathrm{kPa} \text { (absolute) }
$$

Therefore, the delivery pressure is 6808 kPa (absolute).
This compares favorably with the value of 6807 kPa we calculated earlier using the equivalent length approach.

### 3.7 PARALLEL PIPING

Sometimes two or more pipes are connected such that the gas flow splits among the branch pipes and eventually combines downstream into a single pipe, as illustrated in Figure 3.7. Such a piping system is referred to as parallel pipes. It is also called a looped piping system, where each parallel pipe is known as a loop. The reason for installing parallel pipes or loops is to reduce pressure drop in a certain section of the pipeline due to pipe pressure limitation or for increasing the flow rate in a bottleneck section. By installing a pipe loop from B to E, in Figure 3.7 we are effectively reducing the overall pressure drop in the pipeline from A to F , since between B and E the flow is split through two pipes.

In Figure 3.7 we will assume that the entire pipeline system is in the horizontal plane with no changes in pipe elevations. Gas enters the pipeline at A and flows through the pipe segment AB at a flow rate of $Q$. At the junction B , the gas flow splits into the two parallel pipe branches BCE and BDE at the flow rates of $Q_{1}$ and $Q_{2}$, respectively. At E, the gas flows recombine to equal the initial flow rate $Q$ and continue flowing through the single pipe EF.

In order to calculate the pressure drop due to friction in the parallel piping system, we follow two main principles of parallel pipes. The first principle is that of conservation of flow at any junction point. The second principle is that there is a common pressure across each parallel pipe.


Figure 3.7 Parallel piping.

Applying the principle of flow conservation, at junction B , the incoming flow into B must exactly equal the total outflow at B through the parallel pipes.

Therefore, at junction B,

$$
\begin{equation*}
Q=Q_{1}+Q_{2} \tag{3.7}
\end{equation*}
$$

where
$Q=$ inlet flow at A
$Q_{1}=$ flow through pipe branch BCE
$Q_{2}=$ flow through pipe branch BDE
According to the second principle of parallel pipes, the pressure drop in pipe branch BCE must equal the pressure drop in pipe branch BDE . This is due to the fact that both pipe branches have a common starting point (B) and common ending point (E). Therefore, the pressure drop in the branch pipe BCE and branch pipe BDE are each equal to $\left(P_{B}-P_{E}\right)$, where $P_{B}$ and $P_{E}$ are the pressures at junctions B and E , respectively.

Therefore, we can write

$$
\begin{equation*}
\Delta P_{B C E}=\Delta P_{B D E}=P_{B}-P_{E} \tag{3.8}
\end{equation*}
$$

$\Delta P$ represents pressure drop, and $\Delta P_{B C E}$ is a function of the diameter and length of branch BCE and the flow rate $Q_{1}$. Similarly, $\Delta P_{B D E}$ is a function of the diameter and length of branch BDE and the flow rate $Q_{2}$.

In order to calculate the pressure drop in parallel pipes, we must first determine the flow split at junction B. From Equation 3.7, we know that the sum of the two flow rates $Q_{1}$ and $Q_{2}$ must equal the given inlet flow rate $Q$. If both pipe loops BCE and BDE are equal in length and pipe inside diameter, we can infer that the flow rate will be split equally between the two branches.

Thus, for identical pipe loops,

$$
\begin{equation*}
Q_{1}=Q_{2}=\frac{Q}{2} \tag{3.9}
\end{equation*}
$$

In this case, the pressure drop from B to E can be calculated assuming a flow rate of $\frac{Q}{2}$ flowing through one of the pipe loops.

To illustrate this further, suppose we are interested in determining the pressure at A for the given flow rate $Q$ and a specified delivery pressure $\left(P_{F}\right)$ at the pipe terminus F . We start with the last pipe segment EF and calculate the pressure required at E for a flow rate of $Q$ in order to deliver gas at F at a pressure $P_{F}$. We could use the General Flow equation for this and substitute $P_{E}$ for upstream pressure, $P_{1}$, and $P_{F}$ for downstream pressure $P_{2}$. Having calculated $P_{E}$, we can now consider one of the pipe loops, such as BCE, and calculate the upstream pressure $P_{B}$ required for a flow rate of $\frac{Q}{2}$ through BCE for a downstream pressure of $P_{E}$. In the General Flow equation, the upstream pressure $P_{1}=P_{B}$ and the downstream pressure $P_{2}=P_{E}$.

It must be noted that this is correct only for identical pipe loops. Otherwise, the flow rate $Q_{1}$ and $Q_{2}$ through the pipe branches BCE and BDE will be unequal. From the
calculated value of $P_{B}$, we can now determine the pressure required at A by applying the General Flow equation to pipe segment AB that has a gas flow rate of $Q$. The upstream pressure $P_{1}$ will be calculated for a downstream pressure $P_{2}=P_{B}$.

Consider now a situation in which the pipe loops are not identical. This means that the pipe branches BCE and BDE can have different lengths and different diameters. In this case, we must determine the flow split between these two branches by equating the pressure drops through each of the branches in accordance with Equation 3.8. Since $Q_{1}$ and $Q_{2}$ are two unknowns, we will use the flow conservation principle and the common pressure drop principle to determine the values of $Q_{1}$ and $Q_{2}$. From the General Flow equation we can state the following:

The pressure drop due to friction in branch BCE can be calculated from

$$
\begin{equation*}
\left(P_{B}^{2}-P_{E}^{2}\right)=\frac{K_{1} L_{1} Q_{1}^{2}}{D_{1}^{5}} \tag{3.10}
\end{equation*}
$$

where
$K_{1}=$ a parameter that depends on gas properties, gas temperature, etc.
$L_{1}=$ length of pipe branch BCE
$D_{1}=$ inside diameter of pipe branch BCE
$Q_{1}=$ flow rate through pipe branch BCE
Other symbols are as defined earlier.
$K_{1}$ is a parameter that depends on the gas properties, gas temperature, base pressure, and base temperature that will be the same for both pipe branches BCE and BDE in a parallel pipeline system. Hence, we regard this as a constant from branch to branch.

Similarly, the pressure drop due to friction in branch BDE is calculated from

$$
\begin{equation*}
\left(P_{B}^{2}-P_{E}^{2}\right)=\frac{K_{2} L_{2} Q_{2}^{2}}{D_{2}^{5}} \tag{3.11}
\end{equation*}
$$

where
$K_{2}=$ a constant like $K_{1}$
$L_{2}=$ length of pipe branch BDE
$D_{2}=$ inside diameter of pipe branch BDE
$Q_{2}=$ flow rate through pipe branch BDE

Other symbols are as defined earlier.
In Equation 3.10 and Equation 3.11, the constants $K_{1}$ and $K_{2}$ are equal, since they do not depend on the diameter or length of the branch pipes BCE and BDE. Combining both equations, we can state the following for common pressure drop through each branch:

$$
\begin{equation*}
\frac{L_{1} Q_{1}^{2}}{D_{1}^{5}}=\frac{L_{2} Q_{2}^{2}}{D_{2}^{5}} \tag{3.12}
\end{equation*}
$$

Simplifying further, we get the following relationship between the two flow rates $Q_{1}$ and $Q_{2}$ :

$$
\begin{equation*}
\frac{Q_{1}}{Q_{2}}=\left(\frac{L_{2}}{L_{1}}\right)^{0.5}\left(\frac{D_{1}}{D_{2}}\right)^{2.5} \tag{3.13}
\end{equation*}
$$

Combining Equation 3.13 with Equation 3.7, we can solve for the flow rates $Q_{1}$ and $Q_{2}$. To illustrate this, consider the inlet flow $Q=100 \mathrm{MMSCFD}$ and the pipe branches as follows:

$$
\begin{array}{ll}
L_{1}=10 \mathrm{mi} & D_{1}=15.5 \mathrm{in} . \text { for branch BCE } \\
L_{2}=15 \mathrm{mi} & D_{2}=13.5 \mathrm{in} . \text { for branch BDE }
\end{array}
$$

From Equation 3.7, for flow conservation, we get

$$
Q_{1}+Q_{2}=100
$$

From Equation 3.13, we get the ratio of flow rates as

$$
\frac{Q_{1}}{Q_{2}}=\left(\frac{15}{10}\right)^{0.5}\left(\frac{15.5}{13.5}\right)^{2.5}=1.73
$$

Solving these two equations in $Q_{1}$ and $Q_{2}$, we get

$$
\begin{aligned}
& Q_{1}=63.37 \mathrm{MMSCFD} \\
& Q_{2}=36.63 \mathrm{MMSCFD}
\end{aligned}
$$

Once we know the values of $Q_{1}$ and $Q_{2}$, we can easily calculate the common pressure drop in the branch pipes BCE and BDE. An example problem (Example 7) will be used to illustrate this method.

Another method of calculating pressure drops in parallel pipes is using the equivalent diameter. In this method, we replace the pipe loops BCE and BDE with a certain length of an equivalent diameter pipe that has the same pressure drop as one of the branch pipes. The equivalent diameter pipe can be calculated using the General Flow equation, as explained next. The equivalent pipe with the same $\Delta P$ that will replace both branches will have a diameter $D_{e}$ and a length equal to one of the branch pipes, say $L_{1}$.

Since the pressure drop in the equivalent diameter pipe, which flows the full volume $Q$, is the same as that in any of the branch pipes, from Equation 3.10, we can state the following:

$$
\begin{equation*}
\left(P_{B}^{2}-P_{E}^{2}\right)=\frac{K_{e} L_{e} Q^{2}}{D_{e}^{5}} \tag{3.14}
\end{equation*}
$$

where $Q=Q_{1}+Q_{2}$ from Equation 3.7 and $K_{e}$ represents the constant for the equivalent diameter pipe of length $L_{e}$ flowing the full volume $Q$.

Equating the value of $\left(P_{B}{ }^{2}-P_{E}{ }^{2}\right)$ to the corresponding values, considering each branch separately, we get, using Equation 3.10, Equation 3.11, and Equation 3.14:

$$
\begin{equation*}
\frac{K_{1} L_{1} Q_{1}^{2}}{D_{1}^{5}}=\frac{K_{2} L_{2} Q_{2}^{2}}{D_{2}^{5}}=\frac{K_{e} L_{e} Q^{2}}{D_{e}^{5}} \tag{3.15}
\end{equation*}
$$

Also, setting $K_{1}=K_{2}=K_{e}$ and $L_{e}=L_{1}$, we simplify Equation 3.15 as follows:

$$
\begin{equation*}
\frac{L_{1} Q_{1}^{2}}{D_{1}^{5}}=\frac{L_{2} Q_{2}^{2}}{D_{2}^{5}}=\frac{L_{1} Q^{2}}{D_{e}^{5}} \tag{3.16}
\end{equation*}
$$

Using Equation 3.16 in conjunction with Equation 3.7, we solve for the equivalent diameter $D_{e}$ as

$$
\begin{equation*}
D_{e}=D_{1}\left[\left(\frac{1+\text { Const } 1}{\text { Const } 1}\right)^{2}\right]^{1 / 5} \tag{3.17}
\end{equation*}
$$

where

$$
\begin{equation*}
\text { Const } 1=\sqrt{\left(\frac{D_{1}}{D_{2}}\right)^{5}\left(\frac{L_{2}}{L_{1}}\right)} \tag{3.18}
\end{equation*}
$$

and the individual flow rates $Q_{1}$ and $Q_{2}$ are calculated from

$$
\begin{equation*}
Q_{1}=\frac{Q \text { Const } 1}{1+\text { Const } 1} \tag{3.19}
\end{equation*}
$$

and

$$
\begin{equation*}
Q_{2}=\frac{Q}{1+\text { Const } 1} \tag{3.20}
\end{equation*}
$$

To illustrate the equivalent diameter method, consider the inlet flow $Q=100$ MMSCFD and the pipe branches as follows:

$$
\begin{array}{ll}
L_{1}=10 \mathrm{mi} & D_{1}=15.5 \mathrm{in} . \text { for branch BCE } \\
L_{2}=15 \mathrm{mi} & D_{2}=13.5 \mathrm{in} . \text { for branch BDE }
\end{array}
$$

From Equation 3.18,

$$
\text { Const } 1=\sqrt{\left(\frac{15.5}{13.5}\right)^{5}\left(\frac{15}{10}\right)}=1.73
$$

Using Equation 3.17, the equivalent diameter is

$$
D_{e}=15.5\left[\left(\frac{1+1.73}{1.73}\right)^{2}\right]^{1 / 5}=18.60 \mathrm{in}
$$

Thus, the NPS 16 and NPS 14 pipes in parallel can be replaced with an equivalent pipe having an inside diameter of 18.6 in .

From Equation 3.19 and Equation 3.20, we get the flow rates in the two branch pipes as follows:

$$
Q_{1}=\frac{100 \times 1.73}{1+1.73}=63.37 \mathrm{MMSCFD}
$$

and

$$
Q_{2}=36.63 \mathrm{MMSCFD}
$$

Having calculated an equivalent diameter $D_{e}$, we can now calculate the common pressure drop in the parallel branches by considering the entire flow $Q$ flowing through the equivalent diameter pipe. An example problem will illustrate this method.

## Example 7

A gas pipeline consists of two parallel pipes, as shown in Figure 3.7. It is designed to operate at a flow rate of 100 MMSCFD. The first pipe segment AB is 12 miles long and consists of NPS $16,0.250 \mathrm{in}$. wall thickness pipe. The loop BCE is 24 mi long and consists of NPS 14, 0.250 in . wall thickness pipe. The loop BDE is 16 miles long and consists of NPS 12, 0.250 in. wall thickness pipe. The last segment EF is 20 miles long and consists of NPS $16,0.250 \mathrm{in}$. wall thickness pipe. Assuming a gas gravity of 0.6 , calculate the outlet pressure at F and the pressures at the beginning and the end of the pipe loops and the flow rates through them. The inlet pressure at $\mathrm{A}=1200 \mathrm{psig}$. The gas flowing temperature $=80^{\circ} \mathrm{F}$, base temperature $=60^{\circ} \mathrm{F}$, and base pressure $=$ 14.73 psia. The compressibility factor $Z=0.92$. Use the General Flow equation with Colebrook friction factor $f=0.015$.

## Solution

From Equation 3.13, the ratio of the flow rates through the two pipe loops is given by

$$
\frac{Q_{1}}{Q_{2}}=\left(\frac{16}{24}\right)^{0.5}\left(\frac{14-2 \times 0.25}{12.75-2 \times 0.25}\right)^{2.5}=1.041
$$

and from Equation 3.7

$$
Q_{1}+Q_{2}=100
$$

Solving for $Q_{1}$ and $Q_{2}$, we get

$$
Q_{1}=51.0 \mathrm{MMSCFD} \text { and } Q_{2}=49.0 \mathrm{MMSCFD}
$$

Next, considering the first pipe segment $A B$, we will calculate the pressure at $B$ based upon the inlet pressure of 1200 psig at A, using General Flow Equation 2.2, as follows:

$$
100 \times 10^{6}=77.54\left(\frac{1}{\sqrt{0.015}}\right)\left(\frac{520}{14.73}\right)\left[\frac{\left(1214.73^{2}-P_{2}^{2}\right)}{0.6 \times 540 \times 12 \times 0.92}\right]^{0.5} 15.5^{2.5}
$$

Solving for the pressure at B, we get

$$
P_{2}=1181.33 \mathrm{psia}=1166.6 \mathrm{psig}
$$

This is the pressure at the beginning of the looped section at B. Next we calculate the outlet pressure at E of pipe branch BCE, considering a flow rate of 51 MMSCFD through the NPS 14 pipe, starting at a pressure of 1181.33 psia at B.

Using the General Flow equation, we get

$$
51 \times 10^{6}=77.54\left(\frac{1}{\sqrt{0.015}}\right)\left(\frac{520}{14.73}\right)\left[\frac{\left(1181.33^{2}-P_{2}^{2}\right)}{0.6 \times 540 \times 24 \times 0.92}\right]^{0.5} 13.5^{2.5}
$$

Solving for the pressure at E , we get

$$
P_{2}=1145.63 \mathrm{psia}=1130.9 \mathrm{psig}
$$

Next, we use this pressure as the inlet pressure for the last pipe segment EF and calculate the outlet pressure at F using the General Flow equation, as follows:

$$
100 \times 10^{6}=77.54\left(\frac{1}{\sqrt{0.015}}\right)\left(\frac{520}{14.73}\right)\left[\frac{\left(1145.63^{2}-P_{2}^{2}\right)}{0.6 \times 540 \times 20 \times 0.92}\right]^{0.5} 15.5^{2.5}
$$

Solving for the outlet pressure at F, we get

$$
P_{2}=1085.85 \mathrm{psia}=1071.12 \mathrm{psig}
$$

In summary, the calculated results are as follows:
Pressure at the beginning of pipe loops $=1166.6 \mathrm{psig}$
Pressure at the end of pipe loops $=1130.9 \mathrm{psig}$
Outlet pressure at the end of pipeline $=1071.12$ psig
Flow rate in NPS 14 loop $=51$ MMSCFD
Flow rate in NPS 12 loop $=49$ MMSCFD

We will now calculate the pressures using the equivalent diameter method.
From Equation 3.18,

$$
\text { Const } 1=\sqrt{\left(\frac{13.5}{12.25}\right)^{5}\left(\frac{16}{24}\right)}=1.041
$$

From Equation 3.17, the equivalent diameter is

$$
D_{e}=13.5\left[\left(\frac{1+1.041}{1.041}\right)^{2}\right]^{1 / 5}=17.67 \mathrm{in} .
$$

Thus, we can replace the two branch pipes between B and E with a single piece of pipe 24 mi long, having an inside diameter of 17.67 in ., flowing 100 MMSCFD.

The pressure at B was calculated earlier as

$$
P_{B}=1181.33 \mathrm{psia}
$$

Using this pressure, we can calculate the downstream pressure at E for the equivalent pipe diameter as follows:

$$
100 \times 10^{6}=77.54\left(\frac{1}{\sqrt{0.015}}\right)\left(\frac{520}{14.73}\right)\left[\frac{\left(1181.33^{2}-P_{2}^{2}\right)}{0.6 \times 540 \times 24 \times 0.92}\right]^{0.5} 17.67^{2.5}
$$

Solving for the outlet pressure at E , we get

$$
P_{2}=1145.60 \mathrm{psia},
$$

which is almost the same as what we calculated before.

The pressure at F will therefore be the same as what we calculated before.
Therefore, using the equivalent diameter method, the parallel pipes BCE and BDE can be replaced with a single pipe 24 mi long, having an inside diameter of 17.67 in .

## Example 8

A natural gas pipeline DN 500 with 12 mm wall thickness is 60 km long. The gas flow rate is $5.0 \mathrm{Mm}^{3} /$ day at $20^{\circ} \mathrm{C}$. Calculate the inlet pressure required for a delivery pressure of 4 MPa (absolute), using the General Flow equation with the modified Colebrook-White friction factor. The pipe roughness $=0.015 \mathrm{~mm}$. In order to increase the flow rate through the pipeline, the entire line is looped with a DN 500 pipeline, 12 mm wall thickness. Assuming the same delivery pressure, calculate the inlet pressure at the new flow rate of $8 \mathrm{Mm}^{3} /$ day. The gas gravity $=0.65$ and viscosity $=$ 0.000119 Poise. The compressibility factor $Z=0.88$. The base temperature $=15^{\circ} \mathrm{C}$, and the base pressure $=101 \mathrm{kPa}$. If the inlet and outlet pressures are held the same as before, what length of the pipe should be looped to achieve the increased flow?

## Solution

Pipe inside diameter $D=500-2 \times 12=476 \mathrm{~mm}$
Flow rate $Q=5.0 \times 10^{6} \mathrm{~m}^{3} /$ day
Base temperature $T_{b}=15+273=288 \mathrm{~K}$
Gas flow temperature $T_{f}=20+273=293 \mathrm{~K}$
Delivery pressure $P_{2}=4 \mathrm{MPa}$
The Reynolds number, using Equation 2.35, is

$$
R=0.5134\left(\frac{101}{288}\right)\left(\frac{0.65 \times 5 \times 10^{6}}{0.000119 \times 476}\right)=10,330,330
$$

From the modified Colebrook-White Equation 2.47, the transmission factor is

$$
F=-4 \log _{10}\left(\frac{0.015}{3.7 \times 476}+\frac{1.4125 F}{10,330,330}\right)
$$

Solving by successive iteration, we get

$$
F=19.80
$$

Using General Flow Equation 2.8, the inlet pressure is calculated next:

$$
5 \times 10^{6}=5.747 \times 10^{-4} \times 19.80\left(\frac{273+15}{101}\right)\left[\frac{P_{1}^{2}-4000^{2}}{0.65 \times 293 \times 60 \times 0.88}\right]^{0.5} \times(476)^{2.5}
$$

Solving for the inlet pressure, we get

$$
\left.P_{1}=5077 \mathrm{kPa}(\text { absolute })=5.08 \mathrm{MPa} \text { (absolute }\right)
$$

Therefore, the inlet pressure required at $5 \mathrm{Mm}^{3} /$ day flow rate is 5.08 MPa .
Next, at $8 \mathrm{Mm}^{3} /$ day flow rate, we calculate the new inlet pressure with the entire 60 km length looped with an identical DN 500 pipe. Since the loop is the same size as the main line, each parallel branch will carry half the total flow rate or $4 \mathrm{Mm}^{3} / \mathrm{day}$.

We calculate the Reynolds number for flow through one of the loops using Equation 2.35:

$$
R=0.5134\left(\frac{101}{288}\right)\left(\frac{0.65 \times 4 \times 10^{6}}{0.000119 \times 476}\right)=8,264,264
$$

From the modified Colebrook-White Equation 2.47, the transmission factor is

$$
F=-4 \log _{10}\left(\frac{0.015}{3.7 \times 476}+\frac{1.4125 F}{8,264,264}\right)
$$

Solving by successive iteration, we get

$$
F=19.70
$$

Keeping the delivery pressure the same as before ( 4 MPa ), using General Flow Equation 2.8, we calculate the inlet pressure required as follows:

$$
4 \times 10^{6}=5.747 \times 10^{-4} \times 19.70\left(\frac{273+15}{101}\right)\left[\frac{P_{1}^{2}-4000^{2}}{0.65 \times 293 \times 60 \times 0.88}\right]^{0.5} \times(476)^{2.5}
$$

Solving for the inlet pressure, we get

$$
\left.P_{1}=4724 \mathrm{kPa}(\text { absolute })=4.72 \mathrm{MPa} \text { (absolute }\right)
$$

Therefore, for the fully looped pipeline at $8 \mathrm{Mm}^{3} /$ day flow rate, the inlet pressure required is 4.72 MPa .

Next, keeping the inlet and outlet pressures the same at 5077 kPa and 4000 kPa , respectively, at the new flow rate of $8 \mathrm{Mm}^{3} /$ day, we assume $L \mathrm{~km}$ of the pipe from the inlet is looped. We will calculate the value of $L$ by first calculating the pressure at the point where the loop ends. Since each parallel pipe carries $4 \mathrm{Mm}^{3} /$ day, we use the Reynolds number and transmission factor calculated earlier:

$$
R=8,264,264 \quad \text { and } \quad F=19.70
$$

Using General Flow Equation 2.8, we calculate the outlet pressure at the end of the loop of length $L \mathrm{~km}$ as follows:

$$
4 \times 10^{6}=5.747 \times 10^{-4} \times 19.70\left(\frac{273+15}{101}\right)\left[\frac{5077^{2}-P_{2}^{2}}{0.65 \times 293 \times L \times 0.88}\right]^{0.5} \times(476)^{2.5}
$$

Solving for pressure in terms of the loop length $L$, we get

$$
\begin{equation*}
P_{2}^{2}=5077^{2}-105,291.13 L \tag{3.21}
\end{equation*}
$$

Next, we apply the General Flow equation for the pipe segment of length $(60-L) \mathrm{km}$ that carries the full $8 \mathrm{Mm}^{3} /$ day flow rate. The inlet pressure is $P_{2}$ and the outlet pressure is 4000 kPa .

The Reynolds number at $8 \mathrm{Mm}^{3} /$ day is

$$
R=0.5134\left(\frac{101}{288}\right)\left(\frac{0.65 \times 8 \times 10^{6}}{0.000119 \times 476}\right)=16,528,528
$$

From the modified Colebrook-White Equation 2.47, the transmission factor is

$$
F=-4 \log _{10}\left(\frac{0.015}{3.7 \times 476}+\frac{1.4125 F}{16,528,528}\right)
$$

Solving by successive iteration, we get

$$
F=19.96
$$

Using General Flow Equation 2.8, we calculate the inlet pressure for the pipe segment of length $(60-L) \mathrm{km}$ as follows:

$$
8 \times 10^{6}=5.747 \times 10^{-4} \times 19.96\left(\frac{273+15}{101}\right)\left[\frac{P_{2}^{2}-4000^{2}}{0.65 \times 293 \times(60-L) \times 0.88}\right]^{0.5} \times(476)^{2.5}
$$

Simplifying, we get

$$
\begin{equation*}
P_{2}^{2}=4000^{2}+410,263.77(60-L) \tag{3.22}
\end{equation*}
$$

From Equation 3.21 and Equation 3.22, eliminating $P_{2}$, we solve for $L$ as follows:

$$
5077^{2}-105,291.13 L=4000^{2}+410,263.77(60-L)
$$

Therefore,

$$
L=48.66 \mathrm{~km}
$$

Thus, 48.66 km out of the 60 km pipeline length will have to be looped starting at the pipe inlet so that at $8 \mathrm{Mm}^{3} /$ day both inlet and outlet pressures will be the same as before at $5 \mathrm{Mm}^{3} /$ day .

What will be the effect if the loop was installed starting at the downstream end of the pipeline and proceeding toward the upstream end? Will the results be the same? In the next section we will explore the best location to install the pipe loop.

### 3.8 LOCATING PIPE LOOP

In the preceding example, we looked at looping an entire pipeline to reduce pressure drop and increase the flow rate. We also explored looping a portion of the pipe, beginning at the upstream end. How do we determine where the loop should be placed for optimum results? Should it be located upstream, downstream, or in a midsection of the pipe? We will analyze this in this section. Three looping scenarios are presented in Figure 3.8.

In case (a), a pipeline of length $L$ is shown looped with $X$ miles of pipe, beginning at the upstream end A. In case (b), the same length $X$ of pipe is looped, but it is located on the downstream end B. Case (c) shows the midsection of the pipeline being looped. For most practical purposes, we can say that the cost of all three loops will be the same as long as the loop length is the same.


Figure 3.8 Different looping scenarios.

In order to determine which of these cases is optimum, we must analyze how the pressure drop in the pipeline varies with distance from the pipe inlet to outlet. It is found that if the gas temperature is constant throughout, at locations near the upstream end, the pressure drops at a slower rate than at the downstream end. Therefore, there is more pressure drop in the downstream section compared to that in the upstream section. Hence, to reduce the overall pressure drop, the loop must be installed toward the downstream end of the pipe. This argument is valid only if the gas temperature is constant throughout the pipeline. In reality, due to heat transfer between the flowing gas and the surrounding soil (buried pipe) or the outside air (above-ground pipe), the gas temperature will change along the length of the pipeline. If the gas temperature at the pipe inlet is higher than that of the surrounding soil (buried pipe), the gas will lose heat to the soil and the temperature will drop from the pipe inlet to the pipe outlet. If the gas is compressed at the inlet using a compressor, then the gas temperature will be much higher than that of the soil immediately downstream of the compressor. The hotter gas will cause higher pressure drops (examine the General Flow equation and see how the pressure varies with the gas flow temperature). Hence, in this case the upstream segment will have a larger pressure drop compared to the downstream segment. Therefore, considering heat transfer effects, the pipe loop should be installed in the upstream portion for maximum benefit. The installation of the pipe loop in the midsection of the pipeline, as in case (c) in Figure 3.8, will not be the optimum location, based on the preceding discussion. It can therefore be concluded that if the gas temperature is fairly constant along the pipeline, the loop should be installed toward the downstream end, as in case (b). If heat transfer is taken into account and the gas temperature varies along the pipeline, with the hotter gas being upstream, the better location for the pipe loop will be on the upstream end, as in case (a).

Looping pipes will be explored more in Chapter 5 and in Chapter 10, where we discuss pipeline economics.


Figure 3.9 Hydraulic pressure gradient for uniform flow.

### 3.9 HYDRAULIC PRESSURE GRADIENT

The hydraulic pressure gradient is a graphical representation of the gas pressures along the pipeline, as shown in Figure 3.9. The horizontal axis shows the distance along the pipeline starting at the upstream end. The vertical axis depicts the pipeline pressures.

Since pressure in a gas pipeline is nonlinear compared to liquid pipelines, the hydraulic gradient for a gas pipeline appears to be a slightly curved line instead of a straight line. The slope of the hydraulic gradient at any point represents the pressure loss due to friction per unit length of pipe. As discussed earlier, this slope is more pronounced as we move toward the downstream end of the pipeline, since the pressure drop is larger toward the end of the pipeline. If the flow rate through the pipeline is a constant value (no intermediate injections or deliveries) and pipe size is uniform throughout, the hydraulic gradient appears to be a slightly curved line, as shown in Figure 3.9, with no appreciable breaks. If there are intermediate deliveries or injections along the pipeline, the hydraulic gradient will be a series of broken lines, as indicated in Figure 3.10.


Figure 3.10 Hydraulic pressure gradient for deliveries and injections.

A similar broken hydraulic gradient can also be seen in the case of a pipeline with variable pipe diameters and wall thicknesses, even if the flow rate is constant. Unlike liquid pipelines, the breaks in hydraulic pressure gradient are not as conspicuous in gas pipelines.

In a long-distance gas pipeline, due to limitations of pipe pressure, intermediate compressor stations will be installed to boost the gas pressure to the required value so the gas can be delivered at the contract delivery pressure at the end of the pipeline. For example, consider a 200 mi long, NPS 16 pipeline that transports 150 MMSCFD of gas from Compton to a delivery point at Beaumont, as shown in Figure 3.11. Suppose calculations show that 1600 psig pressure is required at Compton in order to deliver gas to Beaumont at 800 psig. If the maximum operating pressure (MOP) of this pipeline is limited to 1350 psig, obviously we will need more than one compressor station. The first compressor station will be located at Compton and will provide a pressure of 1350 psig. As gas flows from Compton toward some intermediate location, such as Sheridan, the gas pressure will drop to some value such as 900 psig. At Sheridan, a second compressor station will boost the gas pressure to 1350 psig on its way to the terminus at Beaumont. By installing the second compressor station at Sheridan, pipeline pressures are maintained within MOP limits. The actual location of the intermediate compressor station at Sheridan will depend upon many factors, including pipeline elevation profile, the gas pressure at Sheridan, and the delivery pressure required at Beaumont. The hydraulic pressure gradient in this case is as shown in Figure 3.11. In the preceding discussion, we picked an arbitrary pressure of 900 psig at Sheridan. This gives us an approximate compression ratio of

$$
\frac{1350+14.7}{914.7}=1.492
$$

which is a reasonable number for centrifugal compressors used in gas pipelines. In reference to Figure 3.11, we will now outline the method of locating the intermediate


NPS 16 pipeline 200 mi long

Figure 3.11 Compton to Beaumont pipeline.
compressor station at Sheridan. In Chapter 4, we will further analyze gas pipelines with multiple compressor stations.

Starting at Compton, with an inlet pressure $P_{1}=1350 \mathrm{psig}$, we calculate the pipe length $L$ that will cause the pressure to drop to 900 psig, using the General Flow equation. Assuming a flow rate of 150 MMSCFD and a friction factor $f=0.01$, we get

$$
150 \times 10^{6}=77.54\left(\frac{1}{\sqrt{0.01}}\right)\left(\frac{520}{14.7}\right)\left[\frac{1364.7^{2}-914.7^{2}}{0.6 \times 520 \times L \times 0.9}\right]^{0.5} \times(15.5)^{2.5}
$$

Solving for $L$, we get

$$
L=109.28 \mathrm{mi}
$$

Thus, the approximate location of the second compressor station at Sheridan is 109.28 mi from Compton. If we allow the compressor at Sheridan to boost the gas pressure to 1350 psig , the compression ratio is

$$
r=\frac{1350+14.7}{914.7}=1.492
$$

which is a reasonable ratio for a centrifugal compressor. Therefore, starting at 1350 psig at Sheridan, we calculate the delivery pressure at Beaumont using the General Flow equation for $(200-109.28)=90.7 \mathrm{mi}$ of pipe as follows:

$$
150 \times 10^{6}=77.54\left(\frac{1}{\sqrt{0.01}}\right)\left(\frac{520}{14.7}\right)\left[\frac{1364.7^{2}-P_{2}^{2}}{0.6 \times 520 \times 90.7 \times 0.9}\right]^{0.5} \times(15.5)^{2.5}
$$

Solving for $P_{2}$, we get

$$
P_{2}=1005.5 \mathrm{psia}=990.82 \mathrm{psig}
$$

This is more than the required delivery pressure at Beaumont of 800 psig. We could go back and repeat the above calculations, considering slightly lower pressure at Compton, say 1300 psig , in order to get the correct delivery pressure of 800 psig at Beaumont. This is left as an exercise for the reader.

Alternatively, we could start with the required delivery pressure of 800 psig at Beaumont and work backward to determine the distance at which the upstream pressure reaches 1350 psig. That would be the location for the Sheridan compressor station. Next, we would determine the pressure at Sheridan, beginning with the 1350 psig upstream pressure at Compton. This would establish the suction pressure of the Sheridan compressor station. Knowing the suction pressure and the discharge pressure at Sheridan, we could calculate the compression ratio required. In Chapter 4 we will discuss multiple compressor stations in more detail.

### 3.10 PRESSURE REGULATORS AND RELIEF VALVES

In a long-distance gas pipeline with intermediate delivery points, there may be a need to regulate the gas pressure at certain delivery points in order to satisfy the customer requirements. Suppose the pressure at the delivery point B in Figure 3.11 is 800 psig , whereas the customer requirement is only 500 psig . Obviously, some means of reducing the gas pressure must be provided so that the customer can utilize the gas for his or her requirements at the correct pressure. This is achieved by means of a pressure regulator that will ensure a constant pressure downstream of the delivery point, regardless of the pressure on the upstream side of the pressure regulator. This concept is further illustrated using the example pipeline shown in Figure 3.12.

The main pipeline from $A$ to $C$ is shown along with a branch pipe $B E$. The flow rate from $A$ to $B$ is 100 MMSCFD, with an inlet pressure of 1200 psig at A. At B, gas is delivered into a branch line BE at the rate of 30 MMSCFD . The remaining volume of 70 MMSCFD is delivered to the pipeline terminus C at a delivery pressure of 600 psig. Based on the delivery pressure requirement of 600 psig at C and a takeoff of 30 MMSCFD at point B , the calculated pressure at B is 900 psig . Starting with 900 psig on the branch line at B , at 30 MMSCFD , gas is delivered to point E at 600 psig . If the actual requirement at E is only 400 psig , a pressure regulator will be installed at E to reduce the delivery pressure by 200 psig .

It can be seen from Figure 3.12 that at point D immediately upstream of the pressure regulator, the gas pressure is approximately 600 psig and is regulated to 400 psig downstream at E . If the mainline flow rate changes from 100 MMSCFD to 90 MMSCFD and the delivery at B is maintained at 30 MMSCFD, the gas pressure at B will reduce to a value below 900 psig. Accordingly, the pressure at point D in the branch pipe BE will also reduce to some value below 600 psig . Regardless, due to the pressure regulator, the pressure at E will be maintained at the required 400 psig . However, if for some reason the pressure upstream of the regulator at D falls below 400 psig, the downstream pressure at E cannot be maintained at the original value of 400 psig . The pressure regulator can only reduce the pressure downstream to the required value. It cannot increase the pressure beyond the pressure on the upstream


Figure 3.12 Pressure regulation.
side. If the pressure at D drops to 300 psig , the pressure regulator is ineffective and will remain fully open, and the delivery pressure at E will be 300 psig as well.

## Example 9

A natural gas pipeline, NPS 16, 0.250 in . wall thickness, 50 mi long, with a branch pipe (NPS 8, 0.250 in . wall thickness, 15 mi long), as shown in Figure 3.13, is used to transport 100 MMSCFD gas (gravity $=0.6$ and viscosity $=0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{s}$ ) from A to B. At B (milepost 20), a delivery of 30 MMSCFD occurs into the branch pipe BE . The delivery pressure at E must be maintained at 300 psig . The remaining volume of 70 MMSCFD is shipped to the terminus $C$ at a delivery pressure of 600 psig. Assume a constant gas temperature of $60^{\circ} \mathrm{F}$ and a pipeline efficiency of 0.95 . The base temperature and base pressure are $60^{\circ} \mathrm{F}$ and 14.7 psia, respectively. The compressibility factor $Z=0.88$.
a) Using the Panhandle A equation, calculate the inlet pressure required at A.
b) Is a pressure regulator required at E ?
c) If the inlet flow at A drops to 60 MMSCFD, what is the impact in the branch pipeline BE if the flow rate of 30 MMSCFD is maintained?

Solution

Pipe inside diameter for pipe segment AB and $\mathrm{BC}=16-2 \times 0.25=15.5 \mathrm{in}$.
First, we will consider the pipe segment BC and calculate the pressure $P_{1}$ at B for a 70 MMSCFD flow rate to deliver gas at 600 psig at C. Using Panhandle A Equation 2.55, neglecting elevation effects,

$$
70 \times 10^{6}=435.87 \times 0.95\left(\frac{60+460}{14.7}\right)^{1.0788}\left(\frac{P_{1}^{2}-614.7^{2}}{0.6^{0.8539} \times 520 \times 30 \times 0.88}\right)^{0.5394}(15.5)^{2.6182}
$$



Figure 3.13 Example problem—pressure regulation.

Solving for pressure at B , we get

$$
P_{1}=660.39 \mathrm{psia}=645.69 \mathrm{psig}
$$

Next, considering pipe segment AB , flowing 100 MMSCFD , we calculate the inlet pressure $P_{1}$ at A using the outlet pressure 660.39 psia we calculated at B .

From Panhandle A Equation 2.55,

$$
100 \times 10^{6}=435.87 \times 0.95\left(\frac{520}{14.7}\right)^{1.0788}\left(\frac{P_{1}^{2}-660.39^{2}}{0.6^{0.8539} \times 520 \times 20 \times 0.88}\right)^{0.5394}(15.5)^{2.6182}
$$

Solving for the pressure at A , we get

$$
P_{1}=715.08 \mathrm{psia}=700.38 \mathrm{psig}
$$

Next, using the pressure 660.39 psia at B , we calculate the outlet pressure of branch BE that flows 30 MMSCFD through the 15 mi NPS 8 pipe.

Using Panhandle A Equation 2.55,

$$
30 \times 10^{6}=435.87 \times 0.95\left(\frac{520}{14.7}\right)^{1.0788}\left(\frac{660.39^{2}-P_{2}^{2}}{0.6^{0.8539} \times 520 \times 15 \times 0.88}\right)^{0.5394}(8.125)^{2.6182}
$$

Solving for the pressure at E ,

$$
P_{2}=544.90 \mathrm{psia}=530.2 \mathrm{psig}
$$

Since the required delivery pressure at E is 300 psig , a pressure regulator will be required at E .

If the flow rate at A drops to 60 MMSCFD and the branch BE flow rate is maintained at 30 MMSCFD , we will calculate the junction pressure at B by using Panhandle A Equation 2.55 for the pipe segment BC, considering a flow rate of 30 MMSCFD and a delivery pressure of 600 psig at C .

$$
30 \times 10^{6}=435.87 \times 0.95\left(\frac{520}{14.7}\right)^{1.0788}\left(\frac{P_{1}^{2}-614.7^{2}}{0.6^{0.8539} \times 520 \times 30 \times 0.88}\right)^{0.5394}(15.5)^{2.6182}
$$

Solving for the pressure at $B$, we get

$$
P_{1}=624.47 \mathrm{psia}=609.77 \mathrm{psig}
$$

Using the pressure at B , we calculate the outlet pressure at E on branch BE for a 30 MMSCFD flow rate using Panhandle A Equation 2.55:

$$
30 \times 10^{6}=435.87 \times 0.95\left(\frac{520}{14.7}\right)^{1.0788}\left(\frac{624.47^{2}-P_{2}^{2}}{0.6^{0.8539} \times 520 \times 15 \times 0.88}\right)^{0.5394}(8.125)^{2.6182}
$$

Solving for the pressure $P_{2}$, we get

$$
P_{2}=500.76 \mathrm{psia}=486.06 \mathrm{psig}
$$

This is the new pressure at E . Comparing this pressure with the previously calculated pressure of 530.2 psig , we see that the delivery pressure at E has dropped by 44 psig , approximately. To maintain the delivery pressure of 300 psig at E , a pressure regulator is still required.

Therefore, the answers are
a) Inlet pressure at $\mathrm{A}=700.38$ psig.
b) A pressure regulator is required at E to reduce the pressure from 530.2 psig to 300 psig.
c) Finally, a pressure regulator is required at E to reduce the pressure from 486.1 psig to 300 psig.

### 3.11 TEMPERATURE VARIATION AND GAS PIPELINE MODELING

In the preceding sections we assumed the gas temperature to be constant (isothermal) along the length of the pipeline. By assuming isothermal flow, we were able to calculate the pressure drop using constant gas properties such as the compressibility factor. In reality, the temperature of gas in a buried pipeline varies along the length of the pipeline due to heat transfer between the gas and the surrounding soil. If the inlet temperature of the gas is $80^{\circ} \mathrm{F}$ and the surrounding soil is $60^{\circ} \mathrm{F}$, the temperature difference will cause heat loss from the gas to the soil. Additionally, in a longdistance pipeline, the soil temperature can vary along the pipeline. This will cause the gas temperature to vary, as shown in Figure 3.14.


Figure 3.14 Temperature variation in a gas pipeline.

Generally, if the pipeline is fairly long, the gas temperature will ultimately equal the soil temperature as gas approaches the delivery point. Due to such variation in gas temperature, calculation of pressure drop must be made by considering short lengths of pipe that make up the total pipeline. For example, if the pipeline is 50 mi long, we will subdivide the pipeline into short segments of 1- or 2-mile lengths and apply the General Flow equation for each pipe segment. Starting with the upstream pressure of segment 1 , the downstream pressure will be calculated, assuming an average temperature for segment 1 . Next, using the calculated downstream pressure as the upstream pressure for segment 2 , we calculate the downstream pressure for segment 2. The process is continued until all segments of the pipeline are covered. It must be noted that the variation of temperature from segment to segment must be taken into account to calculate the compressibility factor to be used in the General Flow equation. The calculation of gas temperature at any point along the pipeline by taking into account the heat transfer between the gas and surrounding soil is quite complicated. It does not lend itself easily to manual calculations. The method of calculation will be discussed briefly in this section for information only. To accurately take into account the temperature variations, a suitable gas pipeline hydraulics simulation program must be used, since, as indicated earlier, manual calculation is quite laborious and time consuming. Several commercial simulation programs are available to model steady-state gas pipeline hydraulics. These programs calculate the gas temperature and pressures by taking into consideration variations of soil temperature, pipe burial depth, and thermal conductivities of pipe, insulation, and soil. One such software program is GASMOD, marketed by SYSTEK Technologies, Inc. (www.systek.us). Appendix A includes a sample simulation of a gas pipeline using the GASMOD software.

Even though manual calculation of the temperature variation and corresponding pressure drop in a gas pipeline is quite tedious, we will present here the basic equations for reference.

Consider a buried pipeline transporting gas from point A to point B. We will analyze a short segment of length $\Delta L$ of this pipe as shown in Figure 3.15 and apply the principles of heat transfer to determine how the gas temperature varies along the pipeline.

The upstream end of the pipe segment of length $\Delta L$ is at a temperature $T_{1}$ and the downstream end at temperature $T_{2}$. The average gas temperature in this segment is


Figure 3.15 Analysis of temperature variation.
represented by $T$. The outside soil temperature at this location is $T_{s}$. Assume steadystate conditions and the mass flow rate of gas to be $m$. The gas flow from the upstream end to the downstream end of the segment causes a temperature drop of $\Delta T$. The heat loss from the gas can be represented by

$$
\begin{equation*}
\Delta H=-m C p \Delta T \tag{3.23}
\end{equation*}
$$

where
$\Delta H=$ heat transfer rate, Btu/h
$m=$ mass flow rate of gas, $\mathrm{lb} / \mathrm{h}$
$C p=$ average specific heat of gas, $\mathrm{Btu} / \mathrm{lb} /{ }^{\circ} \mathrm{F}$
$\Delta T=$ temperature difference $=T_{1}-T_{2}{ }^{\circ} \mathrm{F}$
The negative sign in Equation 3.23 indicates loss of heat from upstream temperature $T_{1}$ to downstream temperature $T_{2}$.

Next, we consider the heat transfer from the gas to the surrounding soil in terms of the overall heat transfer coefficient $U$ and the difference in temperature between the gas and surrounding soil, represented by $\left(T-T_{s}\right)$. Therefore, we can write the following equation for heat transfer:

$$
\begin{equation*}
\Delta H=U \Delta A\left(T-T_{s}\right) \tag{3.24}
\end{equation*}
$$

where
$U=$ overall heat transfer coefficient, Btu $/ \mathrm{h} / \mathrm{ft}^{2} /{ }^{\circ} \mathrm{F}$
$\Delta A=$ surface area of pipe for heat transfer $=\pi D \Delta L$
$T=$ average gas temperature in pipe segment, ${ }^{\circ} \mathrm{F}$
$T_{s}=$ average soil temperature surrounding pipe segment, ${ }^{\circ} \mathrm{F}$
$D=$ pipe inside diameter, ft
Equating the two values of heat transfer rate $\Delta H$ from Equation 3.23 and Equation 3.24, we get

$$
-m C p \Delta T=U \Delta A\left(T-T_{s}\right)
$$

Simplifying, we get

$$
\begin{equation*}
\frac{\Delta T}{T-T_{s}}=-\left(\frac{\pi U D}{m C p}\right) \Delta L \tag{3.25}
\end{equation*}
$$

Rewriting Equation 3.25 in differential form and integrating, we get

$$
\begin{equation*}
\int_{1}^{2} \frac{d T}{T-T_{s}}=\int_{2}^{1}-\left(\frac{\pi U D}{m C p}\right) d L \tag{3.26}
\end{equation*}
$$

Integrating and simplifying, we get

$$
\begin{equation*}
\frac{T_{2}-T_{s}}{T_{1}-T_{s}}=e^{-\theta} \tag{3.27}
\end{equation*}
$$



Figure 3.16 Joule-Thompson effect in gas pipeline.
where

$$
\begin{equation*}
\theta=\frac{\pi U D \Delta L}{m C p} \tag{3.28}
\end{equation*}
$$

Simplifying Equation 3.27 further, we get the downstream temperature of the pipe segment of length $\Delta \mathrm{L}$ as

$$
\begin{equation*}
T_{2}=T_{s}+\left(T_{1}-T_{s}\right) e^{-\theta} \tag{3.29}
\end{equation*}
$$

It can be seen from Equation 3.29 that as the pipe length increases, the term $e^{-\theta}$ approaches zero and the temperature, $T_{2}$, becomes equal to soil temperature, $T_{s}$. Therefore, in a long gas pipeline, the gas temperature ultimately equals the surrounding soil temperature. This is illustrated in Figure 3.14.

In the preceding analysis, we made several simplifying assumptions. We assumed that the soil temperature and the overall heat transfer coefficient remained constant and ignored the Joule-Thompson effect as gas expands through a pipeline. In a long pipeline, the soil temperature can actually vary along the pipeline and, therefore, must be taken into account in these calculations. One approach would be to subdivide the pipeline into segments that have constant soil temperatures and perform calculations for each segment separately. The Joule-Thompson effect causes the gas to cool slightly due to expansion. Therefore, in a long pipeline, the gas temperature at the delivery point may fall below that of the ground or soil temperature, as indicated in Figure 3.16.

### 3.12 LINE PACK

As gas flows through a pipeline from point A to point B , the pressures and temperatures vary along the pipeline length. The volume of gas contained in a given length of pipeline is simply the physical volume of the pipe segment. For example, a 1-mile
section of NPS 16 pipe can have a physical volume of $7000 \mathrm{ft}^{3}$. Therefore, this volume represents the volume of the gas in this 1-mile section at the actual gas temperature and pressure. The quantity of gas contained within the pipeline under pressure, measured at standard conditions (generally 14.7 psia and $60^{\circ} \mathrm{F}$ ), is termed the line pack volume. Consider a segment of pipe, of length $L$, with upstream pressure and temperature of $P_{1}$ and $T_{1}$ and downstream values of $P_{2}$ and $T_{2}$, respectively. We can calculate the line pack using the gas laws discussed in Chapter 1. Suppose the inside diameter of the pipe is $D$; then the physical volume $V_{p}$ of the pipe section is

$$
\begin{equation*}
V_{p}=\frac{\pi}{4} D^{2} L \tag{3.30}
\end{equation*}
$$

This volume is the gas volume at pressures and temperatures ranging from $P_{1}$, $T_{1}$ at the upstream end to $P_{2}, T_{2}$ at the downstream end of the pipe length $L$. In order to convert this volume to standard conditions of pressure, $P_{b}$, and temperature, $T_{b}$, we apply the gas law Equation 1.16 as follows:

$$
\begin{equation*}
\frac{P_{b} V_{b}}{Z_{b} T_{b}}=\frac{P_{\text {avg }} V_{p}}{Z_{\text {avg }} T_{\text {avg }}} \tag{3.31}
\end{equation*}
$$

where
$P_{\text {avg }}=$ average gas pressure in pipe segment
$T_{\text {avg }}=$ average gas temperature in pipe segment
$Z_{\text {avg }}=$ average gas compressibility factor at $T_{\text {avg }}$ and $P_{\text {avg }}$
$Z_{b}=$ compressibility factor at base conditions $=1.00$, approximately
The average pressure, $P_{\text {avg }}$, is calculated from the upstream and downstream pressures $P_{1}$ and $P_{2}$ using Equation 2.14. The average temperature can be taken as the arithmetic mean of the upstream and downstream temperatures $T_{1}$ and $T_{2}$. This approach for average temperature will be accurate only if we consider short segments of pipe.

From Equation 3.31, solving for line pack $V_{b}$ at standard conditions, we get

$$
\begin{equation*}
V_{b}=\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{\text {avg }} V_{p}}{Z_{\text {avg }} T_{\text {avg }}}\right) \tag{3.32}
\end{equation*}
$$

Substituting the value of $V_{p}$ from Equation 3.30 and simplifying, we get

$$
\begin{equation*}
V_{b}=0.7854\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{\text {avg }}}{Z_{\text {avg }} T_{\text {avg }}}\right)\left(D^{2} L\right) \tag{3.33}
\end{equation*}
$$

where
$V_{b}=$ line pack in pipe segment, standard $\mathrm{ft}^{3}$
$D=$ pipe inside diameter, ft
$L=$ pipe segment length, ft

Other symbols are as defined before.

Equation 3.33 is modified in terms of commonly used pipeline units as follows:

$$
\begin{equation*}
V_{b}=28.798\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{\text {avg }}}{Z_{\text {avg avg }} T_{\text {av }}}\right)\left(D^{2} L\right) \quad \text { (USCS units) } \tag{3.34}
\end{equation*}
$$

where
$V_{b}=$ line pack in pipe segment, standard $\mathrm{ft}^{3}$
$D=$ pipe inside diameter, in.
$L=$ pipe segment length, mi
Other symbols are as defined before.
The corresponding equation in SI units is

$$
\begin{equation*}
V_{b}=7.855 \times 10^{-4}\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{\text {avg }}}{Z_{\text {avg avg }}}\right)\left(D^{2} L\right) \quad \text { (SI units) } \tag{3.35}
\end{equation*}
$$

where
$V_{b}=$ line pack in pipe segment, standard $\mathrm{m}^{3}$
$D=$ pipe inside diameter, mm
$L=$ pipe segment length, km

Other symbols are as defined before.
Since the pressure and temperature in a gas pipeline vary along the length, to improve the accuracy of calculations, the line pack volume $V_{b}$ is calculated for short segments of pipe and summed to obtain the line pack of the entire pipeline.

## Example 10

A natural gas pipeline is 10 mi long and has an inlet pressure of 1000 psig and outlet pressure of 900 psig when transporting 100 MMSCFD. The base pressure and base temperature are 14.7 psia and $60^{\circ} \mathrm{F}$, respectively. If the pipe is NPS $16,0.250 \mathrm{in}$. wall thickness, calculate the line pack assuming an average gas temperature of $78^{\circ} \mathrm{F}$. Use an average compressibility factor of 0.90 .

## Solution

Pipe inside diameter $=16-2 \times 0.250=15.5 \mathrm{in}$.

The average pressure is calculated from Equation 2.14 as follows:

$$
P_{\text {avg }}=\frac{2}{3}\left(1014.7+914.7-\frac{1014.7 \times 914.7}{1014.7+914.7}\right)=965.56 \mathrm{psia}
$$

Using Equation 3.34, we calculate the line pack as follows:

$$
V_{b}=28.798\left(\frac{60+460}{14.7}\right)\left(\frac{965.56}{78+460}\right)\left(\frac{15.5^{2} \times 10}{0.90}\right)=4,880,521 \text { standard } \mathrm{ft}^{3}
$$

Therefore, the line pack is $4,880,521$ standard $\mathrm{ft}^{3}$.

## Example 11

A natural gas pipeline is 20 km long and has an inlet pressure of 8000 kPa (gauge) and outlet pressure of 5000 kPa (gauge) when transporting $5 \mathrm{Mm}^{3} / \mathrm{day}$. The base pressure and base temperature are 101 kPa and $15^{\circ} \mathrm{C}$, respectively. If the pipe is DN 500, 12 mm wall thickness, calculate the line pack assuming an average gas temperature of $20^{\circ} \mathrm{C}$. Use an average compressibility factor of 0.90 .

## Solution

Pipe inside diameter $=500-2 \times 12=476 \mathrm{~mm}$

The average pressure is calculated from Equation 2.14 as follows:

$$
P_{\text {avg }}=\frac{2}{3}\left(8101+5101-\frac{8101 \times 5101}{8101+5101}\right)=6714.62 \mathrm{kPa} \text { (absolute) }
$$

Using Equation 3.35, we calculate the line pack as follows:

$$
V_{b}=7.855 \times 10^{-4}\left(\frac{15+273}{101}\right)\left(\frac{6714.62}{20+273}\right)\left(\frac{476^{2} \times 20}{0.9}\right)=258,448 \text { standard } \mathrm{m}^{3}
$$

Therefore, the line pack is 258,448 standard $\mathrm{m}^{3}$.

### 3.13 SUMMARY

In this chapter we continued to look at the application of the pressure drop equations introduced in Chapter 2. Several piping configurations, such as pipes in series, pipes in parallel, and gas pipelines with injections and deliveries, were analyzed to determine pressures required and pipe size needed to satisfy certain requirements. The concepts of equivalent length in series piping and equivalent diameter in pipe loops were explained and illustrated using example problems. The hydraulic pressure gradient and the need for intermediate compressor stations to transport given volumes of gas without exceeding allowable pipeline pressures were also covered. The importance of temperature variation in gas pipelines and how it is taken into account in calculating pipeline pressures were introduced with reference to commercial hydraulic simulation models. The method of calculating the line pack volume in a gas pipeline was also explained. In the next chapter, we will discuss compressor stations, compressor performance, and horsepower requirements.

## PROBLEMS

1. A pipeline, NPS 14 with 0.250 in . wall thickness, 40 mi long, transports natural gas (specific gravity $=0.6$ and viscosity $=0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{s}$ ) at a flow rate of 80 MMSCFD at an inlet temperature of $60^{\circ} \mathrm{F}$. Assuming isothermal flow and neglecting elevation changes, calculate the inlet pressure required for a delivery pressure of 800 psig. The base pressure and base temperature are 14.7 psia and $60^{\circ} \mathrm{F}$, respectively. Use the Colebrook equation with pipe roughness of 0.0007 in .
2. A 100 mi long natural gas pipeline consists of several injections and deliveries. The pipeline is NPS 18, 0.375 in . wall thickness and has an inlet volume of 150 MMSCFD. At points B (milepost 25) and C (milepost 70), 60 MMSCFD and 50 MMSCFD, respectively, are delivered. At D (milepost 90), gas enters the pipeline at 40 MMSCFD . All streams of gas can be assumed to have a specific gravity of 0.60 and a viscosity of $7.5 \times 10^{-6} \mathrm{lb} / \mathrm{ft}-\mathrm{s}$. The pipe is inter-nally coated such that the absolute roughness is $200 \mu \mathrm{in}$. Assume a constant gas flow temperature of $80^{\circ} \mathrm{F}$ and base pressure and base temperature of 14.7 psia and $60^{\circ} \mathrm{F}$, respectively. Use a constant compressibility factor of 0.88 throughout. Neglect elevation differences along the pipeline.
a) Using the modified Colebrook equation, calculate the pressures along the pipeline at points A, B, C, and D for a minimum delivery pressure of 400 psig at the terminus E .
b) What diameter pipe will be required for section DE if the required delivery pressure at E is increased to 600 psig ?
3. A natural gas pipeline, 210 km long, consists of an inlet stream at A and deliveries at B and C. The pipeline is DN 400, 10 mm wall thickness. At A, the gas enters at a flow rate of $3.5 \mathrm{Mm}^{3} /$ day. At points $B(\mathrm{~km} \mathrm{20})$ and $C(\mathrm{~km} \mathrm{100})$, gas is delivered at $0.5 \mathrm{Mm}^{3} /$ day and $1.0 \mathrm{Mm}^{3} /$ day, respectively. At D (km 150), gas enters a branch pipe DF (DN 200, 6 mm wall thickness, 10 km long) at a flow rate of $1.0 \mathrm{Mm}^{3} /$ day. The remaining gas volume of $1.0 \mathrm{Mm}^{3} /$ day is delivered to the pipe terminus E. All streams of gas can be assumed to have a specific gravity of 0.58 and a viscosity of 0.00012 Poise. The pipe's absolute roughness is 0.015 mm throughout. Assume a constant gas flow temperature of $15^{\circ} \mathrm{C}$ and base pressure and base temperature of 101 kPa and $15^{\circ} \mathrm{C}$, respectively. Use a pipeline efficiency of 0.95 and constant compressibility factor of 0.88 throughout. Neglect elevation differences along the pipeline.
a) Using the Panhandle B equation, calculate the pressures along the pipeline at A, B, C, and D for a minimum delivery pressure of 30 Bar at terminus E .
b) What is the delivery pressure of gas at the end of the branch DF?
c) What pipe diameter is needed for the branch DF if the delivery pressure required at F is 40 Bar ?
4. A series piping system consists of 10 mi of NPS $16,0.250 \mathrm{in}$. wall thickness, connected to 20 mi of NPS $14,0.250 \mathrm{in}$. wall thickness and 10 miles of NPS 12, 0.250 in. wall thickness pipes. Calculate the inlet pressure required at the beginning A for a gas flow rate of 85 MMSCFD. Gas is delivered to the terminus B at
a delivery pressure of 600 psig. The gas gravity and viscosity are 0.6 and $0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{s}$, respectively. The gas temperature is assumed constant at $60^{\circ} \mathrm{F}$. Use a compressibility factor of 0.85 and the General Flow equation with Colebrook friction factor of 0.015 . The base temperature and base pressure are $60^{\circ} \mathrm{F}$ and 14.7 psia, respectively.
5. A gas pipeline consists of two single pipes with a couple of parallel pipes in the middle. The inlet flow rate is 120 MMSCFD. The first pipe segment AB is 10 miles long and consists of NPS 16, 0.250 in . wall thickness pipe. The loop BCE is 20 mi long and consists of NPS 14, 0.250 in . wall thickness pipe. The loop BDE is 15 miles long and consists of NPS 12, 0.250 in . wall thickness pipe. The last segment EF is 18 miles long, NPS 16, 0.250 in. wall thickness pipe. Assuming a gas gravity of 0.6 , calculate the outlet pressure at F and the pressures at the beginning and the end of the pipe loops and the flow rates through them. The inlet pressure at $\mathrm{A}=1000 \mathrm{psig}$. The gas flowing temperature $=60^{\circ} \mathrm{F}$, base temperature $=60^{\circ} \mathrm{F}$, and base pressure $=14.73 \mathrm{psia}$. The compressibility factor $Z=0.90$. Use the AGA fully turbulent equation throughout.
6. A natural gas pipeline is 60 km long. The gas flow rate is $5.0 \mathrm{Mm}^{3} /$ day at $20^{\circ} \mathrm{C}$. Calculate the minimum diameter required for an inlet and delivery pressure of 8.5 MPa (absolute) and 5 MPa (absolute), respectively. Use the General Flow equation with the modified Colebrook-White friction factor. The pipe roughness = 0.020 mm . In order to increase the flow rate through the pipeline, the entire line is looped with an identical-diameter pipeline. Assuming the same delivery pressure, calculate the inlet pressure at the new flow rate of $8 \mathrm{Mm}^{3} /$ day. The gas gravity $=$ 0.60 and viscosity $=0.000119$ Poise. The compressibility factor $Z=0.90$, base temperature $=15^{\circ} \mathrm{C}$, and base pressure $=101 \mathrm{kPa}$.
7. A natural gas pipeline is 50 mi long and has an inlet pressure of 1200 psig and outlet pressure of 890 psig when transporting 120 MMSCFD. The base pressure and base temperature are 14.7 psia and $60^{\circ} \mathrm{F}$, respectively. If the pipe is NPS 16 , 0.375 in. wall thickness, calculate the line pack assuming an average gas temperature of $75^{\circ} \mathrm{F}$. Use an average compressibility factor of 0.85 .

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## CHAPTER 4

## Compressor Stations

In this chapter we will discuss compressor stations that are needed to transport gas in a pipeline. The optimum locations and pressures at which compressor stations operate will be analyzed in relation to pipeline throughputs, allowable pipe pressures, and pipeline topography. Centrifugal and positive displacement compressors used in natural gas transportation will be compared with reference to their performance characteristics and cost. Typical compressor station design and equipment used will be covered. Isothermal, adiabatic, and polytropic compression processes and horsepower required will be discussed with illustrative examples. The discharge temperature of compressed gas, its impact on pipeline throughput and gas cooling will be explained.

### 4.1 COMPRESSOR STATION LOCATIONS

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.

Consider a pipeline that is designed to transport 100 MMSCFD of natural gas from Dover to a power plant at Leeds, 50 miles away. According to methods outlined in Chapter 3, we would calculate the pressure required at Dover to ensure delivery of the gas at a pressure of 500 psig at Leeds. This calculated pressure at Dover may be more or less than the maximum allowable pipe pressures. Suppose the maximum allowable operating pressure (MAOP) of the pipeline is 1200 psig , whereas the pressure at Dover is calculated to be 1050 psig . It is clear that there is no violation of pressures and, hence, a single compressor station at Dover would suffice to deliver gas to Leeds at the required delivery pressure. If the pipeline length were 100 miles
instead, calculations would show that in order to deliver the same quantity of gas to Leeds at the same terminus pressure, the pressure required at Dover would have to be 1580 psig. Obviously, since this is greater than the MAOP, we would need more than one compressor station.

As a first step, we will assume that an intermediate compressor station will be needed in addition to the one at Dover. The next question is where would this compressor station be located? A logical location would be the midpoint between Dover and Leeds.

For simplicity, assume the pipeline elevation profile is fairly flat and, therefore, elevation differences can be ignored. Having selected the location of the intermediate pump station at the midpoint, Kent, as shown in Figure 4.1, we will proceed to determine the pressures at the compressor stations.

Since the MAOP is limited to 1200 psig, assume that the compressor at Dover discharges at this pressure. Due to friction, the gas pressure drops as it travels through the pipeline from Dover to Kent, as indicated in Figure 4.1. Suppose the gas pressure reaches 900 psig at Kent and is boosted to 1200 psig by the compressor at Kent. Therefore, the compressor station at Kent is said to have a suction pressure of 900 psig and a discharge pressure of 1200 psig . The gas continues to move from Kent to Leeds, starting at 1200 psig at Kent. As the gas reaches Leeds, the pressure may or may not be equal to the desired pressure of 500 psig . Therefore, if the desired terminus pressure at Leeds is maintained, the pressure at the discharge of the Kent compressor stations may have to be adjusted. Alternatively, Kent could discharge at the same 1200 psig , but its location along the pipeline may have to be adjusted. We selected the 900 psig suction pressure at the Kent compressor station quite arbitrarily. It could have been 700 psig or 1000 psig. The actual number depends upon the "compression ratio" desired. The compression ratio is simply the


Figure 4.1 Gas pipeline with two compressor stations.
ratio of the compressor discharge pressure to its suction pressure, both pressures being expressed in absolute units.

$$
\begin{equation*}
\text { Compression ratio } r=\frac{P_{d}}{P_{s}} \tag{4.1}
\end{equation*}
$$

where the suction and discharge pressures $P_{s}$ and $P_{d}$ are in absolute units.
In the present case, the compression ratio for Kent is

$$
r=\frac{1200+14.7}{900+14.7}=1.33
$$

In the above calculation, we assumed the base pressure to be 14.7 psia. If we had chosen a suction pressure of 700 psig , the compression ratio would be

$$
r=\frac{1200+14.7}{700+14.7}=1.7
$$

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. Therefore, if the suction pressure at Kent is allowed to fall to 700 psig or lower, the average pressure in the pipeline would be lower than if we chose 900 psig for the suction pressure at Kent. Obviously, there is a tradeoff between the number of compressor stations, the suction pressure, and the compression horsepower required. We will discuss this in more detail later in this chapter.

Going back to the example problem above, we concluded that we may have to adjust the location of the Kent compressor station or adjust its discharge pressure to ensure the 500 psig delivery pressure at Leeds. Alternatively, we could leave the intermediate compressor station at the halfway point and discharge at 1200 psig, eventually delivering gas to Leeds at 600 psig. If calculations show that by discharging out of Kent results in 600 psig at Leeds, we have satisfied the minimum pressure requirement at Leeds. However, there is extra energy associated with the extra 100 psig delivery pressure. If the power plant can use this extra energy, then there is no waste. On the other hand, if the power plant requirement is 500 psig maximum, then some means of regulating the pressure must be present at the delivery point. This would mean that 100 psig would be reduced through a pressure regulator or control valve at Leeds and energy would be wasted. Another option would be to keep the Kent compressor at the midpoint but reduce the discharge pressure to a number that would result in the requisite 500 psig at Leeds. Since pressure drop in gas pipelines is nonlinear, remembering our discussion in Chapter 3, Kent discharge pressure may have to be reduced by less than 100 psig to provide the fixed 500 psig delivery pressure at Leeds. This would mean that Dover would operate at 1200 psig discharge, whereas Kent would discharge at 1150 psig. This violates our premise of keeping the average pressure in a gas pipeline as high as possible. However, this is
still a solution, and in order to pick the best option, we must compare two or more alternative approaches, factoring in the total horsepower required as well as the cost involved. Moving the Kent compressor station slightly upstream or downstream would change the suction and discharge pressures and, hence, the horsepower required. From a cost standpoint, the change would not be significant. However, the horsepower variation would result in change in energy cost and, therefore, in annual operating cost. We must therefore take into account the capital cost and annual operating cost in order to come up with the optimum solution. An example will illustrate this method. In Chapter 10, we will cover several different cost scenarios when dealing with pipeline economics.

## Example 1

A natural gas pipeline, 140 miles long from Dover to Leeds, is constructed of NPS $16,0.250 \mathrm{in}$. wall thickness pipe, with an MOP of 1200 psig . The gas specific gravity and viscosity are 0.6 and $8 \times 10^{-6} \mathrm{lb} / \mathrm{ft}-\mathrm{s}$, respectively. The pipe roughness can be assumed to be $700 \mu \mathrm{in}$., and the base pressure and base temperature are 14.7 psia and $60^{\circ} \mathrm{F}$, respectively. The gas flow rate is 175 MMSCFD at $80^{\circ} \mathrm{F}$, 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 will use the Colebrook-White equation to calculate the pressure drop.
The Reynolds number is calculated from Equation 2.34 as follows:

$$
\begin{aligned}
& R=\frac{0.0004778 \times 175 \times 10^{6} \times 0.6 \times 14.7}{15.5 \times 8 \times 10^{-6} \times 520}=11,437,412 \\
& \text { Relative roughness }=\frac{700 \times 10^{-6}}{15.5}=4.5161 \times 10^{-5}
\end{aligned}
$$

Using Colebrook-White Equation 2.39, we get the friction factor

$$
\frac{1}{\sqrt{f}}=-2 \log _{10}\left[\frac{4.516 \times 10^{-5}}{3.7}+\frac{2.51}{11,437,412 \sqrt{f}}\right]
$$

Solving for $f$ by successive iteration, we get

$$
f=0.0107
$$

Using General Flow Equation 2.2, we calculate the pressure required at Dover as, neglecting elevation effects,

$$
175 \times 10^{6}=77.54\left(\frac{1}{\sqrt{0.0107}}\right)\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-814.7^{2}}{0.6 \times 540 \times 140 \times 0.85}\right)^{0.5} \times(15.5)^{2.5}
$$



Figure 4.2 Pipeline with 1200 psig MOP.

Solving for the pressure at Dover, we get

$$
P_{1}=1594 \mathrm{psia}=1579.3 \mathrm{psig}
$$

It can be seen from Figure 4.2 that since the MOP is 1200 psig, we cannot discharge at 1579.3 psig at Dover.

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 4.3.


Figure 4.3 Dover to Leeds pipeline with one compressor station.

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

Using General Flow Equation 2.2,

$$
175 \times 10^{6}=77.54\left(\frac{1}{\sqrt{0.0107}}\right)\left(\frac{520}{14.7}\right)\left(\frac{1214.7^{2}-P_{2}^{2}}{0.6 \times 540 \times 70 \times 0.85}\right)^{0.5} \times(15.5)^{2.5}
$$

Solving for the pressure at Kent (suction pressure),

$$
P_{2}=733 \mathrm{psia}=718 \mathrm{psig}
$$

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

$$
175 \times 10^{6}=77.54\left(\frac{1}{\sqrt{0.0107}}\right)\left(\frac{520}{14.7}\right)\left(\frac{1214.7^{2}-P_{2}^{2}}{0.6 \times 540 \times 70 \times 0.85}\right)^{0.5} \times(15.5)^{2.5}
$$

resulting in a pressure at Leeds of

$$
P_{2}=733 \mathrm{psia}=718 \mathrm{psig}
$$

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

$$
175 \times 10^{6}=77.54\left(\frac{1}{\sqrt{0.0107}}\right)\left(\frac{520}{14.7}\right)\left(\frac{1214.7^{2}-814.7^{2}}{0.6 \times 540 \times L \times 0.85}\right)^{0.5} \times(15.5)^{2.5}
$$

Solving for length $L$, we get

$$
\begin{gathered}
\frac{1214.7^{2}-814.7^{2}}{L}=\frac{1214.7^{2}-733^{2}}{70} \\
L=60.57 \mathrm{miles}
\end{gathered}
$$

Therefore, Kent must be located approximately 61 miles from Leeds. We must now recalculate the suction pressure at the Kent compressor station based on the pipe
length of 79.43 (140 - 60.57) miles between Dover and Kent. From this suction pressure, we must also check the compression ratio.

Using General Flow Equation 2.2 for the pipe segment between Dover and Kent, we get

$$
175 \times 10^{6}=77.54\left(\frac{1}{\sqrt{0.0107}}\right)\left(\frac{520}{14.7}\right)\left(\frac{1214.7^{2}-P_{2}^{2}}{0.6 \times 540 \times 79.43 \times 0.85}\right)^{0.5} \times(15.5)^{2.5}
$$

Solving for $P_{2}$, we get

$$
\frac{1214.7^{2}-P_{2}^{2}}{79.43}=\frac{1214.7^{2}-733^{2}}{70}
$$

or

$$
P_{2}=645.49 \mathrm{psia}=630.79 \mathrm{psig}
$$

Therefore, the suction pressure at $\mathrm{Kent}=630.79$ 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.

Figure 4.4 shows the revised configuration with the new location of the Kent compressor station.


Figure 4.4 Dover to Leeds pipeline with relocated Kent compressor station.

### 4.2 HYDRAULIC BALANCE

In the preceding discussions, we considered each compressor station operating at the same discharge pressure and also considered the same compression ratio. Recalling the definition of compression ratio from Equation 4.1, we can state that each compressor station operates at the same suction and discharge pressures. If there are no intermediate injections or deliveries along the pipeline, as in Example 1, each compressor station is required to compress the same amount of gas. Therefore, with pressures and flow rates being the same, each compressor station will require the same amount of horsepower. This is known as hydraulic balance. In a long pipeline with multiple compressor stations, in which each compressor station adds the same amount of energy to the gas, we say that this is a hydraulically balanced pipeline.

One of the advantages of a hydraulically balanced pipeline is that all compression equipment can be identical, which will reduce inventory of spare parts and minimize maintenance. It is much easier and cheaper to maintain five identical compressor stations of 5000 horsepower each than to maintain two 6000 HP and three 5000 HP compressors. Also, in order to pump the same volume through a pipeline, hydraulically balanced compressor stations will require less total horsepower than if the stations were not located for hydraulic balance.

We will now discuss the different processes by which gas is compressed, such as isothermal, adiabatic (isentropic), and polytropic compression. After that, we will outline the method for calculating horsepower for a compressor station in the subsequent sections.

### 4.3 ISOTHERMAL COMPRESSION

The isothermal compression process is one in which the gas pressure and volume compressed vary in a way that the temperature remains constant. Isothermal compression requires the least amount of work compared to other forms of compression. This process is of theoretical interest since, in reality, maintaining the temperature constant in a gas compressor is virtually impossible.

Figure 4.5 shows the pressure volume diagram for isothermal compression. Point 1 represents the inlet conditions of pressure $\left(P_{1}\right)$, volume $\left(V_{1}\right)$, and at temperature $\left(T_{1}\right)$. Point 2 represents the final compressed conditions of pressure $\left(P_{2}\right)$, volume $\left(V_{2}\right)$, and at constant temperature $\left(T_{1}\right)$.

The relationship between pressure, $P$, and volume, $V$, in an isothermal process is as follows:

$$
\begin{equation*}
P V=C \tag{4.2}
\end{equation*}
$$

where $C$ is a constant.
Therefore, we can state that

$$
\begin{equation*}
P_{1} V_{1}=P_{2} V_{2} \tag{4.3}
\end{equation*}
$$



Figure 4.5 Isothermal compression.

Considering 1 lb of natural gas compressed isothermally, the work done is calculated as follows:

$$
\begin{equation*}
W i=\frac{53.28}{G} T_{1} \log _{e}\left(\frac{P_{2}}{P_{1}}\right) \quad(\text { USCS units }) \tag{4.4}
\end{equation*}
$$

where
$W i=$ isothermal work done, $\mathrm{ft}-\mathrm{lb} / \mathrm{lb}$ of gas
$G=$ gas gravity, dimensionless
$T_{1} \quad=$ suction temperature of gas, ${ }^{\circ} \mathrm{R}$
$P_{1} \quad=$ suction pressure of gas, psia
$P_{2}=$ discharge pressure of gas, psia
$\log _{e}=$ natural logarithm to base $e(e=2.718)$
The ratio $\left(\frac{P_{2}}{P_{1}}\right)$ is also called the compression ratio.
In SI units, the work done in isothermal compression of 1 kg of gas is

$$
\begin{equation*}
W i=\frac{286.76}{G} T_{1} \log _{e}\left(\frac{P_{2}}{P_{1}}\right) \quad \text { (SI units) } \tag{4.5}
\end{equation*}
$$

where
$W i=$ isothermal work done, $\mathrm{J} / \mathrm{kg}$ of gas
$T_{1}=$ suction temperature of gas, K
$P_{1}=$ suction pressure of gas, kPa absolute
$P_{2}=$ discharge pressure of gas, kPa absolute
Other symbols are as defined earlier.

## Example 2

Natural gas is compressed isothermally at $60^{\circ} \mathrm{F}$ from an initial pressure of 500 psig to a pressure of 1000 psig. The gas gravity is 0.6 . Calculate the work done in compressing 5 lb of gas. Use 14.7 psia and $60^{\circ} \mathrm{F}$ for the base pressure and temperature, respectively.

## Solution

Using Equation 4.4, the work done per lb of gas is

$$
W i=\frac{53.28}{0.6}(60+460) \log _{e}\left(\frac{1000+14.7}{500+14.7}\right)=31,343 \mathrm{ft}-\mathrm{lb} / \mathrm{lb}
$$

The total work done in compressing 5 lb of gas is

$$
W_{T}=31,343 \times 5=156,715 \mathrm{ft}-\mathrm{lb}
$$

## Example 3

Calculate the work done in compressing 2 kg of gas (gravity $=0.65$ ) isothermally at $20^{\circ} \mathrm{C}$ from 700 kPa to 2000 kPa . Use 101 kPa and $15^{\circ} \mathrm{C}$ for the base pressure and temperature, respectively.

## Solution

Using Equation 4.5, the work done in isothermal compression of 1 kg of gas is

$$
W i=\frac{286.76}{0.65}(273+20) \log _{e}\left(\frac{2000+101}{700+101}\right)=124,649 \mathrm{~J} / \mathrm{kg}
$$

Therefore, the total work done in compressing 2 kg of gas is

$$
W_{T}=124,649 \times 2=249,298 \mathrm{~J}
$$

### 4.4 ADIABATIC COMPRESSION

The adiabatic compression process is characterized by zero heat transfer between the gas and the surroundings. The terms adiabatic and isentropic are used synonymously, although isentropic really means "constant entropy." An adiabatic process that is also frictionless is referred to as isentropic. In an adiabatic compression process, the relationship between pressure and volume is as follows:

$$
\begin{equation*}
P V^{\gamma}=C \tag{4.6}
\end{equation*}
$$

where
$\gamma=$ ratio of specific heats of gas, $\frac{C_{p}}{C_{v}}$
$C_{p}=$ specific heats of gas at constant pressure
$C_{v}=$ specific heats of gas at constant volume
$C=$ a constant, different from the one for isothermal compression in Equation 4.2
$\gamma$ is also known as the adiabatic or isentropic exponent for the gas. It ranges in value from 1.2 to 1.4.

Therefore, we can state that

$$
\begin{equation*}
P_{1} V_{1}^{\gamma}=P_{2} V_{2}^{\gamma} \tag{4.7}
\end{equation*}
$$

Figure 4.6 shows adiabatic compression similar to the $P$ - $V$ diagram for isothermal compression.

Considering 1 lb of natural gas compressed adiabatically, the work done is calculated as follows:

$$
\begin{equation*}
W a=\frac{53.28}{G} T_{1}\left(\frac{\gamma}{\gamma-1}\right)\left[\left(\frac{P_{2}}{P_{1}}\right)^{\frac{\gamma-1}{\gamma}}-1\right] \quad \text { (USCS units) } \tag{4.8}
\end{equation*}
$$

where
$W a=$ adiabatic work done, $\mathrm{ft}-\mathrm{lb} / \mathrm{lb}$ of gas
$G=$ gas gravity, dimensionless
$T_{1}=$ suction temperature of gas, ${ }^{\circ} \mathrm{R}$
$\gamma=$ ratio of specific heats of gas, dimensionless
$P_{1}=$ suction pressure of gas, psia
$P_{2}=$ discharge pressure of gas, psia


Figure 4.6 Adiabatic compression.

In SI units, the work done in adiabatic compression of 1 kg of gas is

$$
\begin{equation*}
W a=\frac{286.76}{G} T_{1}\left(\frac{\gamma}{\gamma-1}\right)\left[\left(\frac{P_{2}}{P_{1}}\right)^{\frac{\gamma-1}{\gamma}}-1\right] \quad \text { (SI units) } \tag{4.9}
\end{equation*}
$$

where
$W a=$ adiabatic work done, $\mathrm{J} / \mathrm{kg}$ of gas
$T_{1}=$ suction temperature of gas, K
$P_{1}=$ suction pressure of gas, kPa absolute
$P_{2}=$ discharge pressure of gas, kPa absolute
Other symbols are as defined earlier.

## Example 4

Natural gas is compressed adiabatically from an initial temperature and pressure of $60^{\circ} \mathrm{F}$ and 500 psig , respectively, to a final pressure of 1000 psig . The gas gravity is 0.6 and the ratio of specific heat is 1.3 . Calculate the work done in compressing 5 lb of gas. Use 14.7 psia and $60^{\circ} \mathrm{F}$ for the base pressure and temperature, respectively.

## Solution

Using Equation 4.8, the work done in adiabatic compression is

$$
W a=\frac{53.28}{0.6}(60+460)\left(\frac{1.3}{0.3}\right)\left[\left(\frac{1014.7}{514.7}\right)^{\frac{0.3}{1.3}}-1\right]=33,931 \mathrm{ft}-\mathrm{lb} / \mathrm{lb}
$$

Therefore, the total work done in compressing 5 lb of gas is

$$
W_{T}=33,931 \times 5=169,655 \mathrm{ft}-\mathrm{lb}
$$

## Example 5

Calculate the work done in compressing 2 kg of gas (gravity $=0.65$ ) adiabatically from an initial temperature of $20^{\circ} \mathrm{C}$ and pressure of 700 kPa to a final pressure of 2000 kPa . The specific heat ratio of gas is 1.4 and the base pressure and base temperature are 101 kPa and $15^{\circ} \mathrm{C}$, respectively.

## Solution

Using Equation 4.9, the work done in adiabatic compression of 1 kg of gas is

$$
W a=\frac{286.76}{0.65}(20+273)\left(\frac{1.4}{0.4}\right)\left[\left(\frac{2000+101}{700+101}\right)^{\frac{0.4}{1.4}}-1\right]=143,512 \mathrm{~J} / \mathrm{kg}
$$

Therefore, the total work done in compressing 2 kg of gas is

$$
W_{T}=143,512 \times 2=287,024 \mathrm{~J}
$$

### 4.5 POLYTROPIC COMPRESSION

Polytropic compression is similar to adiabatic compression, but there is no requirement of zero heat transfer as in adiabatic compression. In a polytropic process, the relationship between pressure and volume is as follows:

$$
\begin{equation*}
P V^{n}=C \tag{4.10}
\end{equation*}
$$

where
$n=$ polytropic exponent
$C=$ a constant, different from the one for isothermal or adiabatic compression in Equation 4.2 and Equation 4.6

Therefore, we can state that

$$
\begin{equation*}
P_{1} V_{1}^{n}=P_{2} V_{2}^{n} \tag{4.11}
\end{equation*}
$$

Since polytropic compression is similar to adiabatic compression, we can easily calculate the work done in polytropic compression by substituting $n$ for $\gamma$ in Equation 4.8 and Equation 4.9.

## Example 6

Natural gas is compressed polytropically from an initial temperature and pressure of $60^{\circ} \mathrm{F}$ and 500 psig , respectively, to a final pressure of 1000 psig . The gas gravity is 0.6 and the base pressure and base temperature are 14.7 psia and $60^{\circ} \mathrm{F}$, respectively. Calculate the work done in compressing 5 lb of gas using a polytropic exponent of 1.5 .

## Solution

Polytropic compression is similar to adiabatic compression, and, therefore, the same equation can be used for work done, substituting the polytropic exponent $n$ for the adiabatic exponent $\gamma$ (the ratio of specific heat).

Using Equation 4.8, the work done in polytropic compression of 1 lb of gas is

$$
W p=\frac{53.28}{0.6}(60+460)\left(\frac{1.5}{0.5}\right)\left[\left(\frac{1014.7}{514.7}\right)^{\frac{0.5}{1.5}}-1\right]=35,168 \mathrm{ft}-\mathrm{lb} / \mathrm{lb}
$$

Therefore, the total work done in compressing 5 lb of gas is

$$
W_{T}=35,168 \times 5=175,840 \mathrm{ft}-\mathrm{lb}
$$

## Example 7

Calculate the work done in compressing 2 kg of gas (gravity $=0.65$ ) polytropically from an initial temperature of $20^{\circ} \mathrm{C}$ and pressure of 700 kPa to a final pressure of 2000 kPa . Use a polytropic exponent of 1.5 . The base pressure and base temperature are 101 kPa and $15^{\circ} \mathrm{C}$, respectively.

## Solution

Using Equation 4.9, and substituting the polytropic exponent 1.5 in place of $\gamma$, the work done in polytropic compression is

$$
W p=\frac{286.76}{0.65}(20+273)\left(\frac{1.5}{0.5}\right)\left[\left(\frac{2000+101}{700+101}\right)^{\frac{0.5}{1.5}}-1\right]=146,996 \mathrm{~J} / \mathrm{kg}
$$

Therefore, the total work done in compressing 2 kg of gas is

$$
W_{T}=146,996 \times 2=293,992 \mathrm{~J}
$$

### 4.6 DISCHARGE TEMPERATURE OF COMPRESSED GAS

In adiabatic or polytropic compression of natural gas, we can determine the final temperature of the gas knowing the initial temperature and initial and final pressures.

Using Equation 4.6 for adiabatic compression and the perfect gas law, by eliminating the volume, $V$, we can write the following:

$$
\begin{equation*}
\left(\frac{T_{2}}{T_{1}}\right)=\left(\frac{Z_{1}}{Z_{2}}\right)\left(\frac{P_{2}}{P_{1}}\right)^{\frac{\gamma-1}{\gamma}} \tag{4.12}
\end{equation*}
$$

where
$T_{1}=$ suction temperature of gas, ${ }^{\circ} \mathrm{R}$
$T_{2}=$ discharge temperature of gas, ${ }^{\circ} \mathrm{R}$
$Z_{1}=$ gas compressibility factor at suction, dimensionless
$Z_{2}=$ gas compressibility factor at discharge, dimensionless
Other symbols are as defined earlier.
Similarly, for polytropic compression, the discharge temperature can be calculated from the following equation:

$$
\begin{equation*}
\left(\frac{T_{2}}{T_{1}}\right)=\left(\frac{Z_{1}}{Z_{2}}\right)\left(\frac{P_{2}}{P_{1}}\right)^{\frac{n-1}{n}} \tag{4.13}
\end{equation*}
$$

where all symbols are as defined earlier.

## Example 8

Gas is compressed adiabatically $(\gamma=1.4)$ from $60^{\circ} \mathrm{F}$ suction temperature and a compression ratio of 2.0. Calculate the discharge temperature, assuming $Z_{1}=0.99$ and $Z_{2}=0.85$.

Solution

Using Equation 4.12,

$$
\begin{aligned}
& \left(\frac{T_{2}}{60+460}\right)=\left(\frac{0.99}{0.85}\right)(2.0)^{\frac{0.4}{1.4}}=1.4198 \\
& T_{2}=1.4198 \times 520=738.3^{\circ} \mathrm{R}=278.3^{\circ} \mathrm{F}
\end{aligned}
$$

### 4.7 HORSEPOWER 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 $(H P)$, 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.

Suppose the gas flow rate is $Q$ measured in standard $\mathrm{ft}^{3}$ per day (SCFD), and the suction and discharge pressures of the compressor station are $P_{s}$ and $P_{d}$, respectively. The compressor station adds the differential pressure of $\left(P_{d}-P_{s}\right)$ psia to the gas flowing at $Q$ SCFD. Therefore, the rate at which energy is supplied to the gas is $\left(P_{d}-P_{s}\right) \times Q \times$ Const 1 , where Const 1 is a constant depending upon the units employed.

This is a very simplistic approach, since the gas properties vary with temperature and pressure. Also, the compressibility factor and the type of gas compression (adiabatic or polytropic) must be taken into account. Therefore, the calculation for $H P$ will be approached from another angle in what follows.

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. The equation for horsepower can be expressed as follows:

$$
\begin{equation*}
H P=\frac{M \times \Delta H}{\eta} \tag{4.14}
\end{equation*}
$$

where
$H P=$ compressor horsepower
$M=$ mass flow rate of gas, $\mathrm{lb} / \mathrm{min}$
$\Delta H=$ compressor head, $\mathrm{ft}-\mathrm{lb} / \mathrm{lb}$
$\eta$ = compressor efficiency, decimal value
Another more commonly used formula for compressor horsepower that takes into account the compressibility of gas is as follows:

$$
\begin{equation*}
H P=0.0857\left(\frac{\gamma}{\gamma-1}\right) Q T_{1}\left(\frac{Z_{1}+Z_{2}}{2}\right)\left(\frac{1}{\eta_{a}}\right)\left[\left(\frac{P_{2}}{P_{1}}\right)^{\frac{\gamma-1}{\gamma}}-1\right] \tag{4.15}
\end{equation*}
$$

where
$H P=$ compressor horsepower
$\gamma=$ ratio of specific heats of gas, dimensionless
$Q=$ gas flow rate, MMSCFD
$T_{1}=$ suction temperature of gas, ${ }^{\circ} \mathrm{R}$
$P_{1}=$ 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
$\eta_{a}=$ compressor adiabatic (isentropic) efficiency, decimal value
In SI units, the Power equation is as follows:

$$
\begin{equation*}
\text { Power }=4.0639\left(\frac{\gamma}{\gamma-1}\right) Q T_{1}\left(\frac{Z_{1}+Z_{2}}{2}\right)\left(\frac{1}{\eta_{a}}\right)\left[\left(\frac{P_{2}}{P_{1}}\right)^{\frac{\gamma-1}{\gamma}}-1\right] \tag{4.16}
\end{equation*}
$$

where
Power $=$ compression Power, kW
$\gamma \quad=$ ratio of specific heats of gas, dimensionless
$Q \quad=$ gas flow rate, $\mathrm{Mm}^{3} /$ day
$T_{1} \quad=$ suction temperature of gas, K
$P_{1} \quad=$ suction pressure of gas, kPa
$P_{2} \quad=$ discharge pressure of gas, kPa
$Z_{1} \quad=$ compressibility of gas at suction conditions, dimensionless
$Z_{2} \quad=$ compressibility of gas at discharge conditions, dimensionless
$\eta_{a} \quad=$ compressor adiabatic (isentropic) efficiency, decimal value
The adiabatic efficiency $\eta_{a}$ generally ranges from 0.75 to 0.85 . By considering a mechanical efficiency $\eta_{m}$ of the compressor driver, we can calculate the brake horsepower ( $B H P$ ) required to run the compressor as follows:

$$
\begin{equation*}
B H P=\frac{H P}{\eta_{m}} \tag{4.17}
\end{equation*}
$$

where $H P$ is the horsepower calculated from the preceding equations, taking into account the adiabatic efficiency $\eta_{a}$ of the compressor. The mechanical efficiency $\eta_{m}$ of the driver can range from 0.95 to 0.98 . The overall efficiency, $\eta_{o}$, is defined as the product of the adiabatic efficiency, $\eta_{a}$, and the mechanical efficiency, $\eta_{m}$ :

$$
\begin{equation*}
\eta_{o}=\eta_{a} \times \eta_{m} \tag{4.18}
\end{equation*}
$$

From the adiabatic compression Equation 4.6, eliminating the volume $V$, the discharge temperature of the gas is related to the suction temperature and the compression ratio by means of the following equation:

$$
\begin{equation*}
\left(\frac{T_{2}}{T_{1}}\right)=\left(\frac{P_{2}}{P_{1}}\right)^{\frac{\gamma-1}{\gamma}} \tag{4.19}
\end{equation*}
$$

The adiabatic efficiency, $\eta_{a}$, 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 theoretical adiabatic temperature rise is obtained from the adiabatic pressuretemperature relationship as follows, considering the gas compressibility factors similar to Equation 4.12:

$$
\begin{equation*}
\left(\frac{T_{2}}{T_{1}}\right)=\left(\frac{Z_{1}}{Z_{2}}\right)\left(\frac{P_{2}}{P_{1}}\right)^{\frac{\gamma-1}{\gamma}} \tag{4.20}
\end{equation*}
$$

or

$$
\begin{equation*}
T_{2}=T_{1}\left(\frac{Z_{1}}{Z_{2}}\right)\left(\frac{P_{2}}{P_{1}}\right)^{\frac{\gamma-1}{\gamma}} \tag{4.21}
\end{equation*}
$$

Therefore, the theoretical adiabatic temperature rise is

$$
T_{1}\left(\frac{Z_{1}}{Z_{2}}\right)\left(\frac{P_{2}}{P_{1}}\right)^{\frac{\gamma-1}{\gamma}}-T_{1}
$$

Therefore, the adiabatic efficiency is

$$
\begin{equation*}
\eta_{a}=\frac{T_{1}\left(\frac{Z_{1}}{Z_{2}}\right)\left(\frac{P_{2}}{P_{1}}\right)^{\frac{\gamma-1}{\gamma}}-T_{1}}{T_{2}-T_{1}} \tag{4.22}
\end{equation*}
$$

where $T_{2}$ is the actual discharge temperature of the gas.

Simplifying, we get

$$
\begin{equation*}
\eta_{a}=\left(\frac{T_{1}}{T_{2}-T_{1}}\right)\left[\left(\frac{Z_{1}}{Z_{2}}\right)\left(\frac{P_{2}}{P_{1}}\right)^{\frac{\gamma-1}{\gamma}}-1\right] \tag{4.23}
\end{equation*}
$$

For example, if the inlet gas temperature is $80^{\circ} \mathrm{F}$ and the suction and discharge pressures are 800 psia and 1400 psia, respectively, we can calculate the adiabatic efficiency if the outlet temperature is given as $200^{\circ} \mathrm{F}$. Using $\gamma=1.4$, and from Equation 4.23, the adiabatic efficiency is, assuming compressibility factors to be equal to 1.0 ,

$$
\begin{equation*}
\eta_{a}=\left(\frac{80+460}{200-80}\right)\left[\left(\frac{1400}{800}\right)^{\frac{1.4-1}{1.4}}-1\right]=0.7802 \tag{4.24}
\end{equation*}
$$

Thus, the adiabatic compression efficiency is 0.7802 .

## Example 9

Calculate the compressor horsepower required for an adiabatic compression of 106 MMSCFD gas with inlet temperature of $68^{\circ} \mathrm{F}$ 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 $B H P$ is required? Calculate the outlet temperature of the gas.

## Solution

From Equation 4.15, the horsepower required is

$$
H P=0.0857 \times 106\left(\frac{1.40}{0.40}\right)(68+460)\left(\frac{1+0.85}{2}\right)\left(\frac{1}{0.8}\right)\left[\left(\frac{1305}{725}\right)^{\frac{0.40}{1.40}}-1\right]=3550
$$

Using Equation 4.17, we calculate the driver horsepower required based on a mechanical efficiency of 0.95 :

$$
B H P \text { required }=\frac{3550}{0.95}=3737
$$

The outlet temperature of the gas is found from Equation 4.23 after transposing as follows:

$$
T_{2}=(68+460) \times\left[\frac{\left(\frac{1}{0.85}\right)\left(\frac{1305}{725}\right)^{\frac{0.4}{1.4}}-1}{0.8}\right]+(68+460)=786.46^{\circ} \mathrm{R}=326.46^{\circ} \mathrm{F}
$$

The discharge temperature of the gas is $326.46^{\circ} \mathrm{F}$.

## Example 10

Natural gas at $3 \mathrm{Mm}^{3} /$ day and $20^{\circ} \mathrm{C}$ is compressed isentropically $(\gamma=1.4)$ from a suction pressure of 5 MPa absolute to a discharge pressure of 9 MPa absolute in a centrifugal compressor with an isentropic efficiency of 0.80 . Calculate the compressor power required, assuming the compressibility factors at suction and discharge conditions to be $Z_{1}=0.95$ and $Z_{2}=0.85$, respectively. If the mechanical efficiency of the compressor driver is 0.95 , what is the driver power required? Calculate the outlet temperature of the gas.

## Solution

From Equation 4.16, the power required is

$$
\text { Power }=4.0639 \times 3\left(\frac{1.40}{0.40}\right)(20+273)\left(\frac{0.95+0.85}{2}\right)\left(\frac{1}{0.8}\right)\left[\left(\frac{9}{5}\right)^{\frac{0.40}{1.40}}-1\right]=2572 \mathrm{~kW}
$$

$$
\text { Power }=2572 \mathrm{~kW}
$$

Using Equation 4.17, we calculate the driver power required as follows:

$$
\text { Driver power required }=\frac{2572}{0.95}=2708 \mathrm{~kW}
$$

The outlet temperature of the gas is found from Equation 4.23 as follows:

$$
T_{2}=\frac{20+273}{0.8} \times\left[\left(\frac{0.95}{0.85}\right)\left(\frac{9}{5}\right)^{\frac{0.4}{1.4}}-1\right]+(20+273)=410.94 \mathrm{~K}=137.94^{\circ} \mathrm{C}
$$

### 4.8 OPTIMUM COMPRESSOR LOCATIONS

In the foregoing discussion, we looked at a two-compressor station configuration for gas deliveries from Dover to the Leeds power plant. In this section, we will consider various locations of the intermediate compressor stations on a long-distance gas transmission pipeline to arrive at the optimum locations, taking into account the overall horsepower required. In Section 4.2, we discussed hydraulic balance. The advantage of locating the intermediate compressor station, such that the same amount of energy is added to the gas at each compressor station, was pointed out.

In the next example, we will analyze optimum compressor locations by considering both hydraulically balanced and unbalanced compressor station locations.

## Example 11

A gas transmission pipeline is 240 mi long, NPS $30,0.500 \mathrm{in}$. wall thickness, with an origin compressor station at Payson and two intermediate compressor stations


Figure 4.7 Gas pipeline with three compressor stations.
tentatively located at Williams (milepost 80) and Snowflake (milepost 160), as shown in Figure 4.7. There are no intermediate flow deliveries or injections, and the inlet flow rate of 900 MMSCFD at Payson equals the delivery flow rate at Douglas. The delivery pressure required at Douglas is 600 psig and the MOP of the pipeline is 1400 psig throughout. Neglect the effects of elevation and assume constant gas flow temperature of $80^{\circ} \mathrm{F}$ and constant values of transmission factor $F=20$ and compressibility factor $Z=0.85$ throughout the pipeline. The gas gravity $=0.6$, base pressure $=$ 14.7 psia , and base temperature $=60^{\circ} \mathrm{F}$. Use a polytropic compression coefficient of 1.38 and a compression efficiency of 0.9 .

## Solution

Neglecting the effects of elevation, we could calculate for each of the three segmentsPayson to Williams, Williams to Snowflake, and Snowflake to Douglas-the downstream pressure starting with an upstream pressure of 1400 psig. Thus, using the General Flow equation for the Payson to Williams segment, we would calculate the downstream pressure at Williams starting with a pressure of 1400 psig at Payson. This downstream pressure is actually the suction pressure at the Williams compressor station. Next, in a similar fashion, we would calculate the downstream pressure at Snowflake, for the second segment from Williams to Snowflake, based on an upstream pressure of 1400 psig at Williams. This downstream pressure is actually the suction pressure at the Snowflake compressor station. Finally, we would calculate the downstream pressure at Douglas, for the third segment from Snowflake to Douglas, based on an upstream pressure of 1400 psig at Snowflake. This final pressure is the delivery pressure at the Douglas terminus. We have thus calculated the suction pressures at each of the two intermediate compressor stations at Williams and Snowflake and also calculated the final delivery pressure at Douglas. This pressure calculated at Douglas may or may not be equal to the desired delivery pressure of 600 psig , since we performed a forward calculation going from Payson to Douglas. Therefore, since the delivery pressure is usually a desired or contracted value, we will have to adjust the location of the last compressor station at Snowflake to achieve the desired delivery pressure at Douglas.

Another approach would be to perform a backward calculation starting at Douglas and proceeding toward Payson. In this case, we will start with segment 3 and calculate the location of the Snowflake compressor station that will result in an upstream pressure of 1400 psig at Snowflake. Thus, we locate the Snowflake compressor station that will cause a discharge pressure of 1400 psig at Snowflake and a delivery pressure of 600 psig at Douglas. Having located the Snowflake compressor station, we can now recalculate the suction pressure at Snowflake by considering the pipe segment 2 and using an upstream pressure of 1400 psig at Williams. We will not have to repeat calculations for segment 1 , since the location of Williams has not changed and, therefore, the suction pressure at Williams will remain the same as the previously calculated value. We have thus been able to determine the pressures along the pipeline with the given three-compressor-station configuration such that the desired delivery pressure at Douglas has been achieved and each compressor station discharges at an MOP value of 1400 psig. But are these the optimum locations of the intermediate compressor stations Williams and Snowflake? And are all compressor stations in hydraulic balance? We can state that these compressor stations are optimized and are in hydraulic balance only if each compressor station operates at the same compression ratio and, therefore, adds the same amount of horsepower to the gas at each compressor station. The locations of Williams and Snowflake may not result in the same suction pressures even though the discharge pressures are the same. Therefore, chances are that Williams might be operating at a lower compression ratio than Snowflake or Payson, or vice versa, which will not result in hydraulic balance. However, if the compression ratios are close enough that the required compressor sizes are the same, we could still be in hydraulic balance and the stations could be at optimum locations.

Next, perform the actual calculations and determine how much tweaking of the compressor station locations is required to optimize these stations.

First, we will perform the backward calculations for segment 3, starting with a downstream pressure of 600 psig at Douglas and an upstream pressure of 1400 psig at Snowflake. With these constraints, we will calculate the pipe length, $L$, miles between Snowflake and Douglas.

Using General Flow Equation 2.4, neglecting elevations,

$$
900 \times 10^{6}=38.77 \times 20.0\left(\frac{520}{14.7}\right)\left(\frac{1414.7^{2}-614.7^{2}}{0.6 \times 540 \times L \times 0.85}\right)^{0.5}(29)^{2.5}
$$

Solving for pipe length, we get

$$
L=112.31 \mathrm{mi}
$$

Therefore, in order to discharge at 1400 psig at Snowflake and deliver gas at 600 psig at Douglas, the Snowflake compressor station will be located at a distance of 112.31 mi upstream of Douglas-or at milepost $(240-112.31)=127.69$ measured from Payson.

Next, keeping the location of the Williams compressor at milepost 80 , we will calculate the downstream pressure at Snowflake for pipe segment 2 starting at 1400 psig at Williams. This calculated pressure will be the suction pressure of the Snowflake compressor station.

Using General Flow Equation 2.4, neglecting elevations,

$$
900 \times 10^{6}=38.77 \times 20.0\left(\frac{520}{14.7}\right)\left(\frac{1414.7^{2}-P_{2}^{2}}{0.6 \times 540 \times 47.69 \times 0.85}\right)^{0.5}(29)^{2.5}
$$

where the pipeline segment length between Williams and Snowflake was calculated as

$$
127.69-80=47.69 \mathrm{mi}
$$

Solving for suction pressure at Snowflake, we get

$$
P_{2}=1145.42 \mathrm{psia}=1130.72 \mathrm{psig}
$$

Therefore, the compression ratio at Snowflake is $\frac{1414.7}{1145.42}=1.24$.
Next, for pipe segment 1 between Payson and Williams, we will calculate the downstream pressure at Williams, starting at 1400 psig at Payson. This calculated pressure will be the suction pressure of the Williams compressor station.

Using General Flow Equation 2.4, neglecting elevations,

$$
900 \times 10^{6}=38.77 \times 20.0\left(\frac{520}{14.7}\right)\left(\frac{1414.7^{2}-P_{2}^{2}}{0.6 \times 540 \times 80 \times 0.85}\right)^{0.5}(29)^{2.5}
$$

Solving for suction pressure at Williams, we get

$$
P_{2}=919.20 \mathrm{psia}=904.5 \mathrm{psig}
$$

$$
\text { Therefore, the compression ratio at Williams }=\frac{1414.7}{919.2}=1.54
$$

Therefore, from the foregoing calculations, the compressor station at Williams requires a compression ratio $r=1.54$, whereas the compressor station at Snowflake requires a compression ratio $r=1.24$. Obviously, this is not a hydraulically balanced compressor station system. Further, we do not know what the suction pressure is at the Payson compressor station. If we assume that Payson receives gas at approximately the same suction pressure as Williams ( 905 psig ), both the Payson and Williams compressor stations will have the same compression ratio of 1.54 . In this case, the Snowflake compressor station will be the odd one, operating at a compression ratio of 1.24. How do we balance these compressor stations? One way would be to obtain the same compression ratios for all three compressor stations by simply relocating the Snowflake compressor station toward Douglas such that its suction pressure will drop from 1131 psig to 905 psig while keeping the discharge at Snowflake at 1400 psig. This will then ensure that all three compressor stations will be operating at the following suction and discharge pressures and compression ratios:

$$
\begin{aligned}
\text { Suction pressure } P_{s} & =904.5 \mathrm{psig} \\
\text { Discharge pressure } P_{d} & =1400 \mathrm{psig} \\
\text { Compression ratio } r & =\frac{1400+14.7}{904.5+14.7}=1.54
\end{aligned}
$$



NPS 30 pipeline 240 mi long

Figure 4.8 Pressure regulation at Douglas.

However, because the Snowflake compressor station is now located closer to Douglas than before (127.69), the discharge pressure of 1400 psig at Snowflake will result in a higher delivery pressure at Douglas than the required 600 psig , as shown in Figure 4.8.

If the additional pressure at Douglas can be tolerated by the customer, then there will be no problem. But if the customer requires no more than 600 psig , we have to reduce the delivery pressure to 600 psig by installing a pressure regulator at Douglas, as shown in Figure 4.8. Therefore, by balancing the compressor station locations, we have also created a problem of getting rid of the extra pressure at the delivery point. Pressure regulation means wasted horsepower. The advantage of the balanced compressor stations vs. the negative aspect of the pressure regulation must be factored into the decision process.

To illustrate this pressure regulation scenario, we will now determine the revised location of the Snowflake compressor station for hydraulic balance. We will calculate the length of pipe segment 2 by assuming 1400 psig discharge pressure at Williams and a suction pressure of 904.5 psig at Snowflake.

Using General Flow Equation 2.4, neglecting elevations,

$$
900 \times 10^{6}=38.77 \times 20.0\left(\frac{520}{14.7}\right)\left(\frac{1414.7^{2}-919.2^{2}}{0.6 \times 540 \times L \times 0.85}\right)^{0.5}(29)^{2.5}
$$

Solving for pipe length for segment 2 , we get

$$
L=80 \mathrm{mi}
$$

Therefore, the Snowflake compressor station should be located at a distance of 80 mi from Williams or at milepost 160 . We could have arrived at this without the above calculations, since elevations are neglected and the Payson to Williams pressure
profile will be the same as the pressure profile from Williams to Snowflake. With the Snowflake compressor station located at milepost 160 , and discharging at 1400 psig , we conclude that the delivery pressure at Douglas will also be 904.5 psig , since all three pipe segments are hydraulically the same. We see that the delivery pressure at Douglas is approximately 305 psig more than the desired pressure. As indicated earlier, a pressure regulator will be required at Douglas to reduce the delivery pressure to 600 psig . We can compare the hydraulically balanced scenario with the previously calculated case where Payson and Williams operate at a compression ratio of 1.54 and Snowflake operates at lower compression ratio of 1.24. By applying approximate cost per installed horsepower, we can compare these two cases. First, using Equation 4.15, calculate the horsepower required at each compressor station, assuming polytropic compression and a compression ratio of 1.54 for a balanced compressor station:

$$
H P=0.0857 \times 900 \times\left(\frac{1.38}{0.38}\right)(80+460)\left(\frac{1+0.85}{2}\right)\left(\frac{1}{0.9}\right)\left[(1.54)^{\frac{0.38}{1.38}}-1\right]=19,627
$$

Therefore, the total horsepower required in the hydraulically balanced case is

$$
\text { Total } H P=3 \times 19,627=58,881
$$

At a cost of $\$ 2000$ per installed $H P$,

$$
\text { Total } H P \text { cost }=\$ 2000 \times 58,881=\$ 117.76 \text { million }
$$

In the hydraulically unbalanced case, the Payson and Williams compressor stations will operate at a compression ratio of 1.54 each, whereas the Snowflake compressor station will require a compression ratio of 1.24 .

Using Equation 4.15, the horsepower required at the Snowflake compressor station is

$$
H P=0.0857 \times 900 \times\left(\frac{1.38}{0.38}\right)(80+460)\left(\frac{1+0.85}{2}\right)\left(\frac{1}{0.9}\right)\left[(1.24)^{\frac{0.38}{1.38}}-1\right]=9487
$$

Therefore, the total horsepower required in the hydraulically unbalanced case is

$$
\text { Total } H P=(2 \times 19,627)+9487=48,741
$$

At a cost of $\$ 2000$ per installed $H P$,

$$
\text { Total } H P \text { cost }=\$ 2000 \times 48,741=\$ 97.48 \text { million }
$$

The hydraulically balanced case requires $10,140(58,881-48,741) H P$ more and will cost approximately $\$ 20.28$ ( $\$ 117.76$ - $\$ 97.48$ ) million more. In addition to the extra $H P$ cost, the hydraulically balanced case will require a pressure regulator that will waste energy and result in extra equipment cost. Therefore, the advantages of using identical components, by reducing spare parts and inventory in the hydraulically balanced case, must be weighed against the additional cost. It may not be worth spending the extra $\$ 20$ million to obtain this benefit. The preferred solution in this case is for the Payson and Williams compressor stations to be identical (compression ratio $=1.54$ ) and the Snowflake compressor station to be a smaller one (compression ratio $=1.24$ ), requiring the lower compression ratio and horsepower, to provide the required 600 psig delivery pressure at Douglas.


Figure 4.9 Compressors in series.

### 4.9 COMPRESSORS IN SERIES AND PARALLEL

When compressors operate in series, each unit compresses the same amount of gas but at different compression ratios, such that the overall pressure increase of the gas is achieved in stages, as shown in Figure 4.9.

It can be seen from Figure 4.9 that the first compressor compresses gas from a suction pressure of 900 psia to 1080 psia at a compression ratio of 1.2. The second compressor takes the same volume and compresses it from 1080 psia to a discharge pressure of $1080 \times 1.2=1296 \mathrm{psia}$. Thus, the overall compression ratio of the two identical compressors in series is $1296 / 900=1.44$. We have thus achieved the increase in pressure in two stages. At the end of each compression cycle, the gas temperature rises to some value calculated in accordance with Equation 4.19. Therefore, with multiple stages of compression, unless the gas is cooled between stages, the final gas temperature may be too high. High gas temperatures are not desirable, since the throughput capability of a gas pipeline decreases with gas flow temperature. Therefore, with compressors in series, the gas is cooled to the original suction temperature between each stage of compression, such that the final temperature at the end of all compressors in series is not exceedingly high. Suppose the calculated discharge temperature of a compressor is $232^{\circ} \mathrm{F}$, starting at a $70^{\circ} \mathrm{F}$ suction temperature and with a compression ratio of 1.4. If two of these compressors were in series and there were no cooling between compressions, the final gas temperature would reach approximately

$$
\frac{(232+460)(232+460)}{70+460}=903.5^{\circ} \mathrm{R}=443.5^{\circ} \mathrm{F}
$$

This is too high a temperature for pipeline transportation. On the other hand, if we cool the gas back to $70^{\circ} \mathrm{F}$ before compressing it through the second compressor, the final temperature of the gas coming out of the second compressor will be $232^{\circ} \mathrm{F}$, approximately. We will discuss compressors in series in more detail in the subsequent section.


Figure 4.10 Compressors in parallel.
Compressors are installed in parallel so that large volumes necessary can be provided by multiple compressors, each producing the same compression ratio. Three identical compressors with compression ratio of 1.4 can be used to provide a 900 MMSCFD gas flow from a suction pressure of 900 psia In this example, each compressor will compress 300 MMSCFD from 900 psia to a discharge pressure of

$$
P_{2}=900 \times 1.4=1260 \mathrm{psia}
$$

This is illustrated schematically in Figure 4.10.
Unlike compressors in series, the discharge temperature of the gas coming out of the parallel bank of compressors will not be high, since the gas does not undergo multiple compression ratios. The gas temperature on the discharge side of each parallel compressor will be the same as that of a single compressor with the same compression ratio. Therefore, three parallel compressors, each compressing the same volume of gas at a compression ratio of 1.4 , will have a final discharge temperature of $232^{\circ} \mathrm{F}$, starting from a suction temperature of $70^{\circ} \mathrm{F}$. Gas cooling is required at these temperatures in order to achieve efficient gas transportation and also operate at temperatures not exceeding the limits of the pipe coating material. Generally, pipe coating requires the gas temperature not to exceed 140 to $150^{\circ} \mathrm{F}$.

The compression ratio was defined earlier as the ratio of the discharge pressure to the suction pressure. The higher the compression ratio, the higher will be the gas discharge temperature, in accordance with Equation 4.19.

Consider a suction temperature of $80^{\circ} \mathrm{F}$ and the suction and discharge pressures of 900 psia and 1400 psia , respectively. The compression ratio is $1400 / 900=1.56$. Using Equation 4.19, the discharge temperature will be

$$
\begin{aligned}
& \left(\frac{T_{2}}{80+460}\right)=\left(\frac{1400}{900}\right)^{\frac{1.3-1}{1.3}} \\
& T_{2}=598.36^{\circ} \mathrm{R} \text { or } 138.36^{\circ} \mathrm{F}
\end{aligned}
$$

If the compression ratio is increased to 2.0 , the discharge temperature will become $173.67^{\circ} \mathrm{F}$. It can be seen that the discharge temperature of the gas increases
considerably with the compression ratio. Since the throughput capacity of a gas pipeline decreases with gas temperature, we must find a way to reduce the high gas temperature resulting from gas compression. In previous chapters, we solved many problems with a constant gas inlet temperature of $80^{\circ} \mathrm{F}$. In order to maintain throughput, cooling should be provided on the discharge of the compressor.

We prefer centrifugal compressors used in gas pipeline applications to have a compression ratio of 1.5 to 2.0 ; there may be instances in which higher compression ratios are required due to lower gas receipt pressures and higher pipeline discharge pressures to enable a given volume of gas to be transported through a pipeline. Reciprocating compressors are designed to provide higher compression ratios. However, manufacturers limit maximum compression ratios to a range of 4 to 6 . This is due to high forces that are exerted on the compressor components, which cause expensive material requirements as well as complicated safety needs.

Suppose a compressor is required to provide gas at 1500 psia from gas that is received at 200 psia This requires an overall compression ratio of 7.5. Since this is beyond the acceptable range of compression ratios, we will have to provide this compression in stages. If we provide the necessary pressure by using two compressors in series, each compressor will require to be at a compression ratio of $\sqrt{7.5}$, or approximately 2.74 . The first compressor raises the pressure from 200 psia to $200 \times 2.74=548 \mathrm{psia}$. The second compressor will then boost the gas pressure from 548 psia to $548 \times 2.74=1500 \mathrm{psia}$, approximately. In general, if $n$ compressors are installed in series to achieve the required compression ratio $r$, we can state that each compressor will operate at a compression ratio of

$$
\begin{equation*}
r=\left(r_{t}\right)^{\frac{1}{n}} \tag{4.25}
\end{equation*}
$$

where
$r=$ compression ratio, dimensionless
$r_{t}=$ overall compression ratio, dimensionless
$n=$ number of compressors in series
It has been found that by providing the overall compression ratio by means of identical compressors in series, power requirements will be minimized. Thus, in the preceding example, we assumed that two identical compressors in series, each providing a compression ratio of 2.74 resulting in an overall compression ratio of 7.5 , will be a better option than if we had a compressor with a compression ratio of 3.0 in series with another compressor with a compression ratio of 2.5 . To illustrate this further, if an overall compression ratio of 20 were required and we were to use three compressors in series, the most economical option would be to use identical compressors, each with a compression ratio of $(20)^{\frac{1}{3}}=2.71$.

## Example 12

A compressor station with multiple compressors in series is to provide a gas discharge pressure of 1500 psia . The gas inlet pressure and temperature are 100 psia and $80^{\circ} \mathrm{F}$, respectively. How many compressors in series will be required if the discharge temperature is limited to $250^{\circ} \mathrm{F}$ ? The ratio of specific heats $\gamma=1.4$.

Solution

The overall compression ratio is

$$
r=\frac{1500}{100}=15.0
$$

Since this is more than the maximum recommended compression ratio of 4 to 6 , we need two or more compressors in series. Initially, consider two compressors in series.

The compression ratio for each compressor is

$$
r=(15)^{\frac{1}{2}}=3.873
$$

This is acceptable, but the discharge temperature needs to be checked. From Equation 4.19, the discharge temperature for the first compressor is

$$
T_{2}=(80+460)(3.8)^{\frac{0.4}{1.4}}=790.76^{\circ} \mathrm{R}
$$

or $330.76^{\circ} \mathrm{F}$. This temperature is higher than the $250^{\circ} \mathrm{F}$ allowable. Therefore, we will need to consider three stages of compression. Using three compressors in series, the compression ratio is

$$
r=(15)^{\frac{1}{3}}=2.466
$$

Therefore, the discharge temperature for each compressor is

$$
T_{2}=(80+460)(2.466)^{\frac{0.4}{1.4}}=698.86^{\circ} \mathrm{R}
$$

or $239^{\circ} \mathrm{F}$. Since this is less than $250^{\circ} \mathrm{F}$ allowed, the three compressors in series are the choice. However, the gas must be cooled to the initial inlet temperature of $80^{\circ} \mathrm{F}$ between each compressor to limit discharge temperatures to $239^{\circ} \mathrm{F}$.

### 4.10 TYPES OF COMPRESSORS—CENTRIFUGAL AND POSITIVE DISPLACEMENT

Compressors used in natural gas transportation systems are either positive displacement (PD) type or centrifugal (CF) type. Positive displacement compressors generate the pressure required by trapping a certain volume of gas within the compressor and increasing the pressure by reduction of volume. The high-pressure gas is then released through the discharge valve into the pipeline. Piston-operated reciprocating compressors fall within the category of positive displacement compressors. These compressors
have a fixed volume and are able to produce high compression ratios. Centrifugal compressors, on the other hand, develop the pressure required by the centrifugal force due to rotation of the compressor wheel that translates the kinetic energy into pressure energy of the gas. Centrifugal compressors are more commonly used in gas transmission systems due to their flexibility. Centrifugal compressors have lower installed cost and lower maintenance expenses. They can handle larger volumes within a small area compared to positive displacement compressors. They also operate at high speeds and are of balanced construction. However, centrifugal compressors have less efficiency than positive displacement compressors.

Positive displacement compressors have flexibility in pressure range, have higher efficiency, and can deliver compressed gas at a wide range of pressures. They are also not very sensitive to the composition of the gas. Positive displacement compressors have pressure ranges up to $30,000 \mathrm{psi}$ and range from very low $H P$ to more than $20,000 H P$ per unit. Positive displacement compressors can be single stage or multistage, depending upon the compression ratio required. The compression ratio per stage for positive displacement compressors is limited to 4.0, because higher ratios cause higher discharge pressures, which affect the valve life of positive displacement compressors. Heat exchangers are used between stages of compression so that the compressed heated gas is cooled to the original suction temperature before being compressed in the next stage. The $H P$ required in a positive displacement compressor is usually estimated from charts provided by the compressor manufacturer. The following equation can be used for large slow-speed compressors with compression ratios greater than 2.5 and for gas specific gravity of 0.65 .

$$
\begin{equation*}
B H P=22 r N Q F \tag{4.26}
\end{equation*}
$$

where
$B H P=$ brake horsepower
$r \quad=$ compression ratio per stage
$N \quad=$ number of stages
$Q=$ gas flow rate, MMSCFD at suction temperature and 14.4 psia
$F \quad=$ factor that depends on the number of compression stages
$=1.0$ for single-stage compression
$=1.08$ for two-stage compression
$=1.10$ for three-stage compression

In Equation 4.26, the constant 22 is changed to 20 when gas gravity is between 0.8 and 1.0. Also, for compression ratios between 1.5 and 2.0 , the constant 22 is replaced with a number between 16 and 18 .

## Example 13

Calculate the $B H P$ required to compress 5 MMSCFD gas at 14.4 psia and $70^{\circ} \mathrm{F}$, with an overall compression ratio of 7 , considering two-stage compression.

Solution
Considering two identical stages, the compression ratio per stage $=\sqrt{7.0}=2.65$.
Using Equation 4.26, we get

$$
B H P=22 \times 2.65 \times 2 \times 5 \times 1.08=629.64
$$

Centrifugal compressors can be a single-wheel or single-stage compressor or multiwheel or multistage compressor. Single-stage centrifugal compressors have a volume range of 100 to $150,000 \mathrm{ft}^{3} / \mathrm{min}$ at actual conditions (ACFM). Multistage centrifugal compressors handle a volume range of 500 to 200,000 ACFM. The operational speeds of centrifugal compressors range from 3000 to $20,000 \mathrm{r} / \mathrm{min}$. The upper limit of speed will be limited by the wheel tip speed and stresses induced in the impeller. Advances in technology have produced compressor wheels operating at speeds in excess of $30,000 \mathrm{r} / \mathrm{min}$. Centrifugal compressors are driven by electric motors, steam turbines, or gas turbines. Sometimes speed increasers are used to increase the speeds necessary to generate the pressure.

### 4.11 COMPRESSOR PERFORMANCE CURVES

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.

The limiting curve on the left-hand side is known as the surge line, and the corresponding curve on the right side is known as the stone wall limit.

Generally, the performance of a centrifugal compressor follows the "affinity laws." According to the affinity laws, as the rotational speed of the centrifugal compressor is changed, the inlet flow and head vary as the speed and the square of the speed, respectively, as indicated in the following equations.

For compressor speed change,

$$
\begin{gather*}
\frac{Q_{2}}{Q_{1}}=\frac{N_{2}}{N_{1}}  \tag{4.27}\\
\frac{H_{2}}{H_{1}}=\left(\frac{N_{2}}{N_{1}}\right)^{2} \tag{4.28}
\end{gather*}
$$

where
$Q_{1}, Q_{2}=$ initial and final flow rates
$H_{1}, H_{2}=$ initial and final heads
$N_{1}, N_{2}=$ initial and final compressor speeds

In addition, the horsepower for compression varies as the cube of the speed change as follows:

$$
\begin{equation*}
\frac{H P_{2}}{H P_{1}}=\left(\frac{N_{2}}{N_{1}}\right)^{3} \tag{4.29}
\end{equation*}
$$

An example problem of using the affinity laws to predict the performance of a centrifugal compressor is illustrated next.

## Example 14

The compressor head and volume flow rate for a centrifugal compressor at 18,000 rpm are as follows:

| Flow Rate, $\boldsymbol{Q}$ | Head, $\boldsymbol{H}$ |
| :---: | :---: |
| ACFM | ft-lb/lb |
| 360 | 10,800 |
| 450 | 10,200 |
| 500 | 9700 |
| 600 | 8200 |
| 700 | 5700 |
| 730 | 4900 |

Using the affinity laws, determine the performance of this compressor at a speed of 20,000 rpm.

Solution

The ratio of speed is

$$
\frac{20000}{18000}=1.11
$$

The multiplier for the flow rate is 1.11 and the multiplier for the head is $(1.11)^{2}$ or 1.232 .
Using the affinity laws, the performance of the centrifugal compressor at $20,000 \mathrm{rpm}$ is as follows:

| Flow Rate, $\boldsymbol{Q}$ | Head, $\boldsymbol{H}$ |
| :---: | ---: |
| ACFM | $\mathrm{ft-ll/l}$ |
| 399.6 | 13,306 |
| 499.5 | 12,566 |
| 555.0 | 11,950 |
| 666.0 | 10,102 |
| 777.0 | 7,022 |
| 810.0 | 6,037 |

Next, we will explore how the head developed by a centrifugal compressor is calculated from the suction and discharge pressures, the compressibility factor, and the polytropic or adiabatic exponent. The calculation for the actual or inlet flow rate (ACFM) from the standard flow rate will also be illustrated. Finally, knowing the maximum head that can be generated per stage, the number of stages needed will be calculated.

Suppose a centrifugal compressor is used to raise the gas pressure from 800 psia to 1440 psia starting at a suction temperature of $70^{\circ} \mathrm{F}$ and gas flow rate of 80 MMSCFD . The average compressibility factor from the suction to the discharge side is 0.95 . The compressibility factor at the inlet is assumed to be 1.0 , and the polytropic exponent is 1.3 . Gas gravity is 0.6 . The head generated by the compressor is calculated as

$$
H=\frac{53.28}{0.6} \times 0.95 \times(70+460)\left(\frac{1.3}{0.3}\right)\left[\left(\frac{1440}{800}\right)^{\frac{0.3}{1.3}}-1\right]=28,146 \mathrm{ft}-\mathrm{lb} / \mathrm{lb}
$$

The actual flow rate at inlet conditions is calculated using the gas law as

$$
Q_{\text {act }}=\frac{80 \times 14.7 \times 1.0}{800} \times \frac{70+460}{60+460} \times \frac{10^{6}}{24 \times 60}=1040.5 \mathrm{ft}^{3} / \mathrm{min}(\mathrm{ACFM})
$$

If this particular compressor, according to vendor data, can produce a maximum head per stage of $10,000 \mathrm{ft}-\mathrm{lb} / \mathrm{lb}$, the number of stages required to produce the required head is

$$
n=\frac{28146}{10000}=3 \text {, approximately. }
$$

Next, suppose that this compressor has a maximum design speed of $16,000 \mathrm{rpm}$. The actual operating speed necessary for the three-stage compressor is, according to the affinity laws,

$$
N_{\mathrm{act}}=16,000 \sqrt{\frac{28,146}{3 \times 10,000}}=15,498
$$

Therefore, in order to generate $28,146 \mathrm{ft}-\mathrm{lb} / \mathrm{lb}$ of head at a gas flow rate of 1040.5 ACFM, this three-stage compressor must run at a speed of $15,498 \mathrm{rpm}$.

### 4.12 COMPRESSOR STATION PIPING LOSSES

As the gas enters the suction side of the compressor, it flows through a complex piping system within the compressor station. Similarly, the compressed gas leaving the compressor traverses the compressor station discharge piping system that consists of valves and fittings before entering the main pipeline on its way to the next compressor station or delivery terminus. This is illustrated in Figure 4.12.

It can be seen from Figure 4.12 that at the compressor station boundary A on the suction side, the gas pressure is $P_{1}$. This pressure drops to a value $P_{s}$ at the compressor suction, as the gas flows through the suction piping from A to B. This suction piping, consisting of valves, fittings, filters, and meters, causes a pressure drop of $\Delta P_{s}$ to occur. Therefore, the actual suction pressure at the compressor is

$$
\begin{equation*}
P_{s}=P_{1}-\Delta P_{s} \tag{4.30}
\end{equation*}
$$

where
$P_{s}=$ compressor suction pressure, psia
$P_{1}=$ compressor station suction pressure, psia
$\Delta P_{s}=$ pressure loss in compressor station suction piping, psi
At the compressor, the gas pressure is raised from $P_{s}$ to $P_{d}$ through a compression ratio $r$ as follows:

$$
\begin{equation*}
r=\frac{P_{d}}{P_{s}} \tag{4.31}
\end{equation*}
$$



Figure 4.12 Compressor station suction and discharge piping.
where
$r=$ compression ratio, dimensionless
$P_{d}=$ compressor discharge pressure, psia

The compressed gas then flows through the station discharge piping and loses pressure until it reaches the station discharge valve at the boundary D of the compressor station. If the station discharge pressure is $P_{2}$, we can write

$$
\begin{equation*}
P_{2}=P_{d}-\Delta P_{d} \tag{4.32}
\end{equation*}
$$

where
$P_{2}=$ compressor station discharge pressure, psia
$\Delta P_{d}=$ pressure loss in compressor station discharge piping, psi
Generally, the values of $\Delta P_{s}$ and $\Delta P_{d}$ range from 5 to 15 psi .

## Example 15

A compressor station on a gas transmission pipeline has the following pressures at the station boundaries. The station suction pressure $=850 \mathrm{psia}$, and the station discharge pressure $=1430$ psia. The pressure losses in the suction piping and discharge piping are 5 psi and 10 psi , respectively. Calculate the compression ratio of this compressor station.

Solution
From Equation 4.30, the compressor suction pressure is

$$
P_{s}=850-5=845 \mathrm{psia}
$$

Similarly, the compressor discharge pressure is

$$
P_{d}=1430+10=1440 \mathrm{psia}
$$

Therefore, the compression ratio is

$$
r=\frac{1440}{845}=1.70
$$

### 4.13 COMPRESSOR STATION SCHEMATIC

A typical compressor station schematic showing the arrangement of the valves, piping, and the compressor itself is shown in Figure 4.13.


Figure 4.13 Compressor station schematic.

### 4.14 SUMMARY

We discussed compressing a gas to generate the pressure needed to transport the gas from one point to another along a pipeline. An important parameter known as the compression ratio determines the horsepower required to compress a certain volume of gas and also influences the discharge temperature of the gas exiting the compressor. In a long-distance gas transmission pipeline, the method of locating intermediate compressor stations and minimizing energy lost was discussed. Hydraulically balanced and optimized compressor station locations were also discussed. Calculation of isothermal, adiabatic, and polytropic compression processes was explained and illustrated with sample problems. The $H P$ required for a given compression ratio and calculation of the gas discharge temperature were explained. The different types of compressors, such as positive displacement and centrifugal, were explained, along with their advantages and disadvantages. The need for configuring compressors in series and parallel was explored. The centrifugal compressor performance curve was discussed, and the effect of rotational speed on the flow rate and head using the affinity laws was illustrated with examples. Finally, the impact of the compressor station yard piping pressure drops and how they affect the compression ratio and horsepower were discussed.

## PROBLEMS

1. A natural gas pipeline 120 mi long from Dover to Leeds is constructed of NPS 14 and .250 in . wall thickness pipe, with an MOP of 1400 psig . The gas specific gravity and viscosity are 0.6 and $8 \times 10^{-6} \mathrm{lb} / \mathrm{ft}-\mathrm{s}$, respectively. The pipe roughness can be assumed to be $600 \mu \mathrm{in}$., and the base pressure and base temperature are 14.7 psia and $60^{\circ} \mathrm{F}$, respectively. The gas flow rate is 120 MMSCFD at $70^{\circ} \mathrm{F}$, and the delivery pressure required at Leeds is 700 psig. Determine the number and locations of compressor stations required, neglecting elevation difference along the pipeline. Assume $Z=0.90$.
2. Calculate the compressor horsepower required for an adiabatic compression of 80 MMSCFD gas with inlet temperature of $70^{\circ} \mathrm{F}$ and 800 psia pressure. The discharge pressure is 1400 psia. Assume the compressibility factors at suction and discharge conditions to be $Z_{1}=0.95$ and $Z_{2}=0.88$, respectively, and the adiabatic exponent $\gamma=1.3$, with the adiabatic efficiency $\eta_{a}=0.82$. If the mechanical efficiency of the compressor driver is 0.94 , what $B H P$ is required? Also, calculate the outlet temperature of the gas.
3. Natural gas at $4 \mathrm{Mm}^{3} /$ day and $24^{\circ} \mathrm{C}$ is compressed isentropically $(\gamma=1.3)$ from a suction pressure of 6.2 MPa to a discharge pressure of 9.4 MPa in a centrifugal compressor with an isentropic efficiency of 0.82 . Calculate the compressor power required, assuming the compressibility factors at suction and discharge conditions to be $Z_{1}=0.96$ and $Z_{2}=0.87$, respectively. If the mechanical efficiency of the compressor driver is 0.94 , what is the driver power required? Also, calculate the outlet temperature of the gas.
4. Determine the horsepower required to compress natural gas in a pipeline at a flow rate of 350 MMSCFD and at a compression ratio of 1.6, discharging at 1400 psig pressure. The suction temperature is $80^{\circ} \mathrm{F}$. The base temperature and base pressure are $60^{\circ} \mathrm{F}$ and 14.7 psia, respectively. The gas specific gravity is 0.65 , and the compression efficiency is 0.85 . What is the discharge temperature of the gas, assuming a polytropic compression exponent of 1.39 ? The compressibility factor $Z=1.0$ at suction conditions and $Z=0.86$ at discharge conditions.
5. Determine the horsepower required to compress natural gas in a pipeline at a flow rate of 500 MMSCFD and at a compression ratio of 1.4, discharging at 1200 psia pressure. The suction temperature is $70^{\circ} \mathrm{F}$. The base temperature and base pressure are $60^{\circ} \mathrm{F}$ and 14.7 psia , respectively. The gas specific gravity is 0.6 , and assume a compression efficiency of 0.9 . What is the discharge temperature of the gas, assuming the polytropic compression coefficient of 1.38 ? $Z=1.0$ at suction conditions and $Z=0.86$ at discharge conditions.
6. A gas transmission pipeline is 220 mi long, NPS 24, 0.500 in . wall thickness, and runs from Taylor to Jenks. There is an origin compressor station at Taylor and two intermediate compressor stations at Trent (milepost 70) and Beaver (milepost 130). There are no intermediate flow deliveries or injections, and the inlet flow rate of 500 MMSCFD at Taylor equals the delivery flow rate at Jenks. The delivery pressure required at Jenks is 700 psig , and the MOP of the pipeline is 1440 psig throughout. Neglect the effects of elevation, and assume a constant gas flow temperature of $70^{\circ} \mathrm{F}$ and constant values of transmission factor $F=20$ and
compressibility factor $Z=0.87$ throughout the pipeline. The gas gravity $=0.6$, base pressure $=14.7 \mathrm{psia}$, and base temperature $=60^{\circ} \mathrm{F}$. Use a polytropic compression coefficient of 1.4 and a compression efficiency of 0.85 . Determine the best locations for the intermediate compressor stations at Trent and Beaver. If the flow rate drops to 350 MMSCFD, will a single intermediate compressor station be sufficient at the reduced flow rate?
7. A compressor station with multiple compressors in series is to provide a gas discharge pressure of 1400 psia. The gas inlet pressure and temperature are 200 psia and $70^{\circ} \mathrm{F}$, respectively. How many compressors in series will be required if the discharge temperature is limited to $200^{\circ} \mathrm{F}$ ? The ratio of specific heats $\gamma=1.3$.
8. Calculate the $B H P$ required to compress 8 MMSCFD gas at 14.4 psia and $80^{\circ} \mathrm{F}$, with an overall compression ratio of 8 , considering two-stage compression.
9. The compressor head and volume flow rates for a centrifugal compressor at $15,000 \mathrm{rpm}$ are as follows:

| Q-ACFM | 720 | 900 | 1000 | 1200 | 1400 |
| :--- | ---: | ---: | ---: | ---: | ---: |
| H-ft-lb/lb | 5400 | 5100 | 4900 | 4100 | 2800 |

Using the affinity laws, determine the performance of this compressor at a speed of $12,000 \mathrm{rpm}$.
10. A compressor station on a gas transmission pipeline has the following pressures at the station boundaries: station suction pressure $=840 \mathrm{psig}$ and station discharge pressure $=1410$ psig. The pressure losses in the suction piping and discharge piping are 6 psi and 12 psi , respectively. Calculate the compression ratio of this compressor station.

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## CHAPTER 5

## Pipe Loops versus Compression

In this chapter we will explore the need for installing pipe loops in order to increase the throughput in a gas pipeline. Looping will be compared to another means of increasing pipeline capacity, such as installing compressor stations. The advantages and disadvantages of looping pipes vs. adding compressor stations will be discussed.

### 5.1 PURPOSE OF A PIPE LOOP

The purpose of a pipe loop that is installed in a segment of a pipeline is to essentially reduce the amount of pressure drop in that section of pipe. By doing so, the overall pressure drop in the pipeline will be reduced. This, in turn, will result in an increased pipeline flow rate at the same inlet pressure. Alternatively, if the flow rate is kept constant, reduction in total pressure required will cause a reduction in pumping horsepower. This is illustrated in Figure 5.1.

The pipe loop can be constructed of the same-diameter pipe as the main pipeline, or in some cases it can be of a different size. As we have seen in the analysis of parallel pipes in Chapter 3, the same diameter of pipe loop will result in equal volumes of gas flow in the main pipe as well as the loop. Thus, an NPS 20 pipe looped with an identical NPS 20 pipe segment will reduce the flow to one-half its original value in each pipe. If the loop is larger or smaller in diameter compared to the main pipeline, the volume distribution will not be equal. An NPS 20 pipe looped with an NPS 16 pipe will result in approximately $64 \%$ of the flow rate going through the larger-diameter pipe and $36 \%$ through the smaller-diameter pipe.


Figure 5.1 Effect of pipe loop.

### 5.2 PURPOSE OF COMPRESSION

We have seen in Chapter 3 and Chapter 4 that installing intermediate compressor stations along a pipeline will increase the flow rate and also reduce the operating pressure in a long gas transmission pipeline. The installation of the intermediate compressor station will result in additional operational and maintenance issues in comparison with pipe loops. Sometimes, additional compression is installed to increase flow rate in preference to looping the pipeline, since looping will involve additional permitting and right-of-way issues and could cost considerably more than adding the new compressor station. Installation of an intermediate compressor to increase flow rate is illustrated in Figure 5.2.


Figure 5.2 Adding a compressor station.

### 5.3 INCREASING PIPELINE CAPACITY

Consider an existing pipeline that is currently limited by the operating pressure that is close to the MAOP of the pipeline. Suppose the capacity of an NPS 16 pipeline is 100 MMSCFD and the discharge pressure at the originating compressor station is 1440 psig, as shown in Figure 5.2. It can be seen that, at the given flow rate and discharge pressure, the delivery pressure is 800 psig. If the pipeline flow rate is increased to 120 MMSCFD without changing the originating pressure of 1440 psig , the increased flow will cause greater pressure drop and, hence, the delivery pressure at the pipeline terminus will drop to some value such as 600 psig . The reduced delivery pressure may or may not be acceptable to the customer receiving the gas. However, we cannot increase the discharge pressure at the beginning of the pipeline to compensate for the drop in delivery pressure because the pressure is already at the MAOP level. How can we increase the flow rate and still provide the same delivery pressure as before? By installing an intermediate compressor station as shown in Figure 5.2, we can pump the increased volume approximately halfway and then boost the pressure at the new compressor station to the same MAOP level for ultimate delivery to the pipeline terminus at 800 psig , as before. This is illustrated in Figure 5.2.

Thus, we have been able to achieve the increased pipeline capacity of 120 MMSCFD by installing an additional compressor station at approximately the halfway point along the pipeline. Suppose we want to increase the flow rate further without changing the discharge pressure or the delivery pressure. It is clear that we could install additional intermediate compressor stations as needed to achieve the increased throughput, while maintaining the same delivery pressure. This is illustrated in Figure 5.3, where two additional compressor stations have been installed to increase the pipeline throughput while maintaining the desired delivery pressure at the pipeline terminus.


Figure 5.3 Multiple compressor stations.


Figure 5.4 Windsor to Cardiff pipeline.
However, there is a limit to the number of compressor stations that can be installed in a given pipeline system, since the $H P$ required continues to increase with flow rate and, hence, the capital cost and operating costs increase as well. At some point, the cost increases at a very high rate compared to the increase in flow rate. Each pipe size has a particular volume that can be economically transported based upon cost. An additional factor that must be taken into consideration as the flow rate is increased is the resulting higher velocity. As indicated in Chapter 2, the gas velocity must be well below the erosional velocity for the pipe.

## Example 1

A natural gas pipeline is 100 mi long and is constructed of NPS 16 and 0.250 in . wall thickness and runs from Windsor to Cardiff, as shown in Figure 5.4.

1. Neglecting elevation effects, calculate the maximum throughput capability of this pipeline, based upon an MAOP of 1440 psig and a delivery pressure of 800 psig at Cardiff. The suction pressure at Windsor is 800 psig.
2. Determine the requirement for two expansion scenarios. The phase 1 expansion will increase pipeline throughput by 50 MMSCFD and phase 2 will increase throughput by another 50 MMSCFD. In each case, calculate the number of compressor stations and $H P$ required. The gas flow velocities must be checked to ensure that they are within erosional limits.
3. Also estimate the approximate cost for each of these cases, using an overall installed cost of $\$ 2000$ per $H P$ for the compressor stations.
4. Compare these expansion cases using pipe loop instead of compression. Thus, for phase 1 , instead of building intermediate compressor stations, calculate the amount
of pipe loop needed to reduce the pressure drop at the higher flow rate. Similarly, for the phase 2 flow rate, calculate the looping necessary to maintain pressures without adding compressor stations. Estimate the cost of the expansion scenarios using pipe loops instead of compressor stations, based upon an overall installed cost of \$500,000 per mile of loop.

Assume a transmission factor of 20 , gas flow temperature of $80^{\circ} \mathrm{F}$, and compressibility factor of 0.85 throughout. Additional data are as follows: gas gravity $=0.6$, ratio of specific heats $=1.4$, base temperature $=60^{\circ} \mathrm{F}$, and base pressure $=14.7 \mathrm{psia}$. The compressor isentropic efficiency $=0.8$, and the mechanical efficiency of the compressor driver is 0.95 .

## Solution

1. First, determine the initial capacity, considering one compressor station at Windsor providing the pressure of 1440 psig needed for delivery pressure of 800 psig at Cardiff.

Using General Flow Equation 2.4, we calculate the initial capacity, $Q$, of the pipeline as follows:

$$
Q=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{1454.7^{2}-814.7^{2}}{0.6 \times 540 \times 100 \times 0.85}\right)^{0.5}(15.5)^{2.5}=188,410,280 \mathrm{SCFD}
$$

The $H P$ required is calculated from Equation 4.15:

$$
\begin{gathered}
H P=0.0857 \times 188.41\left(\frac{1.40}{0.40}\right)(540)\left(\frac{1+0.85}{2}\right)\left(\frac{1}{0.8}\right)\left[\left(\frac{1454.7}{814.7}\right)^{\frac{0.40}{1.40}}-1\right]=6357 \\
B H P \text { required }=\frac{6357}{0.95}=7064
\end{gathered}
$$

Checking gas velocities using Equation 2.26, the gas velocity at Windsor is

$$
u_{1}=0.002122\left(\frac{188.41 \times 10^{6}}{15.5^{2}}\right)\left(\frac{14.7}{520}\right)\left(\frac{540}{1454.7}\right)=17.46 \mathrm{ft} / \mathrm{s}
$$

The velocity at Cardiff is

$$
u_{1}=0.002122\left(\frac{188.41 \times 10^{6}}{15.5^{2}}\right)\left(\frac{14.7}{520}\right)\left(\frac{540}{814.7}\right)=31.18 \mathrm{ft} / \mathrm{s}
$$

The erosion velocity from Equation 2.31 is

$$
u_{\max }=100 \sqrt{\frac{0.85 \times 10.73 \times 540}{29 \times 0.6 \times 814.7}}=58.94 \mathrm{ft} / \mathrm{s}
$$

2. Next, we will calculate the compressor station requirement for the phase 1 flow rate of

$$
Q=188.41+50=238.41 \mathrm{MMSCFD}
$$

Assume that an additional compressor station is needed for this flow rate. This will be located at Avon at a distance of $L$ miles from Cardiff, such that a discharge pressure of 1440 psig at Avon will produce a delivery pressure of 800 psig at Cardiff. We will calculate the value of the pipe length, $L$, using General Flow Equation 2.4 as follows:

$$
238.41 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{1454.7^{2}-814.7^{2}}{0.6 \times 540 \times L \times 0.85}\right)^{0.5}(15.5)^{2.5}
$$

Solving for $L$, we get

$$
L=62.45 \mathrm{mi}
$$

Next, calculate the suction pressure at Avon using 1440 psig at Windsor and considering a pipe length of $37.55(100-62.45)$ mi between Windsor and Avon.

Using General Flow Equation 2.4, we get

$$
238.41 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{1454.7^{2}-P_{2}^{2}}{0.6 \times 540 \times 37.55 \times 0.85}\right)^{0.5}(15.5)^{2.5}
$$

Solving for the suction pressure at Avon, we get

$$
P_{2}=1114.85 \mathrm{psia}=1100.15 \mathrm{psig}
$$

Therefore, the compression ratio at Avon is

$$
r=\frac{1454.7}{1114.85}=1.30
$$

This is a satisfactory compression ratio for a centrifugal compressor.

The $H P$ required at Windsor and Avon for phase 1 will be calculated using Equation 4.15.

For Windsor, assuming the compressibility factor at suction is 1.0 ,

$$
H P=0.0857 \times 238.41\left(\frac{1.40}{0.40}\right)(540)\left(\frac{1+0.85}{2}\right)\left(\frac{1}{0.8}\right)\left[\left(\frac{1454.7}{814.7}\right)^{\frac{0.40}{1.40}}-1\right]=8044
$$

Therefore, the $B H P$ required at Windsor for phase $1=\frac{8044}{0.95}=8468$.
Similarly, the $H P$ required at Avon is

$$
H P=0.0857 \times 238.41\left(\frac{1.40}{0.40}\right)(540)\left(\frac{1+0.85}{2}\right)\left(\frac{1}{0.8}\right)\left[(1.30)^{\frac{0.40}{1.40}}-1\right]=3476
$$

Therefore, the $B H P$ required at Avon $=\frac{3476}{0.95}=3659$.
The total compressor $H P$ required at both compressor stations for the phase 1 flow rate of 238 MMSCFD is

$$
8468+3659=12,127 H P
$$

Therefore, the incremental $H P$ for phase 1 is

$$
\Delta H P=12,127-7064=5063 H P
$$

3. This represents the additional compressor $H P$ required for phase 1 for the extra 50 MMSCFD flow rate. The cost of this incremental $H P$, based on $\$ 2000$ per installed $H P$, is

$$
\Delta \text { Cost }=5063 \times 2000=\$ 10.13 \text { million }
$$

Next, check the gas velocity at the increased flow rate in phase 1 from Equation 2.26.
The velocity at Cardiff is

$$
u_{1}=0.002122\left(\frac{238.41 \times 10^{6}}{15.5^{2}}\right)\left(\frac{14.7}{520}\right)\left(\frac{540}{814.7}\right)=39.45 \mathrm{ft} / \mathrm{s}
$$

This velocity is acceptable, since it is less than the erosion velocity of $58.94 \mathrm{ft} / \mathrm{s}$ calculated earlier. The velocity at Windsor at the higher pressure of 1440 psig will be lower and, hence, less than the erosion velocity.

Next, consider the phase 2 flow rate of

$$
Q=238.41+50=288.41 \text { MMSCFD }
$$

Since phase 2 occurs after phase 1 , where the Avon compressor station is already built, we might have to install one compressor station between Windsor and Avon and another between Avon and Cardiff. If we consider this phase independent of phase 1 , we could probably install two additional compressor stations between Windsor and Cardiff to handle the phase 2 flow of 288.41 MMSCFD. For now, we will consider a compressor station at Jenks between Windsor and Avon and another one at Hart located between Avon and Cardiff. We will calculate the distance of $L$ miles from Hart to Cardiff, such that a discharge pressure of 1440 psig at Hart will produce a delivery pressure of 800 psig at Cardiff. The value of $L$ is calculated, using General Flow Equation 2.4, as we did before for locating the Avon compressor station, as follows:

$$
288.41 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{1454.7^{2}-814.7^{2}}{0.6 \times 540 \times L \times 0.85}\right)^{0.5}(15.5)^{2.5}
$$

Solving for $L$, we get

$$
L=42.67 \mathrm{mi}
$$

Therefore, the Hart compressor station will be located at a distance of 19.78 (62.45 42.67) mi from Avon. The suction pressure at Hart is calculated next.

Using General Flow Equation 2.4, we get

$$
288.41 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{1454.7^{2}-P_{2}^{2}}{0.6 \times 540 \times 19.78 \times 0.85}\right)^{0.5}(15.5)^{2.5}
$$

Solving for the suction pressure at Hart, we get

$$
P_{2}=1201.24 \mathrm{psia}=1186.54 \mathrm{psig}
$$

Therefore, the compression ratio at Hart is

$$
r=\frac{1454.7}{1201.24}=1.21
$$

This is a satisfactory compression ratio.
Before determining the location of the Jenks compressor station between Windsor and Avon, calculate the suction pressure at Avon, assuming Jenks doesn't exist and that the Windsor compressor station pumps directly into Avon, as in phase 1.

The suction pressure at Avon, considering 1440 psig at Windsor, is calculated using General Flow Equation 2.4:

$$
288.41 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{1454.7^{2}-P_{2}^{2}}{0.6 \times 540 \times 37.55 \times 0.85}\right)^{0.5}(15.5)^{2.5}
$$

Solving for the suction pressure at Avon, we get

$$
P_{2}=915.54 \mathrm{psia}=900.84 \mathrm{psig}
$$

Therefore, the compression ratio at Avon is

$$
r=\frac{1454.7}{915.54}=1.59
$$

This is a satisfactory compression ratio.
Therefore, for phase 2, we will need only two compressor stations besides Windsor, Avon at milepost 37.55 and Hart at milepost 57.33.

Next, calculate the total $H P$ required at Windsor, Avon, and Hart at phase 2 flow rates.

The $H P$ required at Windsor is, using Equation 4.15,

$$
H P=0.0857 \times 288.41\left(\frac{1.40}{0.40}\right)(540)\left(\frac{1+0.85}{2}\right)\left(\frac{1}{0.8}\right)\left[\left(\frac{1454.7}{814.7}\right)^{\frac{0.40}{1.40}}-1\right]=9731
$$

Therefore, the $B H P$ required at Windsor for phase $2=\frac{9731}{0.95}=10,243$.

Similarly, the $H P$ required at Avon is

$$
H P=0.0857 \times 288.41\left(\frac{1.40}{0.40}\right)(540)\left(\frac{1+0.85}{2}\right)\left(\frac{1}{0.8}\right)\left[(1.59)^{\frac{0.40}{1.40}}-1\right]=7652
$$

Therefore, the $B H P$ required at Avon $=\frac{7652}{0.95}=8055$.
The $H P$ required at Hart is

$$
H P=0.0857 \times 288.41\left(\frac{1.40}{0.40}\right)(540)\left(\frac{1+0.85}{2}\right)\left(\frac{1}{0.8}\right)\left[(1.21)^{\frac{0.40}{1.40}}-1\right]=3023
$$

Therefore, the $B H P$ required at Hart $=\frac{3023}{0.95}=3182$.
The total compressor $H P$ required at all three compressor stations for phase 2 is

$$
10,243+8055+3182=21,480 H P
$$

The incremental $H P$ for phase 2 compared to phase 1 is

$$
\Delta H P=21,480-12,127=9353 H P
$$

This represents the additional compression $H P$ required for phase 2 compared to phase 1 , for the additional 50 MMSCFD flow rate. The cost of this incremental $H P$, based on $\$ 2000$ per installed $H P$, is

$$
\Delta \text { Cost }=9353 \times 2000=\$ 18.71 \text { million }
$$

Next, check the velocity at increased flow rates in phase 2 from Equation 2.26.
The velocity at Cardiff is

$$
u_{1}=0.002122\left(\frac{288.41 \times 10^{6}}{15.5^{2}}\right)\left(\frac{14.7}{520}\right)\left(\frac{540}{814.7}\right)=47.72 \mathrm{ft} / \mathrm{s}
$$

This velocity is acceptable, since it is less than the erosion velocity. The velocity at higher pressures will be well within the limits.
4. In the preceding analysis, we accomplished the increase in flow rates for phase 1 and phase 2 by adding intermediate compressor stations. The capital cost for phase 1 expansion was $\$ 10.13$ million and for the phase 2 expansion was an additional $\$ 18.71$ million.

Next, we will explore the two expansions by installing pipe loops without additional intermediate compressor stations.

For phase 1, assume that $L$ miles of the pipe near Cardiff will be looped. The reason we picked this section is because in Chapter 3, we found that looping close to the
downstream end is more beneficial than looping near the upstream end, as long as the flowing temperature was constant. Following the methodology of Chapter 3, we will determine the equivalent diameter of the pipe loop as follows:

Assuming the loop to be of the same diameter as the main piping and $L_{1}=L_{2}$, using Equation 3.18, we get

$$
\text { Const } 1=1.0
$$

Therefore, the equivalent diameter, using Equation 3.17, is

$$
D_{e}=D_{1}\left[\left(\frac{1+1}{1}\right)^{2}\right]^{1 / 5}=1.32 D_{1}=1.32 \times 15.5=20.46 \mathrm{in} .
$$

Considering $L$ miles of pipe of inside diameter 20.46 in., calculate the upstream pressure at the beginning of the loop as shown in Figure 5.5. The downstream pressure at Cardiff is 800 psig , and the upstream pressure P is unknown. Using General Flow Equation 2.4,

$$
\begin{equation*}
238.41 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{P^{2}-814.7^{2}}{0.6 \times 540 \times L \times 0.85}\right)^{0.5}(20.46)^{2.5} \tag{5.1}
\end{equation*}
$$

There are two unknowns, $P$ and $L$, in Equation 5.1. We need another equation to solve for both variables. For this, the pipe segment from Windsor to the start of the loop will be examined.

Considering 1440 psig at Windsor, calculate the downstream pressure $P$ at the beginning of the loop for a pipe length of $(100-L)$. Using General Flow Equation 2.4,

$$
\begin{equation*}
238.41 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{1454.7^{2}-P^{2}}{0.6 \times 540 \times(100-L) \times 0.85}\right)^{0.5}(15.5)^{2.5} \tag{5.2}
\end{equation*}
$$



Figure 5.5 Pipe loop for phase 1.

Eliminating $P$ from Equation 5.1 and Equation 5.2, we solve for $L$ as follows:

$$
L=50.03 \mathrm{mi}
$$

Substituting this value of $L$ in Equation 5.1 and solving for $P$,

$$
P=976.76 \mathrm{psia}=962.06 \mathrm{psig}
$$

Therefore, for phase 1, without an intermediate compressor station, flow increase can be achieved by looping 50.03 mi of pipe upstream of Cardiff.

The installed cost of this pipe loop is

$$
50.03 \times \$ 500,000=\$ 25.02 \text { million }
$$

In addition to this cost of pipe loop, we must also include the increased horsepower requirement at Windsor for the phase 1 flow rate. Since the discharge pressure at Windsor is still 1440 psig as before, the $H P$ is the same as that calculated earlier. The incremental HP is $(8468-7064)=1404 H P$.

At $\$ 2000$ per installed $H P$, the extra cost for incremental $H P$ is

$$
2000 \times 1404=\$ 2.81 \text { million }
$$

Thus, for phase 1, the cost of looping pipe upstream of Cardiff and increased HP cost at Windsor compressor station is

$$
\$ 25.02+\$ 2.81=\$ 27.83 \text { million }
$$

This compares with $\$ 10.13$ million calculated earlier for phase 1 using a compressor station at Avon. Even though at first sight the looping appears to be a more expensive option, we must also consider the increased operating cost when adding a compressor station. The annual operating cost for the compressor station can be estimated considering the fuel consumption, operating and maintenance costs, and other costs. In Chapter 10, we will discuss more details of capital cost, operating cost, and cost of service. For now, we will only look at capital costs.

For phase 2, at a flow rate of 288.41 MMSCFD , similarly calculate the amount of pipe loop needed, without adding any intermediate compression.

The length of loop $L$ required is calculated as follows:
Using General Flow Equation 2.4,

$$
\begin{equation*}
288.41 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{P^{2}-814.7^{2}}{0.6 \times 540 \times L \times 0.85}\right)^{0.5}(20.46)^{2.5} \tag{5.3}
\end{equation*}
$$

Considering 1440 psig at Windsor, calculate the downstream pressure $P$ at the beginning of the loop for a pipe length of $(100-L)$.

Using General Flow Equation 2.4,

$$
\begin{equation*}
288.41 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{1454.7^{2}-P^{2}}{0.6 \times 540 \times(100-L) \times 0.85}\right)^{0.5}(15.5)^{2.5} \tag{5.4}
\end{equation*}
$$

Eliminating $P$ from Equation 5.3 and Equation 5.4, we solve for the loop length $L$ as

$$
L=76.26 \mathrm{mi}
$$



NPS 16 pipeline 100 mi long

Figure 5.6 Pipe loop for phase 2.

This is shown in Figure 5.6.
Substituting this value of $L$ in Equation 5.4 and solving for $P$,

$$
P=1144.54 \mathrm{psia}=1129.84 \mathrm{psig}
$$

Therefore, the installed cost of this pipe loop is

$$
76.26 \times \$ 500,000=\$ 38.13 \text { million }
$$

In addition to the cost of pipe loop, we must include the increased horsepower requirement at Windsor for phase 2. Since the discharge pressure at Windsor is still 1440 psig as before, the $H P$ is the same as that calculated earlier. The incremental $H P$ is $(10,243-8468)=1775 H P$ more than that required for phase 1 . At $\$ 2000$ per installed $H P$, the incremental cost is

$$
1775 \times \$ 2000=\$ 3.55 \text { million }
$$

compared to phase 1.

Thus, for phase 2 the total incremental cost of additional looping over phase 1 and increased HP at the Windsor compressor station is

$$
(\$ 38.13-\$ 25.02)+\$ 3.55=\$ 16.66 \text { million }
$$

The costs of the initial case and the two expansion scenarios for the compressor station option and the pipe loop option are summarized in Table 5.1 and Table 5.2.

Table 5.1 Windsor to Cardiff Pipeline Expansion-Compressor Station Option

| Phase | $\begin{array}{c}\text { Flow, } \\ \text { MMSCFD }\end{array}$ | Compressor BHP Required |  |  | $\begin{array}{c}\text { Compression } \\ \text { Comdsor }\end{array}$ | Avon | Hart |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | \(\left.\begin{array}{c}Incremental <br>

Cost, \$ million\end{array}\right)\)

Table 5.2 Windsor to Cardiff Pipeline Expansion-Pipe Loop Option

| Phase | Flow, MMSCFD | Compressor BHP Windsor | Pipe Loop, mi | Compression Cost, \$ million | Pipe <br> Loop <br> Cost, <br> \$ million | Total Cost, \$ million | Incrementa Cost, \$ million |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Initial | 188.41 | 7,064 | - | 14.13 | - | 14.13 | - |
| 1 | 238.41 | 8,468 | 50.03 | 16.94 | 25.02 | 41.96 | 27.83 |
| 2 | 288.41 | 10,243 | 76.26 | 20.49 | 38.13 | 58.62 | 16.66 |

### 5.4 REDUCING POWER REQUIREMENTS

In an existing pipeline for a given flow rate, we can calculate the $H P$ required based upon the number of compressor stations, their suction and discharge pressures, and flow rate. Suppose we are interested in reducing the $H P$ required and, hence, the annual operating cost of the pipeline. If the flow rate is not reduced, the only way power consumption can be reduced is to reduce the overall pressure drop between compressor stations. If the pipeline is 100 miles long and at a flow rate of 100 MMSCFD, an origin compressor station and an intermediate compressor station are required, each station operating at 900 psia suction and 1400 psia discharge pressures. The HP required will depend upon the compression ratio of $(1400 / 900)$ or 1.56 . Since the flow rate is constant, $H P$ can be reduced by increasing the suction pressure or decreasing the discharge pressure, both of which reduce the compression ratio. Since the objective is to operate a gas pipeline at the highest possible pressure for efficiency, we will not reduce the discharge pressure. That leaves us the option of only increasing the suction pressure. Suction pressure can be increased by reducing the pressure drop in the pipeline segment upstream of the compressor station. Since the flow rate and pipe diameter are fixed, the pressure drop in a pipe segment can be decreased by installing a pipe loop. Therefore, looping a segment of pipeline, thereby reducing the pressure drop, will result in a decrease in $H P$ and annual operating cost. We will illustrate this using an example.

## Example 2

A natural gas (specific gravity $=0.60$ ) pipeline is 130 mi long and is constructed of NPS 20, 0.500 in . wall thickness pipeline ( $\mathrm{MAOP}=1440 \mathrm{psig}$ ) that runs from Anaheim to Ventura. At a flow rate of 300 MMSCFD, an intermediate compressor at Brentwood (milepost 70) is needed. Calculate the total HP required for both compressor stations. In order to reduce the power consumption by $30 \%$ at the present flow rate, it is proposed to loop the pipeline. Calculate the extent of looping required. For simplicity, use the General Flow equation with a transmission factor $F=20$ and compressibility factor of 0.90 . The gas flow temperature is $60^{\circ} \mathrm{F}$ and base pressure is 14.7 psia . The base temperature is $60^{\circ} \mathrm{F}$. The delivery pressure required at Ventura is 800 psig . The discharge pressure at Anaheim is 1440 psig , and the suction pressure is 900 psig . Use $80 \%$ isentropic efficiency and $95 \%$ mechanical efficiency for compressors. The gas specific heat ratio $\gamma=1.4$.

Solution

Using General Flow Equation 2.4, calculate the discharge pressure required at the Brentwood compressor station for 800 psig delivery pressure at Ventura.

$$
300 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-814.7^{2}}{0.6 \times 520 \times 60 \times 0.9}\right)^{0.5}(19.0)^{2.5}
$$

Solving for the discharge pressure at Brentwood,

$$
P_{1}=1216 \mathrm{psia}
$$

Next, calculate the suction pressure at Brentwood, applying the General Flow equation to the pipeline segment 70 mi long between Anaheim and Brentwood.

$$
300 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{1454.7^{2}-P_{2}^{2}}{0.6 \times 520 \times 70 \times 0.9}\right)^{0.5}(19.0)^{2.5}
$$

Solving for the suction pressure at Brentwood,

$$
P_{2}=1079.25 \mathrm{psia}
$$

The compression ratio at Anaheim is

$$
r=\frac{1454.7}{914.7}=1.59
$$

The $H P$ required at Anaheim is calculated using Equation 4.15:

$$
H P=0.0857 \times 300\left(\frac{1.40}{0.40}\right)(520)\left(\frac{1+0.9}{2}\right)\left(\frac{1}{0.8}\right)\left[\left(\frac{1454.7}{914.7}\right)^{\frac{0.40}{1.40}}-1\right]=7876
$$

where the compressibility factor for suction conditions is assumed to be 1.0 .
Considering a mechanical efficiency of $95 \%$, the BHP required at Anaheim is

$$
B H P=\frac{7876}{0.95}=8291
$$

Similarly, calculate the compression ratio and BHP for the Brentwood compressor station.

The compression ratio at Brentwood is

$$
r=\frac{1216}{1079.25}=1.127
$$

The $H P$ required at Brentwood is

$$
H P=0.0857 \times 300\left(\frac{1.40}{0.40}\right)(520)\left(\frac{1.9}{2}\right)\left(\frac{1}{0.8}\right)\left[(1.127)^{\frac{0.40}{1.40}}-1\right]=1931
$$

and the $B H P$ is

$$
B H P=\frac{1931}{0.95}=2033
$$

By looping the pipe segment between Anaheim and Brentwood using NPS 20 pipe, the flow rate through each pipe will be one-half the inlet flow of 300 MMSCFD at Anaheim.

The suction pressure at Brentwood is calculated using the General Flow equation as

$$
150 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{1454.7^{2}-P_{2}^{2}}{0.6 \times 520 \times 70 \times 0.9}\right)^{0.5}(19.0)^{2.5}
$$

Solving for $P_{2}$, we get

$$
P_{2}=1371 \mathrm{psia}
$$

Since this pressure is more than the discharge pressure of 1216 psia calculated earlier for Brentwood, we conclude that the Brentwood station will not be needed if we loop the entire 70 mi pipe segment from Anaheim to Brentwood. This would reduce the total $B H P$ required to 8291 from $(8291+2033)$ calculated earlier for the Anaheim and Brentwood compressor stations.

The reduction in $B H P$ is

$$
\Delta B H P=\frac{2033}{8291+2033}=0.197, \text { or } 19.7 \%
$$

Since the objective is to reduce the power consumption by $30 \%$, we must do more than just loop the pipe segment between Anaheim and Brentwood. We will recalculate the discharge pressure at Anaheim without the Brentwood compressor station, such that the delivery pressure at Ventura is 800 psig . The reduced discharge pressure at Anaheim due to the 70 mi pipe loop will reduce the compression ratio and, hence, the $H P$ at Anaheim.

Using the General Flow equation for the pipe segment between Anaheim and Brentwood, the downstream pressure at Brentwood must equal the 1216 psia calculated earlier to ensure 800 psig delivery at Ventura.

Therefore, considering half the total flow rate through each NPS 20 pipe section of the loop,

$$
150 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-1216^{2}}{0.6 \times 520 \times 70 \times 0.9}\right)^{0.5}(19.0)^{2.5}
$$

Solving for the discharge pressure at Anaheim, we get

$$
P_{1}=1310 \mathrm{psia}
$$

This reduced discharge pressure at Anaheim causes the compression ratio to be

$$
r=\frac{1310}{914.7}=1.432
$$

The revised $H P$ at Anaheim is calculated as

$$
\begin{gathered}
H P=0.0857 \times 300\left(\frac{1.40}{0.40}\right)(520)\left(\frac{1.9}{2}\right)\left(\frac{1}{0.8}\right)\left[(1.432)^{\frac{0.40}{1.40}}-1\right]=6003 \\
B H P=\frac{6003}{0.95}=6319
\end{gathered}
$$

Therefore, the total reduction in $H P$ is

$$
(8291+2033)-6319=4005
$$

Or the percentage reduction in $H P$ is

$$
\frac{4005}{8291+2033}=0.39 \text { or } 39 \%
$$

This is well above the $30 \%$ reduction in power required. If we reduce the loop pipe length slightly from 70 mi , we will realize the required $30 \%$ reduction. This is left as an exercise for the reader.

### 5.5 LOOPING IN DISTRIBUTION PIPING

Another example of pipe loops is as follows. Consider a distribution piping system as shown in Figure 5.7. Gas enters the pipeline at A at a flow rate of 60 MMSCFD , and after making gas deliveries at B of 20 MMSCFD and at C of 30 MMSCFD , the remaining 10 MMSCFD of gas proceeds to D , where an additional 10 MMSCFD enters the pipeline, which is delivered to the terminus at E . The last segment of pipe has a flow rate of ( $60-20-30+10$ ) or 20 MMSCFD . Suppose it is desired to bring in an extra 10 MMSCFD gas at D so that the delivery at E is increased to 30 MMSCFD. If the delivery pressure at E is to remain the same at 600 psig , it is clear that the pressure at D will need to be increased to handle the extra flow rate in pipe segment DE. This, in turn, will raise all pressures upstream of D . Thus, the pressures


Figure 5.7 Distribution piping.
at $\mathrm{A}, \mathrm{B}$, and C will all increase, resulting in an increased $H P$ requirement at A . However, by looping the section DE, we can maintain all pressures the same as before.

Assume that for the initial case, where the injection at D is 10 MMSCFD, the pressure at D is 900 psig. The delivery pressure at E is to be maintained constant at 600 psig . If the entire length of pipe DE is looped with an identical pipe size, the equivalent diameter $D_{e q}$ is such that at 30 MMSCFD , the pressure drop in the diameter $D_{e q}$ is the same as the pressure drop in the original pipe diameter $D$ at 20 MMSCFD.

From General Flow Equation 2.4, the flow rate is directly proportional to the square root of $\left(P_{1}{ }^{2}-P_{2}{ }^{2}\right)$ and also to the pipe diameter raised to the power of 2.5, keeping everything else the same. $P_{1}$ and $P_{2}$ are the upstream and downstream pressures in a pipe segment.

Since we want the upstream and downstream pressures for the pipe segment DE to be the same at both flow rates, at 20 MMSCFD,

$$
\begin{equation*}
20=C\left(P_{1}^{2}-P_{2}^{2}\right)^{0.5} D^{2.5} \tag{A}
\end{equation*}
$$

and at 30 MMSCFD,

$$
\begin{equation*}
30=C\left(P_{1}^{2}-P_{2}^{2}\right)^{0.5} D_{e q}^{2.5} \tag{B}
\end{equation*}
$$

where $C$ is a constant for the pipe segment.
By dividing one equation by the other, we get

$$
\left(\frac{D_{e q}}{D}\right)^{2.5}=\frac{30}{20}=1.5
$$

The equivalent diameter is

$$
D_{e q}=D(1.5)^{\frac{1}{2.5}}
$$

If the initial pipe size of DE was 12.00 in . inside diameter, we need an equivalent diameter of

$$
D_{e q}=12 \times(1.5)^{\frac{1}{2.5}}=14.11 \mathrm{in}
$$

Next, we need to determine the loop diameter required that will produce the equivalent diameter just calculated. Since the pressure drop in each pipe loop is the same, if $Q_{1}$ and $Q_{2}$ represent the flow rates in the main pipe and loop respectively,

$$
\begin{equation*}
Q_{1}+Q_{2}=30 \tag{5.5}
\end{equation*}
$$

and

$$
\begin{equation*}
\frac{30}{D_{e q}^{2.5}}=\frac{Q_{1}}{(12)^{2.5}}=\frac{Q_{2}}{(D)^{2.5}} \tag{5.6}
\end{equation*}
$$

where $D$ is the loop diameter to be calculated and the main pipe is 12.00 in . diameter.
Solving for $Q_{1}$, we get

$$
Q_{1}=30\left(\frac{12}{14.11}\right)^{2.5}=20.01 \mathrm{MMSCFD}
$$

and the flow rate through the loop is

$$
Q_{2}=30-20.01=9.99 \mathrm{MMSCFD}
$$

Therefore, from Equation 5.6,

$$
\frac{9.99}{D^{2.5}}=\frac{30}{(14.11)^{2.5}}
$$

Solving for $D$, we get

$$
D=\left(\frac{9.99}{30}\right)^{\frac{1}{2.5}} \times 14.11=9.09 \mathrm{in} .
$$

Therefore, by looping the entire length DE of the existing 12 in . diameter pipe with a pipe having an inside diameter of 9.09 in., we will maintain the same pressure at all points as before.

A slightly different case of looping is one in which the inlet flow at A needs to be increased so that the increased volume can be delivered at B , while keeping all pressures the same as before. Suppose the delivery volume at B needs to be increased to 30 MMSCFD , without changing other deliveries or receipt. The inlet volume at A will increase from 60 MMSCFD to 70 MMSCFD , and the delivery volume at B will increase from 20 MMSCFD to 30 MMSCFD . We will loop the section AB such that the pressure at A and B remain the same as before, so that the volumes and pressures at all points downstream of $B$ remain the same. This will be illustrated by calculating the pressures and the size of the pipe loop required in the next example.

## Example 3

In a gas distribution pipeline, 60 MMSCFD enters the pipeline at $A$, as shown in Figure 5.8. If the delivery at B is increased from 20 MMSCFD to 30 MMSCFD by increasing the inlet flow at A, keeping all downstream flow rates the same, calculate the looping necessary for section AB to ensure pressures are not changed throughout the pipeline. Pipe AB is NPS 14, 0.250 in . wall thickness; BC is NPS 12, 0.250 in .


Figure 5.8 Looping a distribution piping.
wall thickness; CD is NPS 10, 0.250 in. wall thickness; and DE is NPS 12, 0.250 in . wall thickness. The delivery pressure at E is fixed at 600 psig . The pipe lengths are as follow:

$$
\begin{aligned}
& \mathrm{AB}=12 \mathrm{mi} \\
& \mathrm{BC}=18 \mathrm{mi} \\
& \mathrm{CD}=20 \mathrm{mi} \\
& \mathrm{DE}=8 \mathrm{mi}
\end{aligned}
$$

The gas gravity is 0.60 , and the flow temperature is $60^{\circ} \mathrm{F}$. The compressibility factor and transmission factor can be assumed to be 0.85 and 20 , respectively, throughout the pipeline. The base pressure and base temperature are 14.7 psia and $60^{\circ} \mathrm{F}$, respectively.

## Solution

First, the pressures at A and B must be calculated for the initial flow rates. Starting at E , for a delivery pressure of 600 psig at E , the pressures at $\mathrm{D}, \mathrm{C}$, and B will be calculated sequentially. Applying the General Flow equation for the 8 mi section DE of inside diameter 12.25 in . and at a flow rate of 20 MMSCFD ,

$$
20 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{P_{D}^{2}-614.7^{2}}{0.6 \times 520 \times 8 \times 0.85}\right)^{0.5}(12.25)^{2.5}
$$

Solving for $P_{D}$, we get

$$
P_{D}=618.02 \mathrm{psia}=603 \mathrm{psig}
$$

Next, calculate the pressure at C , considering 10 MMSCFD flow through the 20 mi section of pipe CD, of inside diameter 10.25 in .:

$$
10 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{P_{C}^{2}-618.02^{2}}{0.6 \times 520 \times 20 \times 0.85}\right)^{0.5}(10.25)^{2.5}
$$

Solving for $P_{C}$, we get

$$
P_{C}=623.04 \mathrm{psia}=608.34 \mathrm{psig}
$$

Similarly, the pressure at B is calculated considering 40 MMSCFD flow through the 18 mi section of pipe BC, with an inside diameter 12.25 in .:

$$
40 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{P_{B}^{2}-623.04^{2}}{0.6 \times 520 \times 18 \times 0.85}\right)^{0.5}(12.25)^{2.5}
$$

Solving for $P_{B}$, we get

$$
P_{B}=651.90 \mathrm{psia}=637.20 \mathrm{psig}
$$

Next, calculate the pressure at A considering 60 MMSCFD flow through the 12 mi section of pipe AB , with an inside diameter 13.5 in.:

$$
60 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{P_{A}^{2}-651.90^{2}}{0.6 \times 520 \times 12 \times 0.85}\right)^{0.5}(13.5)^{2.5}
$$

Solving for $P_{A}$, we get

$$
P_{A}=677.45 \mathrm{psia}=662.75 \mathrm{psig}
$$

Therefore, the pipe section AB , when flowing 60 MMSCFD of gas, has the following pressures:

$$
\begin{aligned}
& P_{A}=677.45 \mathrm{psia} \\
& P_{B}=651.90 \mathrm{psia}
\end{aligned}
$$

When the delivery rate at B is increased from 20 MMSCFD to 30 MMSCFD, the flow rate in pipe segment $A B$ increases from 60 MMSCFD to 70 MMSCFD. Since the pressures at $A$ and $B$ are to remain the same as before, the pipe segment $A B$ must be looped to reduce the pressure drop at the higher flow rate. We will assume the entire 12 mi length will be looped. Next, we calculate the equivalent diameter required for segment AB , using the General Flow equation, so the pressures at A and B are the same as before.

$$
70 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{677.45^{2}-651.90^{2}}{0.6 \times 520 \times 12 \times 0.85}\right)^{0.5}(D)^{2.5}
$$

Solving for the diameter,

$$
D=14.36 \mathrm{in} .
$$

The equivalent diameter of the looped line AB must be 14.36 in. to keep pressures the same as calculated. From Equation 3.17 and Equation 3.18, the diameter of the loop can be calculated, knowing the equivalent diameter just calculated.

From Equation 3.17,

$$
14.36=13.50\left[\left(\frac{1+\text { Const1 }}{\text { Const1 }}\right)^{2}\right]^{\frac{1}{5}}
$$

Solving for Const 1

$$
\text { Const } 1=5.99
$$

From Equation 3.18, and since $L_{1}=L_{2}=12 \mathrm{mi}$,

$$
5.99=\sqrt{\left(\frac{13.5}{D_{2}}\right)^{5}}
$$

Solving for the pipe loop diameter $D_{2}$,

$$
D_{2}=6.6 \mathrm{in} .
$$

Therefore, the pipe section AB must be looped with a pipe of inside diameter 6.6 in. for the entire length of 12 mi . We could also increase the loop diameter and reduce the pipe length that is looped to get the same effect. For example, increasing the loop diameter to 10 in . will reduce the length of looping needed. Suppose we decide on an NPS $10,0.250 \mathrm{in}$. wall thickness pipe for the loop length of $L$ mi. upstream of B.

The equivalent diameter will be calculated using Equation 3.17 and Equation 3.18.
From Equation 3.18,

$$
\text { Const } 1=\sqrt{\left(\frac{13.5}{10.25}\right)^{5}}=1.9908
$$

and from Equation 3.17, the equivalent diameter is

$$
D_{e}=13.50\left[\left(\frac{1+1.9908}{1.9908}\right)^{2}\right]^{\frac{1}{5}}=15.89 \mathrm{in}
$$

The pressure at the start of the loop will be calculated from General Flow Equation 2.4:

$$
70 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{P^{2}-651.90^{2}}{0.6 \times 520 \times L \times 0.85}\right)^{0.5}(15.89)^{2.5}
$$

Simplifying,

$$
\begin{equation*}
P^{2}-651.90^{2}=1705 L \tag{5.7}
\end{equation*}
$$

Next, consider the unlooped portion of pipe AB from A to the starting point of the loop.
Using General Flow Equation 2.4, we get

$$
70 \times 10^{6}=38.77 \times 20\left(\frac{520}{14.7}\right)\left(\frac{677.45^{2}-P^{2}}{0.6 \times 520 \times(12-L) \times 0.85}\right)^{0.5}(13.5)^{2.5}
$$

Simplifying,

$$
\begin{equation*}
677.45^{2}-P^{2}=3851.92(12-L) \tag{5.8}
\end{equation*}
$$

Eliminating $P$ from Equation 5.7 and Equation 5.8 and solving for $L$, we get

$$
L=5.71 \mathrm{mi}
$$

Therefore, by looping the existing NPS 14 pipe from A to B with an identical NPS 14 pipe, 5.71 mi long (measured upstream from B), the pressures will be the same as before the increased delivery volume at B.

### 5.6 SUMMARY

We discussed two ways to increase the throughput of a gas pipeline: using intermediate compressor stations and installing pipe loops. With intermediate compressor stations, the flow rate can be increased to fully utilize pipe MAOP. However, adding compressor stations causes increased capital cost as well as annual operating and maintenance costs. On the other hand, by installing a pipe loop, the effective diameter of the pipe is increased, resulting in a lower pressure drop. Therefore, additional flow rate can be realized without installing an intermediate compressor station. Looping an existing pipeline causes increase in capital but very little increase in operating and maintenance costs compared to installing intermediate compressor stations. We also discussed how the $H P$ required can be reduced by installing a pipe loop. On distribution piping, an example of increasing delivery rate to certain locations using pipe loops, without changing pipe pressures in the rest of the pipeline, was also illustrated.

## PROBLEMS

1. A natural gas pipeline from Compton to Merced is 100 mi long and is constructed of NPS $14,0.250 \mathrm{in}$. wall thickness. The pipeline elevation profile is essentially flat. The MAOP of the pipeline is 1280 psig. The gas delivery pressure at Merced is 600 psig . What is the maximum pipeline throughput with an origin compressor station at Compton? The gas gravity is 0.6 and gas flowing temperature is $80^{\circ} \mathrm{F}$. Use the Colebrook equation for pressure drop with a friction factor of 0.01 . The compressibility factor can be assumed to be constant at 0.88 . If the flow rate increases by 50 MMSCFD , calculate the increased $H P$ required at Compton and the $H P$ required at an intermediate compressor station at Vale. Instead of the intermediate compressor station at Vale, a portion of the pipe is looped. What length of NPS 14 loop will be needed? The base pressure and base temperature are 14.7 psia and $60^{\circ} \mathrm{F}$, respectively. The compressor isentropic efficiency $=0.8$, and the mechanical efficiency of the compressor driver is 0.95 .
2. A natural gas (specific gravity $=0.60$ ) pipeline is 120 mi long and is constructed of NPS 20, 0.500 in . wall thickness (MAOP $=1000 \mathrm{psig}$ ), and runs from Akers to Coburn. At a flow rate of 250 MMSCFD , an intermediate compressor at

Bradley (milepost 65) is required. Calculate the total $H P$ required. In order to reduce the power consumption by $20 \%$ at the present flow rate, it is proposed to loop the pipeline. Calculate the length of looping required. Use the Panhandle A equation with $95 \%$ efficiency. The compressibility factor can be assumed constant at 0.90 . The gas flow temperature is $70^{\circ} \mathrm{F}$, and the base pressure and base temperature are 14.7 psia and $60^{\circ} \mathrm{F}$, respectively. The delivery pressure required at Coburn is 750 psig . The discharge pressure at Akers is 1000 psig , and the suction pressure is 850 psig. Use $80 \%$ isentropic efficiency and $95 \%$ mechanical efficiency for the compressors.
3. In a gas distribution pipeline, similar to that shown in Figure 5.8, gas enters the pipeline at A at a flow rate of 50 MMSCFD . At B and C , deliveries of 10 MMSCFD and 20 MMSCFD are made. At D, an additional volume of gas at 15 MMSCFD enters the pipeline. Calculate the pressures at the various pipe nodes A, B, C, and D , considering a delivery pressure of 500 psig at E . If the incoming volume at D is increased to 25 MMSCFD and all pressures are to remain the same, how much of the pipe DE should be looped? The pipe lengths are as follows:

AB: NPS $12,0.250$ length $=18 \mathrm{mi}$
BC: NPS 10, 0.250 length $=24 \mathrm{mi}$
CD: NPS $8,0.250$ length $=16 \mathrm{mi}$
DE: NPS 12, 0.250 length $=20 \mathrm{mi}$

## REFERENCES

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2. Engineering Data Book, 10th ed., Gas Processors Suppliers Association, Tulsa, OK, 1994.
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## CHAPTER 6

## Pipe Analysis

In this chapter we will discuss the mechanical strength needed for a pipeline transporting gas. We will analyze the impact of pipe diameter, wall thickness, material of construction, and specific safety requirements dictated by design codes and state and federal regulations. Also covered will be testing requirements and classification of pipelines based upon their proximity to human dwellings and industrial establishments and population density. The importance of mainline block valves and calculation of blowdown time to isolate sections of a gas pipeline will also be discussed.

### 6.1 PIPE WALL THICKNESS

In Chapter 3 we calculated the pressure needed to transport a given volume of gas through a pipeline. The internal pressure in a pipe causes the pipe wall to be stressed, and if allowed to reach the yield strength of the pipe material, it could cause permanent deformation of the pipe and ultimate failure. Obviously, the pipe should have sufficient strength to handle the internal pressure safely. In addition to the internal pressure due to gas flowing through the pipe, the pipe might also be subjected to external pressure.

External pressure can result from the weight of the soil above the pipe in a buried pipeline and also by the loads transmitted from vehicular traffic in areas where the pipeline is located below roads, highways, and railroads. The deeper the pipe is buried, the higher will be the soil load on the pipe. However, the pressure transmitted to the pipe due to vehicles above ground will diminish with the depth of the pipe below the ground surface. Thus, the external pressure due to vehicular loads on a buried pipeline that is 6 ft below ground will be less than that on a pipeline that is at a depth of 4 ft . In most cases involving buried pipelines transporting gas and other compressible fluids, the effect of the internal pressure is more than that of external loads. Therefore, the necessary minimum wall thickness will be dictated by the internal pressure in a gas pipeline.

The minimum wall thickness required to withstand the internal pressure in a gas pipeline will depend upon the pressure, pipe diameter, and pipe material. The larger
the pressure or diameter, the larger would be the wall thickness required. Higherstrength steel pipes will require less wall thickness to withstand the given pressure compared to low-strength materials. The commonly used formula to determine the wall thickness for internal pressure is known as Barlow's equation. This equation has been modified to take into account design factors and type of pipe joints (seamless, welded, etc.) and is incorporated into design codes such as DOT Code of Federal Regulations Part 192 and ASME B31.8 Standards. See Chapter 9 for a full list of design codes and standards used in the design, construction, and operation of gas pipelines.

### 6.2 BARLOW'S EQUATION

When a circular pipe is subject to internal pressure, the pipe material at any point will have two stress components at right angles to each other. The larger of the two stresses is known as the hoop stress and acts along the circumferential direction. Hence, it is also called the circumferential stress. The other stress is the longitudinal stress, also known as the axial stress, which acts in a direction parallel to the pipe axis. Figure 6.1 shows a cross section of a pipe subject to internal pressure. An element of the pipe wall material is shown with the two stresses $S_{h}$ and $S_{a}$ in perpendicular directions. Both stresses will increase as the internal pressure is increased. As will be shown shortly, the hoop stress $S_{h}$ is the larger of the two stresses and, hence, will govern the minimum wall thickness required for a given internal pressure.

In its basic form, Barlow's equation relates the hoop stress in the pipe wall to the internal pressure, pipe diameter, and wall thickness as follows:

$$
\begin{equation*}
S_{h}=\frac{P D}{2 t} \tag{6.1}
\end{equation*}
$$



Figure 6.1 Stresses in pipe subject to internal pressure.
where
$S_{h}=$ hoop or circumferential stress in pipe material, psi
$P=$ internal pressure, psi
$D=$ pipe outside diameter, in.
$t=$ pipe wall thickness, in.
Similar to Equation 6.1, the axial (or longitudinal) stress, $S_{a}$, is given by the following equation:

$$
\begin{equation*}
S_{a}=\frac{P D}{4 t} \tag{6.2}
\end{equation*}
$$

Note that in these equations the pipe diameter used is the outside diameter, not the inside diameter as we used in Chapter 2 and Chapter 3.

For example, consider an NPS 20 pipe, 0.500 in . wall thickness, that is subject to an internal gas pressure of 1200 psig. The pipe wall material will be stressed in the circumferential direction by the hoop stress given by Equation 6.1 as follows:

$$
S_{h}=\frac{1200 \times 20}{2 \times 0.500}=24,000 \mathrm{psig}
$$

and in accordance with Equation 6.2, the axial stress in the pipe wall is

$$
S_{a}=\frac{1200 \times 20}{4 \times 0.500}=12,000 \mathrm{psig}
$$

Barlow's equation is valid only for thin-walled cylindrical pipes. Most pipelines transporting gases and liquids generally fall in this category. There are instances in which pipes carrying gases and petroleum liquids, subject to high external loads, such as deep submarine pipelines, may be classified as thick-walled pipes. The governing equations for such thick-walled pipes are different and more complex. We will introduce these formulas for information only.

### 6.3 THICK-WALLED PIPES

Consider a thick-walled pipe with an outside diameter $D_{O}$ and inside diameter of $D_{i}$, subject to an internal pressure of P. The greatest stress in the pipe wall will be found to occur in the circumferential direction near the inner surface of the pipe. This stress can be calculated from the following equation:

$$
\begin{equation*}
S_{\max }=\frac{P\left(D_{o}^{2}+D_{i}^{2}\right)}{\left(D_{o}^{2}-D_{i}^{2}\right)} \tag{6.3}
\end{equation*}
$$

The pipe wall thickness is

$$
\begin{equation*}
t=\frac{D_{o}-D_{i}}{2} \tag{6.4}
\end{equation*}
$$

Rewriting Equation 6.3 in terms of outside diameter and wall thickness, we get

$$
S_{\max }=P\left[\frac{D_{o}^{2}+\left(D_{o}-2 t\right)^{2}}{D_{o}^{2}-\left(D_{o}-2 t\right)^{2}}\right]
$$

Simplifying further,

$$
\begin{equation*}
S_{\max }=\frac{P D_{o}}{2 t}\left[\frac{1-\left(\frac{t}{D_{o}}\right)+2\left(\frac{t}{D_{o}}\right)^{2}}{1+\left(\frac{t}{D_{o}}\right)}\right] \tag{6.5}
\end{equation*}
$$

In the limiting case, a thin-walled pipe is one in which the wall thickness is very small compared to the diameter $D_{o}$. In this case $(t / D)$ is small compared to 1 and, therefore, can be neglected in Equation 6.5. Therefore, the approximation for thinwalled pipes from Equation 6.5 becomes

$$
S_{\max }=\frac{P D_{o}}{2 t}
$$

which is the same as Barlow's Equation 6.1 for hoop stress.

## Example 1

A gas pipeline is subject to an internal pressure of 1400 psig . It is constructed of steel pipe with 24 in . outside diameter and 0.75 in . wall thickness. Calculate the maximum hoop stress in the pipeline, considering both the thin-walled approach and the thick-walled equation. What is the error in assuming that the pipe is thin walled?

## Solution

Pipe inside diameter $=24-2 \times 0.75=22.5$ in.
From Equation 6.1 for thin-walled pipe, Barlow's equation gives the maximum hoop stress as

$$
S_{h}=\frac{1400 \times 24}{2 \times 0.75}=22,400 \mathrm{psig}
$$

Considering the thick-walled pipe formula Equation 6.3,

$$
S_{\max }=\frac{1400\left(24^{2}+22.5^{2}\right)}{\left(24^{2}-22.5^{2}\right)}=21,723 \mathrm{psig}
$$

Therefore, by assuming thin-walled pipe, the hoop stress is overestimated by approximately

$$
\frac{22,400-21,723}{21,723}=0.0312 \text { or } 3.12 \%
$$

### 6.4 DERIVATION OF BARLOW'S EQUATION

Since Barlow's equation is the basic equation for pipes under internal pressure, it is appropriate to understand how the formula is derived, which is the subject of this section.

Consider a circular pipe of length $L$, outside diameter $D$, and wall thickness $t$ as shown in Figure 6.1. We consider the cross section of one-half portion of this pipe. The pipe is subject to an internal pressure of $P$ psig. Within the pipe material, the hoop stress $S_{h}$ and the axial stress $S_{a}$ act at right angles to each other as shown.

Considering the one-half section of the pipe, for balancing the forces in the direction of the hoop stress $S_{h}$, we can say that $S_{h}$, acting on the two rectangular areas $L \times t$, balances the internal pressure on the projected area $D \times L$.

Therefore,

$$
\begin{equation*}
P \times D \times L=S_{h} \times L \times t \times 2 \tag{6.6}
\end{equation*}
$$

Solving for $S_{h}$, we get the derivation of Equation 6.1 as

$$
S_{h}=\frac{P D}{2 t}
$$

Now we will look at the balancing of longitudinal forces. The internal pressure $P$ acting on the cross-sectional area of pipe $\frac{\pi}{4} D^{2}$ produces the bursting force. This is balanced by the axial resisting force $S_{a}$ acting on the area $\pi D t$. Therefore,

$$
\begin{equation*}
\frac{\pi}{4} D^{2}=S_{a} \times \pi D t \tag{6.7}
\end{equation*}
$$

Solving for $S_{a}$, we get the derivation of Equation 6.2 as

$$
S_{a}=\frac{P D}{4 t}
$$

It can be seen from the preceding equations that the hoop stress is twice the axial stress and, therefore, is the governing stress. Consider a pipe with 20 in . outside diameter and 0.500 in . wall thickness subject to an internal pressure of 1000 psig . From Barlow's Equation 6.1 and Equation 6.2, we calculate the hoop stress and axial stress as follows:

$$
\begin{aligned}
& S_{h}=\frac{1000 \times 20}{2 \times 0.500}=20,000 \mathrm{psig} \\
& S_{a}=\frac{1000 \times 20}{4 \times 0.500}=10,000 \mathrm{psig}
\end{aligned}
$$

Therefore, we are able to determine the stress levels in the pipe material for a given internal pressure, pipe diameter, and wall thickness. If the above-calculated
values are within the stress limits of the pipe material, we can conclude that the NPS 20 pipe with 0.500 in . wall thickness is adequate for the internal pressure of 1000 psig . The yield stress of the pipe material represents the stress at which the pipe material yields and undergoes permanent deformation. Therefore, we must ensure that the stress calculations above do not come dangerously close to the yield stress.

Frequently, we have to solve the reverse problem of determining the wall thickness of a pipeline for a given pressure. For example, suppose the pipe is constructed of steel with a yield strength of $52,000 \mathrm{psi}$ and we are required to determine what wall thickness is needed for NPS 20 pipe to withstand 1400 psig internal pressure. If we are allowed to stress the pipe material to no more than $60 \%$ of the yield stress, we can easily calculate the minimum wall thickness required using Equation 6.1, as follows:

$$
0.6 \times 52,000=\frac{1400 \times 20}{2 t}
$$

Here, we have equated the hoop stress per Barlow's equation to $60 \%$ yield strength of the pipe material.

Solving for pipe wall thickness, we get

$$
t=0.4487 \mathrm{in} .
$$

Suppose we used the nearest standard wall thickness of 0.500 in . The actual hoop stress can then be calculated from Barlow's equation as

$$
S_{h}=\frac{1400 \times 20}{2 \times 0.5}=28,000 \mathrm{psi}
$$

Therefore, the pipe will be stressed to $\frac{28,000}{52,000}=0.54$ or $54 \%$ of yield stress, which is less than the $60 \%$ we started with.

Incidentally, the actual axial or longitudinal stress in the preceding example will be one-half the hoop stress or 14,000 psi.

Therefore, in this basic example, we used Barlow's equation to calculate the pipe wall thickness required for a NPS 20 pipe to withstand an internal pressure of 1400 psig without stressing the pipe material beyond $60 \%$ of its yield strength.

In the foregoing, we arbitrarily picked $60 \%$ of the yield stress of pipe material to calculate the pipe wall thickness. We did not use $100 \%$ of the yield stress because, in this case, the pipe material would yield at the given pressure, which obviously cannot be allowed. In design, we generally use a design factor that is a number less than 1.00 that represents the fraction of the yield stress of the pipe material that the pipe can be stressed to. Gas pipelines are designed with various design factors ranging from 0.4 to 0.72 . This means that the pipe hoop stress is allowed to be between 40 and $72 \%$ of the yield strength of pipe material. The actual percentage will depend on various factors and will be discussed shortly. The yield stress used in the calculation of pipe wall thickness is called the specified minimum yield strength (SMYS) of pipe material. Thus, in the preceding example, we calculated the pipe wall thickness based on a design factor of 0.6 or allowed the pipe stress to go up to $60 \%$ of the SMYS.

Table 6.1 Pipe Material and Yield Strength

| Pipe Material <br> API 5LX Grade | Specified Minimum Yield Strength <br> (SMYS), psi |
| :---: | :---: |
| X42 | 42,000 |
| X46 | 46,000 |
| X52 | 52,000 |
| X56 | 56,000 |
| X60 | 60,000 |
| X65 | 65,000 |
| X70 | 70,000 |
| X80 | 80,000 |
| X90 | 90,000 |

### 6.5 PIPE MATERIAL AND GRADE

Steel pipes used in gas pipeline systems generally conform to API 5L and 5LX specifications. These are manufactured in grades ranging from X42 to X90 with SMYS, as shown in Table 6.1.

Sometimes API 5L grade B pipe with 35,000 psi SMYS is also used in certain installations.

### 6.6 INTERNAL DESIGN PRESSURE EQUATION

We indicated earlier in this chapter that Barlow's equation, in a modified form, is used in designing gas pipelines. The following form of Barlow's equation is used in design codes for petroleum transportation systems to calculate the allowable internal pressure in a pipeline based upon given diameter, wall thickness, and pipe material.

$$
\begin{equation*}
P=\frac{2 t S E F T}{D} \tag{6.8}
\end{equation*}
$$

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.
$F=$ design factor, usually 0.72 for cross-country gas pipelines, but can be as low as 0.4 , depending on class location and type of construction
$T=$ temperature deration factor $=1.00$ for temperatures below $250^{\circ} \mathrm{F}$
It must be noted that in the foregoing, we used the outside diameter of the pipe and not the inside diameter as used in Chapter 2 and Chapter 3 for pressure drop calculations.

Table 6.2 Pipe Seam Joint Factors

| Specification | Pipe Class | Seam <br> Joint Factor <br> (E) |
| :--- | :--- | :---: |
| ASTM A53 | Seamless | 1 |
|  | Electric Resistance Welded | 1 |
|  | Furnace Lap Welded | 0.8 |
|  | Furnace Butt Welded | 0.6 |
| ASTM A106 | Seamless | 1 |
| ASTM A134 | Electric Fusion Arc Welded | 0.8 |
| ASTM A135 | Electric Resistance Welded | 1 |
| ASTM A139 | Electric Fusion Welded | 0.8 |
| ASTM A211 | Spiral Welded Pipe | 0.8 |
| ASTM A333 | Seamless | 1 |
| ASTM A333 | Welded | 1 |
| ASTM A381 | Double Submerged |  |
|  | Arc Welded | 1 |
| ASTM A671 | Electric-Fusion-Welded | 1 |
| ASTM A672 | Electric-Fusion-Welded | 1 |
| ASTM A691 | Electric-Fusion-Welded | 1 |
| API 5L | Seamless | 1 |
|  | Electric Resistance Welded | 1 |
|  | Electric Flash Welded | 1 |
|  | Submerged Arc Welded | 1 |
|  | Furnace Lap Welded | 0.8 |
| API 5LX | Furnace Butt Welded | 0.6 |
|  | Seamless | 1 |
|  | Electric Resistance Welded | 1 |
| API 5LS | Electric Flash Welded | 1 |
|  | Submerged Arc Welded | 1 |
|  | Electric Resistance Welded | 1 |

The seam joint factor E used in Equation (6.8) varies with the type of pipe material and welding employed. Seam joint factors are given in Table 6.2 for the most commonly used pipe and joint types.

The internal design pressure calculated from Equation (6.8) is known as the maximum allowable operating pressure (MAOP) of the pipeline. This term has been shortened to maximum operating pressure (MOP) in recent years. Throughout this book we will use MOP and MAOP interchangeably. The design factor $F$ has values ranging from 0.4 to 0.72 , as mentioned earlier. Table 6.3 lists the values of the design factor based upon class locations. The class locations, in turn, depend on the population density in the vicinity of the pipeline.

Table 6.3 Design Factors for Steel Pipe

| Class Location | Design Factor, $\boldsymbol{F}$ |
| :---: | :---: |
| 1 | 0.72 |
| 2 | 0.60 |
| 3 | 0.50 |
| 4 | 0.40 |



Figure 6.2 Class location unit.

### 6.7 CLASS LOCATION

The following definitions of class 1 through class 4 are taken from DOT 49 CFR, Part 192 (see Reference section for details). The class location unit (CLU) is defined as an area that extends 220 yards on either side of the center line of a 1-mi section of pipe, as indicated in Figure 6.2.

## Class 1

Offshore gas pipelines are Class 1 locations. For onshore pipelines, any class location unit that has 10 or fewer buildings intended for human occupancy is termed Class 1.
Class 2
This is any class location unit that has more than 10 but fewer than 46 buildings intended for human occupancy.
Class 3
This is any class location unit that has 46 or more buildings intended for human occupancy or an area where the pipeline is within 100 yards of a building or a playground, recreation area, outdoor theatre, or other place of public assembly that is occupied by 20 or more people at least 5 days a week for 10 weeks in any 12 -month period. The days and weeks need not be consecutive.

## Class 4

This is any class location unit where buildings with four or more stories above ground exist.

The temperature deration factor $T$ is equal to 1.00 up to gas temperature $250^{\circ} \mathrm{F}$, as indicated in Table 6.4.

Table 6.4 Temperature Deration Factors

| Temperature |  |  |
| :--- | :---: | :---: |
| ${ }^{\circ} \mathbf{F}$ | ${ }^{\circ} \mathbf{C}$ | Deration Factor $\boldsymbol{T}$ |
| 250 or less | 121 or less | 1.000 |
| 300 | 149 | 0.967 |
| 350 | 177 | 0.033 |
| 400 | 204 | 0.900 |
| 450 | 232 | 0.867 |

## Example 2

A gas pipeline is constructed of API 5L X65 steel, NPS 16, 0.250 in . wall thickness. Calculate the MAOP of this pipeline for class 1 through class 4 locations. Use a temperature deration factor of 1.00 .

Solution

Using Equation 6.8, the MAOP is given by

$$
P=\frac{2 \times 0.250 \times 65,000 \times 1.0 \times 0.72 \times 1.0}{16}=1462.5 \mathrm{psig} \text { for class } 1
$$

Similarly,

$$
\begin{aligned}
& \mathrm{MAOP}=1462.5 \times \frac{0.6}{0.72}=1218.8 \mathrm{psig} \text { for class } 2 \\
& \mathrm{MAOP}=1462.5 \times \frac{0.5}{0.72}=1015.62 \mathrm{psig} \text { for class } 3 \\
& \mathrm{MAOP}=1462.5 \times \frac{0.4}{0.72}=812.5 \mathrm{psig} \text { for class } 4
\end{aligned}
$$

### 6.8 MAINLINE VALVES

Mainline valves are installed in gas pipelines so that portions of the pipeline can be isolated for hydrostatic testing and maintenance. Valves are also necessary to separate sections of pipe and minimize gas loss that can occur due to pipe rupture from construction damage. Design codes specify the spacing of these valves based upon class location, which in turn depends on the population density around the pipeline. The following lists the maximum spacing between mainline valves in gas transmission piping. These are taken from ASME B31.8 code.

| Class Location | Valve Spacing |
| :---: | :---: |
| 1 | 20 miles |
| 2 | 15 miles |
| 3 | 10 miles |
| 4 | 5 miles |

It can be seen from the preceding that the valve spacing is shorter as the pipeline traverses high-population areas. This is necessary as a safety feature to protect the inhabitants in the vicinity of the pipeline by restricting the amount of gas that might escape due to rupture of the pipeline. These mainline valves must be full-opening, through-conduit type valves such that scraper pigs and inspection tools can pass through these valves without any obstruction. Therefore, ball valves and gate valves are used of the welded construction rather than flanged type. Buried valves have
extended stems with elevated valve operators located above ground, with lubrication and bleed lines brought above ground for easy access and maintenance.

### 6.9 HYDROSTATIC TEST PRESSURE

When a pipeline is designed to operate at a certain MOP, it must be tested to ensure that it is structurally sound and can withstand safely the internal pressure before being put into service. Generally, gas pipelines are hydrotested with water by filling the test section of the pipe with water and pumping the pressure up to a value higher than the MAOP and holding it at this test pressure for a period of 4 to 8 hours. The magnitude of the test pressure is specified by design code, and it is usually $125 \%$ of the operating pressure. Thus, a pipeline designed to operate continuously at 1000 psig will be hydrotested to a minimum pressure of 1250 psig .

Consider a pipeline NPS 24, with 0.375 in. wall thickness, constructed of API 5L X65 pipe. Using a temperature deration factor of 1.00 , we calculate the MOP of this pipeline from Equation 6.8 for class 1 location as follows:

$$
P=\frac{2 \times 0.375 \times 65,000 \times 1.0 \times 0.72 \times 1.0}{24}=1462.5 \mathrm{psig}
$$

Since the pipe fittings and valves will be ANSI 600, we will establish an MOP of 1440 psig for this pipeline.

Therefore, the hydrotest pressure will be

$$
1.25 \times 1440=1800 \mathrm{psig}
$$

If the pipeline is designed to be below ground, the test pressure is held constant for a period of 8 hours, and it is thoroughly checked for leaks. Above-ground pipelines are tested for a period of 4 hours. If the design factor used in the MOP calculation is 0.72 (class 1), the hoop stress is allowed to reach $72 \%$ of the SMYS of pipe material. Testing this pipe at $125 \%$ of MOP will result in the hoop stress reaching a value of $1.25 \times 0.72=0.90$ or $90 \%$ of SMYS. Thus, by hydrotesting the pipe at 1.25 times the operating pressure, we are stressing the pipe material to $90 \%$ of the yield strength.

Generally, the hydrotest pressure is given such that the hoop stress has a range of values, such as 90 to $95 \%$ of SMYS. Therefore, in the preceding example, the minimum and maximum hydrotest pressures will be as follows:

$$
\begin{aligned}
& \text { Minimum hydrotest pressure }=1.25 \times 1440=1800 \mathrm{psig} \\
& \text { Maximum hydrotest pressure }=1800 \times(95 / 90)=1900 \mathrm{psig}
\end{aligned}
$$

It can be seen from Equation 6.1 that the 1800 psig internal pressure will cause a hoop stress of

$$
S_{h}=\frac{1800 \times 24}{2 \times 0.375}=57,600 \mathrm{psi}
$$

Dividing this hoop stress by the SMYS, we get the lower limit of the hydrotest pressure as

$$
\frac{57,600}{65,000}=0.89=89 \% \text { of SMYS }
$$

Similarly, by proportion, the maximum hydrotest pressure of 1900 psig will cause a hoop stress of

$$
S_{h}=\frac{1900 \times 57,600}{1800}=60,800=94 \% \text { of SMYS }
$$

Therefore, in this example, the hydrotest envelope of 1800 to 1900 psig is equivalent to stressing the pipe in the range of 89 to $94 \%$ of SMYS.

In the preceding analysis we have not taken into consideration the pipeline elevation profile in calculating the hydrotest pressures. Generally, a long pipeline is divided into test sections and the hydrotest pressures are established for each section, taking into account the elevations along the pipeline profile. The reason for subdividing the pipeline into sections for hydrotesting will be evident from the following example.

Consider, for example, a pipeline 50 mi long with an elevation profile as shown in Figure 6.3. The elevation of the starting point, Norwalk, is 300 ft , whereas the pipeline terminus, Lakewood, is at an elevation of 1200 ft . If the entire 50 mi length of the pipeline were filled with water for hydrotesting, the static pressure difference between the two ends due to elevation will be as follows:

$$
\text { Pressure difference }=(1200-300) \times 0.433=389.7 \mathrm{psig}
$$

The factor 0.433 is the conversion factor from feet of water to pressure in psig.
It can be seen that the pipe section at the low elevation point at Norwalk will be at a higher pressure than the pipeline at the high elevation end at Lakewood by almost 390 psig. Therefore, if we pump the water in the line to the required hydrotest


Figure 6.3 Pipeline with elevation profile-impact on hydrotest.
pressure of 1800 psig at Norwalk, the corresponding water pressure at Lakewood will be

$$
1800-390=1410 \mathrm{psig}
$$

Conversely, if we pump the water in the line to the required hydrotest pressure of 1800 psig at Lakewood, the corresponding water pressure at Norwalk will be

$$
1800+390=2190 \mathrm{psig}
$$

This is shown in Figure 6.3.
The pressure of 2190 psig at Norwalk will result in a hoop stress of

$$
S_{h}=\frac{2190 \times 24}{2 \times 0.375}=70,080 \mathrm{psi}
$$

This is equivalent to

$$
\frac{70,080}{65,000}=1.08=108 \% \text { of SMYS }
$$

Obviously, we have exceeded the yield strength of the pipe material, and this is not acceptable.

On the other hand, with 1800 psig test pressure at Norwalk, the corresponding test pressure at Lakewood is calculated to be 1410 psig. Even though the pipe section at the low end at Norwalk has the requisite test pressure ( $125 \%$ MOP), the pipe section at the higher elevation at Lakewood will see only

$$
\frac{1410}{1800} \times 125=98 \% \mathrm{MOP}
$$

This will not be an acceptable hydrotest, because we have not been able to test the entire pipeline at the correct hydrotest pressure, which must be at least $125 \%$ of the MAOP. The solution to this dilemma is to break the length of 50 mi into several sections such that each section can be tested separately at the required test pressure. These test sections will have smaller elevation differences between the ends of the test sections. Therefore, each section will be hydrotested to pressures close to the required minimum pressure. Figure 6.4 shows such a pipeline subdivided into sections suitable for hydrotesting. Using the hydrotest envelope of 90 to $95 \%$ of SMYS, we will be able to adjust the test pressures for each section such that even with some elevation difference between the ends of each test section, the hydrotest pressures may be close to the required pressures. This will not be possible if we have one single test section with significant elevation difference between the two ends, as illustrated in Figure 6.4.

Table 6.5 through Table 6.13 list the internal design pressure and hydrostatic test pressure for various pipe diameters and pipe materials ranging from X42 to X90.


Figure 6.4 Hydrotesting by subdividing pipeline.

## Example 3

A gas pipeline, NPS 20, 0.500 in. wall thickness, is constructed of API 5L X52 pipe.
(a) Calculate the design pressures for class 1 through class 4 locations.
(b) What is the range of hydrotest pressures for each of these class locations?

Assume joint factor $=1.00$ and temperature deration factor $=1.00$.

## Solution

Using Equation 6.8, the internal design pressure is

$$
P=\frac{2 \times 0.500 \times 52,000 \times 1.00 \times 1.0 \times F}{20}=2600 \mathrm{~F}
$$

where
$F=$ design factor $=0.72$ for class 1

Therefore, the design pressures for class 1 through class 4 are as follows:

Class $1=2600 \times 0.72=1872 \mathrm{psig}$
Class $2=2600 \times 0.60=1560$ psig
Class $3=2600 \times 0.50=1300 \mathrm{psig}$
Class $4=2600 \times 0.40=1040 \mathrm{psig}$

The range of hydrotest pressures is such that the hoop stress will be between 90 and $95 \%$ of SMYS.

Table 6.5 Pipeline Internal Design Pressures and Test Pressures

| Pipe Material API 5L X42 |  | SMYS <br> Weight lb/ft | 42000 psig |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Diameter | Wall Thickness |  | Internal Design Pressure, psig |  |  |  | Hydrostatic Test Pressure, psig |  |  |
| in. | in. |  | Class 1 | Class 2 | Class 3 | Class 4 | 90\% SMYS | 95\% SMYS | 100\% SMYS |
| 4.5 | 0.237 | 10.79 | 3185 | 2654 | 2212 | 1770 | 3982 | 4203 | 4424 |
|  | 0.337 | 14.98 | 4529 | 3774 | 3145 | 2516 | 5662 | 5976 | 6291 |
|  | 0.437 | 18.96 | 5873 | 4894 | 4079 | 3263 | 7342 | 7749 | 8157 |
|  | 0.531 | 22.51 | 7137 | 5947 | 4956 | 3965 | 8921 | 9416 | 9912 |
| 6.625 | 0.250 | 17.02 | 2282 | 1902 | 1585 | 1268 | 2853 | 3011 | 3170 |
|  | 0.280 | 18.97 | 2556 | 2130 | 1775 | 1420 | 3195 | 3373 | 3550 |
|  | 0.432 | 28.57 | 3944 | 3286 | 2739 | 2191 | 4930 | 5204 | 5477 |
|  | 0.562 | 36.39 | 5131 | 4275 | 3563 | 2850 | 6413 | 6769 | 7126 |
| 8.625 | 0.250 | 22.36 | 1753 | 1461 | 1217 | 974 | 2191 | 2313 | 2435 |
|  | 0.277 | 24.70 | 1942 | 1619 | 1349 | 1079 | 2428 | 2563 | 2698 |
|  | 0.322 | 28.55 | 2258 | 1882 | 1568 | 1254 | 2822 | 2979 | 3136 |
|  | 0.406 | 35.64 | 2847 | 2372 | 1977 | 1582 | 3559 | 3756 | 3954 |
| 10.75 | 0.250 | 28.04 | 1407 | 1172 | 977 | 781 | 1758 | 1856 | 1953 |
|  | 0.307 | 34.24 | 1727 | 1439 | 1199 | 960 | 2159 | 2279 | 2399 |
|  | 0.365 | 40.48 | 2054 | 1711 | 1426 | 1141 | 2567 | 2709 | 2852 |
|  | 0.500 | 54.74 | 2813 | 2344 | 1953 | 1563 | 3516 | 3712 | 3907 |
| 12.75 | 0.250 | 33.38 | 1186 | 988 | 824 | 659 | 1482 | 1565 | 1647 |
|  | 0.330 | 43.77 | 1565 | 1304 | 1087 | 870 | 1957 | 2065 | 2174 |
|  | 0.375 | 49.56 | 1779 | 1482 | 1235 | 988 | 2224 | 2347 | 2471 |
|  | 0.406 | 53.52 | 1926 | 1605 | 1337 | 1070 | 2407 | 2541 | 2675 |
|  | 0.500 | 65.42 | 2372 | 1976 | 1647 | 1318 | 2965 | 3129 | 3294 |
| 14.00 | 0.250 | 36.71 | 1080 | 900 | 750 | 600 | 1350 | 1425 | 1500 |
|  | 0.312 | 45.61 | 1348 | 1123 | 936 | 749 | 1685 | 1778 | 1872 |
|  | 0.375 | 54.57 | 1620 | 1350 | 1125 | 900 | 2025 | 2138 | 2250 |
|  | 0.437 | 63.30 | 1888 | 1573 | 1311 | 1049 | 2360 | 2491 | 2622 |
|  | 0.500 | 72.09 | 2160 | 1800 | 1500 | 1200 | 2700 | 2850 | 3000 |

Table 6.5 Pipeline Internal Design Pressures and Test Pressures (Continued)

| Pipe Material API 5L X42 |  | SMYS <br> Weight lb/ft | 42000 psig |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Diameter | Wall Thickness |  | Internal Design Pressure, psig |  |  |  | Hydrostatic Test Pressure, psig |  |  |
| in. | in. |  | Class 1 | Class 2 | Class 3 | Class 4 | 90\% SMYS | 95\% SMYS | 100\% SMYS |
| 16.00 | 0.250 | 42.05 | 945 | 788 | 656 | 525 | 1181 | 1247 | 1313 |
|  | 0.312 | 52.27 | 1179 | 983 | 819 | 655 | 1474 | 1556 | 1638 |
|  | 0.375 | 62.58 | 1418 | 1181 | 984 | 788 | 1772 | 1870 | 1969 |
|  | 0.437 | 72.64 | 1652 | 1377 | 1147 | 918 | 2065 | 2180 | 2294 |
|  | 0.500 | 82.77 | 1890 | 1575 | 1313 | 1050 | 2363 | 2494 | 2625 |
| 18.00 | 0.250 | 47.39 | 840 | 700 | 583 | 467 | 1050 | 1108 | 1167 |
|  | 0.312 | 58.94 | 1048 | 874 | 728 | 582 | 1310 | 1383 | 1456 |
|  | 0.375 | 70.59 | 1260 | 1050 | 875 | 700 | 1575 | 1663 | 1750 |
|  | 0.437 | 81.97 | 1468 | 1224 | 1020 | 816 | 1835 | 1937 | 2039 |
|  | 0.500 | 93.45 | 1680 | 1400 | 1167 | 933 | 2100 | 2217 | 2333 |
| 20.00 | 0.312 | 65.60 | 943 | 786 | 655 | 524 | 1179 | 1245 | 1310 |
|  | 0.375 | 78.60 | 1134 | 945 | 788 | 630 | 1418 | 1496 | 1575 |
|  | 0.437 | 91.30 | 1321 | 1101 | 918 | 734 | 1652 | 1744 | 1835 |
|  | 0.500 | 104.13 | 1512 | 1260 | 1050 | 840 | 1890 | 1995 | 2100 |
|  | 0.562 | 116.67 | 1699 | 1416 | 1180 | 944 | 2124 | 2242 | 2360 |
| 22.00 | 0.375 | 86.61 | 1031 | 859 | 716 | 573 | 1289 | 1360 | 1432 |
|  | 0.500 | 114.81 | 1375 | 1145 | 955 | 764 | 1718 | 1814 | 1909 |
|  | 0.625 | 142.68 | 1718 | 1432 | 1193 | 955 | 2148 | 2267 | 2386 |
|  | 0.750 | 170.21 | 2062 | 1718 | 1432 | 1145 | 2577 | 2720 | 2864 |
| 24.00 | 0.375 | 94.62 | 945 | 788 | 656 | 525 | 1181 | 1247 | 1313 |
|  | 0.437 | 109.97 | 1101 | 918 | 765 | 612 | 1377 | 1453 | 1530 |
|  | 0.500 | 125.49 | 1260 | 1050 | 875 | 700 | 1575 | 1663 | 1750 |
|  | 0.562 | 140.68 | 1416 | 1180 | 984 | 787 | 1770 | 1869 | 1967 |
|  | 0.625 | 156.03 | 1575 | 1313 | 1094 | 875 | 1969 | 2078 | 2188 |
|  | 0.750 | 186.23 | 1890 | 1575 | 1313 | 1050 | 2363 | 2494 | 2625 |
| 26.00 | 0.375 | 102.63 | 872 | 727 | 606 | 485 | 1090 | 1151 | 1212 |
|  | 0.500 | 136.17 | 1163 | 969 | 808 | 646 | 1454 | 1535 | 1615 |
|  | 0.625 | 169.38 | 1454 | 1212 | 1010 | 808 | 1817 | 1918 | 2019 |
|  | 0.750 | 202.25 | 1745 | 1454 | 1212 | 969 | 2181 | 2302 | 2423 |


| 28.00 | 0.375 | 110.64 | 810 | 675 | 563 | 450 | 1013 | 1069 | 1125 |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
|  | 0.500 | 146.85 | 1080 | 900 | 750 | 600 | 1350 | 1425 | 1500 |
|  | 0.625 | 182.73 | 1350 | 1125 | 938 | 750 | 1688 | 1781 | 1875 |
|  | 0.750 | 218.27 | 1620 | 1350 | 1125 | 900 | 2025 | 2138 | 2250 |
| 30.00 | 0.375 | 118.65 | 756 | 630 | 525 | 420 | 945 | 998 | 1050 |
|  | 0.500 | 157.53 | 1008 | 840 | 700 | 560 | 1260 | 1330 | 1400 |
|  | 0.625 | 196.08 | 1260 | 1050 | 875 | 700 | 1575 | 1663 | 1750 |
|  | 0.750 | 234.29 | 1512 | 1260 | 1050 | 840 | 1890 | 1995 | 2100 |
| 32.00 | 0.375 | 126.66 | 709 | 591 | 492 | 394 | 886 | 935 | 984 |
|  | 0.500 | 168.21 | 945 | 788 | 656 | 525 | 1181 | 1247 | 1313 |
|  | 0.625 | 209.43 | 1181 | 984 | 820 | 656 | 1477 | 1559 | 1641 |
|  | 0.750 | 250.31 | 1418 | 1181 | 984 | 788 | 1772 | 1870 | 1969 |
| 34.00 | 0.375 | 134.67 | 667 | 556 | 463 | 371 | 834 | 880 | 926 |
|  | 0.500 | 178.89 | 889 | 741 | 618 | 494 | 1112 | 1174 | 1235 |
|  | 0.625 | 222.78 | 1112 | 926 | 772 | 618 | 1390 | 1467 | 1544 |
|  | 0.750 | 266.33 | 1334 | 1112 | 926 | 741 | 1668 | 1760 | 1853 |
| 36.00 | 0.375 | 142.68 | 630 | 525 | 438 | 350 | 788 | 831 | 875 |
|  | 0.500 | 189.57 | 840 | 700 | 583 | 467 | 1050 | 1108 | 1167 |
|  | 0.625 | 236.13 | 1050 | 875 | 729 | 583 | 1313 | 1385 | 1458 |
|  | 0.750 | 282.35 | 1260 | 1050 | 875 | 700 | 1575 | 1663 | 1750 |
| 42.00 | 0.375 | 166.71 | 540 | 450 | 375 | 300 | 675 | 713 | 750 |
|  | 0.500 | 221.61 | 720 | 600 | 500 | 400 | 900 | 950 | 1000 |
|  | 0.625 | 276.18 | 900 | 750 | 625 | 500 | 1125 | 1188 | 1250 |
|  | 0.750 | 330.41 | 1080 | 900 | 750 | 600 | 1350 | 1425 | 1500 |
|  | 1.000 | 437.88 | 1440 | 1200 | 1000 | 800 | 1800 | 1900 | 2000 |

Table 6．6 Pipeline Internal Design Pressures and Test Pressures

| Pipe Material API 5L X46 |  | SMYS <br> Weight lb／ft | 46000 psig |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Diameter | Wall Thickness |  | Internal Design Pressure，psig |  |  |  | Hydrostatic Test Pressure，psig |  |  |
| in． | in． |  | Class 1 | Class 2 | Class 3 | Class 4 | 90\％SMYS | 95\％SMYS | 100\％SMYS |
| 4.5 | 0.237 | 10.79 | 3489 | 2907 | 2423 | 1938 | 4361 | 4603 | 4845 |
|  | 0.337 | 14.98 | 4961 | 4134 | 3445 | 2756 | 6201 | 6545 | 6890 |
|  | 0.437 | 18.96 | 6433 | 5361 | 4467 | 3574 | 8041 | 8488 | 8934 |
|  | 0.531 | 22.51 | 7816 | 6514 | 5428 | 4342 | 9770 | 10313 | 10856 |
| 6.625 | 0.250 | 17.02 | 2500 | 2083 | 1736 | 1389 | 3125 | 3298 | 3472 |
|  | 0.280 | 18.97 | 2800 | 2333 | 1944 | 1555 | 3499 | 3694 | 3888 |
|  | 0.432 | 28.57 | 4319 | 3599 | 3000 | 2400 | 5399 | 5699 | 5999 |
|  | 0.562 | 36.39 | 5619 | 4683 | 3902 | 3122 | 7024 | 7414 | 7804 |
| 8.625 | 0.250 | 22.36 | 1920 | 1600 | 1333 | 1067 | 2400 | 2533 | 2667 |
|  | 0.277 | 24.70 | 2127 | 1773 | 1477 | 1182 | 2659 | 2807 | 2955 |
|  | 0.322 | 28.55 | 2473 | 2061 | 1717 | 1374 | 3091 | 3263 | 3435 |
|  | 0.406 | 35.64 | 3118 | 2598 | 2165 | 1732 | 3898 | 4114 | 4331 |
| 10.75 | 0.250 | 28.04 | 1540 | 1284 | 1070 | 856 | 1926 | 2033 | 2140 |
|  | 0.307 | 34.24 | 1892 | 1576 | 1314 | 1051 | 2365 | 2496 | 2627 |
|  | 0.365 | 40.48 | 2249 | 1874 | 1562 | 1249 | 2811 | 2968 | 3124 |
|  | 0.500 | 54.74 | 3081 | 2567 | 2140 | 1712 | 3851 | 4065 | 4279 |
| 12.75 | 0.250 | 33.38 | 1299 | 1082 | 902 | 722 | 1624 | 1714 | 1804 |
|  | 0.330 | 43.77 | 1714 | 1429 | 1191 | 952 | 2143 | 2262 | 2381 |
|  | 0.375 | 49.56 | 1948 | 1624 | 1353 | 1082 | 2435 | 2571 | 2706 |
|  | 0.406 | 53.52 | 2109 | 1758 | 1465 | 1172 | 2637 | 2783 | 2930 |
|  | 0.500 | 65.42 | 2598 | 2165 | 1804 | 1443 | 3247 | 3427 | 3608 |
| 14.00 | 0.250 | 36.71 | 1183 | 986 | 821 | 657 | 1479 | 1561 | 1643 |
|  | 0.312 | 45.61 | 1476 | 1230 | 1025 | 820 | 1845 | 1948 | 2050 |
|  | 0.375 | 54.57 | 1774 | 1479 | 1232 | 986 | 2218 | 2341 | 2464 |
|  | 0.437 | 63.30 | 2068 | 1723 | 1436 | 1149 | 2585 | 2728 | 2872 |
|  | 0.500 | 72.09 | 2366 | 1971 | 1643 | 1314 | 2957 | 3121 | 3286 |
| 16.00 | 0.250 | 42.05 | 1035 | 863 | 719 | 575 | 1294 | 1366 | 1438 |
|  | 0.312 | 52.27 | 1292 | 1076 | 897 | 718 | 1615 | 1704 | 1794 |


|  | 0.375 | 62.58 | 1553 | 1294 | 1078 | 863 | 1941 | 2048 | 2156 |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
|  | 0.437 | 72.64 | 1809 | 1508 | 1256 | 1005 | 2261 | 2387 | 2513 |
|  | 0.500 | 82.77 | 2070 | 1725 | 1438 | 1150 | 2588 | 2731 | 2875 |
| 18.00 | 0.250 | 47.39 | 920 | 767 | 639 | 511 | 1150 | 1214 | 1278 |
|  | 0.312 | 58.94 | 1148 | 957 | 797 | 638 | 1435 | 1515 | 1595 |
|  | 0.375 | 70.59 | 1380 | 1150 | 958 | 767 | 1725 | 1821 | 1917 |
|  | 0.437 | 81.97 | 1608 | 1340 | 1117 | 893 | 2010 | 2122 | 2234 |
|  | 0.500 | 93.45 | 1840 | 1533 | 1278 | 1022 | 2300 | 2428 | 2556 |
| 20.00 | 0.312 | 65.60 | 1033 | 861 | 718 | 574 | 1292 | 1363 | 1435 |
|  | 0.375 | 78.60 | 1242 | 1035 | 863 | 690 | 1553 | 1639 | 1725 |
|  | 0.437 | 91.30 | 1447 | 1206 | 1005 | 804 | 1809 | 1910 | 2010 |
|  | 0.500 | 104.13 | 1656 | 1380 | 1150 | 920 | 2070 | 2185 | 2300 |
|  | 0.562 | 116.67 | 1861 | 1551 | 1293 | 1034 | 2327 | 2456 | 2585 |
| 22.00 | 0.375 | 86.61 | 1129 | 941 | 784 | 627 | 1411 | 1490 | 1568 |
|  | 0.500 | 114.81 | 1505 | 1255 | 1045 | 836 | 1882 | 1986 | 2091 |
|  | 0.625 | 142.68 | 1882 | 1568 | 1307 | 1045 | 2352 | 2483 | 2614 |
|  | 0.750 | 170.21 | 2258 | 1882 | 1568 | 1255 | 2823 | 2980 | 3136 |
| 24.00 | 0.375 | 94.62 | 1035 | 863 | 719 | 575 | 1294 | 1366 | 1438 |
|  | 0.437 | 109.97 | 1206 | 1005 | 838 | 670 | 1508 | 1591 | 1675 |
|  | 0.500 | 125.49 | 1380 | 1150 | 958 | 767 | 1725 | 1821 | 1917 |
|  | 0.562 | 140.68 | 1551 | 1293 | 1077 | 862 | 1939 | 2047 | 2154 |
|  | 0.625 | 156.03 | 1725 | 1438 | 1198 | 958 | 2156 | 2276 | 2396 |
|  | 0.750 | 186.23 | 2070 | 1725 | 1438 | 1150 | 2588 | 2731 | 2875 |
| 26.00 | 0.375 | 102.63 | 955 | 796 | 663 | 531 | 1194 | 1261 | 1327 |
|  | 0.500 | 136.17 | 1274 | 1062 | 885 | 708 | 1592 | 1681 | 1769 |
|  | 0.625 | 169.38 | 1592 | 1327 | 1106 | 885 | 1990 | 2101 | 2212 |
|  | 0.750 | 202.25 | 1911 | 1592 | 1327 | 1062 | 2388 | 2521 | 2654 |
| 28.00 | 0.375 | 110.64 | 887 | 739 | 616 | 493 | 1109 | 1171 | 1232 |
|  | 0.500 | 146.85 | 1183 | 986 | 821 | 657 | 1479 | 1561 | 1643 |
|  | 0.625 | 182.73 | 1479 | 1232 | 1027 | 821 | 1848 | 1951 | 2054 |
|  | 0.750 | 218.27 | 1774 | 1479 | 1232 | 986 | 2218 | 2341 | 2464 |
| 30.00 | 0.375 | 118.65 | 828 | 690 | 575 | 460 | 1035 | 1093 | 1150 |

Table 6.6 Pipeline Internal Design Pressures and Test Pressures (Continued)

| Pipe Material API 5L X46 |  | SMYS <br> Weight lb/ft | 46000 psig |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Diameter | Wall Thickness |  | Internal Design Pressure, psig |  |  |  | Hydrostatic Test Pressure, psig |  |  |
| in. | in. |  | Class 1 | Class 2 | Class 3 | Class 4 | 90\% SMYS | 95\% SMYS | 100\% SMYS |
|  | 0.500 | 157.53 | 1104 | 920 | 767 | 613 | 1380 | 1457 | 1533 |
|  | 0.625 | 196.08 | 1380 | 1150 | 958 | 767 | 1725 | 1821 | 1917 |
|  | 0.750 | 234.29 | 1656 | 1380 | 1150 | 920 | 2070 | 2185 | 2300 |
| 32.00 | 0.375 | 126.66 | 776 | 647 | 539 | 431 | 970 | 1024 | 1078 |
|  | 0.500 | 168.21 | 1035 | 863 | 719 | 575 | 1294 | 1366 | 1438 |
|  | 0.625 | 209.43 | 1294 | 1078 | 898 | 719 | 1617 | 1707 | 1797 |
|  | 0.750 | 250.31 | 1553 | 1294 | 1078 | 863 | 1941 | 2048 | 2156 |
| 34.00 | 0.375 | 134.67 | 731 | 609 | 507 | 406 | 913 | 964 | 1015 |
|  | 0.500 | 178.89 | 974 | 812 | 676 | 541 | 1218 | 1285 | 1353 |
|  | 0.625 | 222.78 | 1218 | 1015 | 846 | 676 | 1522 | 1607 | 1691 |
|  | 0.750 | 266.33 | 1461 | 1218 | 1015 | 812 | 1826 | 1928 | 2029 |
| 36.00 | 0.375 | 142.68 | 690 | 575 | 479 | 383 | 863 | 910 | 958 |
|  | 0.500 | 189.57 | 920 | 767 | 639 | 511 | 1150 | 1214 | 1278 |
|  | 0.625 | 236.13 | 1150 | 958 | 799 | 639 | 1438 | 1517 | 1597 |
|  | 0.750 | 282.35 | 1380 | 1150 | 958 | 767 | 1725 | 1821 | 1917 |
| 42.00 | 0.375 | 166.71 | 591 | 493 | 411 | 329 | 739 | 780 | 821 |
|  | 0.500 | 221.61 | 789 | 657 | 548 | 438 | 986 | 1040 | 1095 |
|  | 0.625 | 276.18 | 986 | 821 | 685 | 548 | 1232 | 1301 | 1369 |
|  | 0.750 | 330.41 | 1183 | 986 | 821 | 657 | 1479 | 1561 | 1643 |
|  | 1.000 | 437.88 | 1577 | 1314 | 1095 | 876 | 1971 | 2081 | 2190 |

Table 6.7 Pipeline Internal Design Pressures and Test Pressures

| Pipe Material API 5L X52 |  | SMYS <br> Weight lb/ft | 52000 psig |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Diameter | Wall Thickness |  | Internal Design Pressure, psig |  |  |  | Hydrostatic Test Pressure, psig |  |  |
| in. | in. |  | Class 1 | Class 2 | Class 3 | Class 4 | 90\% SMYS | 95\% SMYS | 100\% SMYS |
| 4.5 | 0.237 | 10.79 | 3944 | 3286 | 2739 | 2191 | 4930 | 5203 | 5477 |
|  | 0.337 | 14.98 | 5608 | 4673 | 3894 | 3115 | 7010 | 7399 | 7788 |
|  | 0.437 | 18.96 | 7272 | 6060 | 5050 | 4040 | 9090 | 9595 | 10100 |
|  | 0.531 | 22.51 | 8836 | 7363 | 6136 | 4909 | 11045 | 11658 | 12272 |
| 6.625 | 0.250 | 17.02 | 2826 | 2355 | 1962 | 1570 | 3532 | 3728 | 3925 |
|  | 0.280 | 18.97 | 3165 | 2637 | 2198 | 1758 | 3956 | 4176 | 4395 |
|  | 0.432 | 28.57 | 4883 | 4069 | 3391 | 2713 | 6103 | 6443 | 6782 |
|  | 0.562 | 36.39 | 6352 | 5293 | 4411 | 3529 | 7940 | 8381 | 8822 |
| 8.625 | 0.250 | 22.36 | 2170 | 1809 | 1507 | 1206 | 2713 | 2864 | 3014 |
|  | 0.277 | 24.70 | 2405 | 2004 | 1670 | 1336 | 3006 | 3173 | 3340 |
|  | 0.322 | 28.55 | 2796 | 2330 | 1941 | 1553 | 3494 | 3689 | 3883 |
|  | 0.406 | 35.64 | 3525 | 2937 | 2448 | 1958 | 4406 | 4651 | 4896 |
| 10.75 | 0.250 | 28.04 | 1741 | 1451 | 1209 | 967 | 2177 | 2298 | 2419 |
|  | 0.307 | 34.24 | 2138 | 1782 | 1485 | 1188 | 2673 | 2822 | 2970 |
|  | 0.365 | 40.48 | 2542 | 2119 | 1766 | 1412 | 3178 | 3355 | 3531 |
|  | 0.500 | 54.74 | 3483 | 2902 | 2419 | 1935 | 4353 | 4595 | 4837 |
| 12.75 | 0.250 | 33.38 | 1468 | 1224 | 1020 | 816 | 1835 | 1937 | 2039 |
|  | 0.330 | 43.77 | 1938 | 1615 | 1346 | 1077 | 2423 | 2557 | 2692 |
|  | 0.375 | 49.56 | 2202 | 1835 | 1529 | 1224 | 2753 | 2906 | 3059 |
|  | 0.406 | 53.52 | 2384 | 1987 | 1656 | 1325 | 2981 | 3146 | 3312 |
|  | 0.500 | 65.42 | 2936 | 2447 | 2039 | 1631 | 3671 | 3875 | 4078 |
| 14.00 | 0.250 | 36.71 | 1337 | 1114 | 929 | 743 | 1671 | 1764 | 1857 |
|  | 0.312 | 45.61 | 1669 | 1391 | 1159 | 927 | 2086 | 2202 | 2318 |
|  | 0.375 | 54.57 | 2006 | 1671 | 1393 | 1114 | 2507 | 2646 | 2786 |
|  | 0.437 | 63.30 | 2337 | 1948 | 1623 | 1299 | 2922 | 3084 | 3246 |
|  | 0.500 | 72.09 | 2674 | 2229 | 1857 | 1486 | 3343 | 3529 | 3714 |

Table 6.7 Pipeline Internal Design Pressures and Test Pressures (Continued)

| Pipe Material API 5L X52 |  | SMYS <br> Weight lb/ft | 52000 psig |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Diameter | Wall Thickness |  | Internal Design Pressure, psig |  |  |  | Hydrostatic Test Pressure, psig |  |  |
| in. | in. |  | Class 1 | Class 2 | Class 3 | Class 4 | 90\% SMYS | 95\% SMYS | 100\% SMYS |
| 16.00 | 0.250 | 42.05 | 1170 | 975 | 813 | 650 | 1463 | 1544 | 1625 |
|  | 0.312 | 52.27 | 1460 | 1217 | 1014 | 811 | 1825 | 1927 | 2028 |
|  | 0.375 | 62.58 | 1755 | 1463 | 1219 | 975 | 2194 | 2316 | 2438 |
|  | 0.437 | 72.64 | 2045 | 1704 | 1420 | 1136 | 2556 | 2698 | 2841 |
|  | 0.500 | 82.77 | 2340 | 1950 | 1625 | 1300 | 2925 | 3088 | 3250 |
| 18.00 | 0.250 | 47.39 | 1040 | 867 | 722 | 578 | 1300 | 1372 | 1444 |
|  | 0.312 | 58.94 | 1298 | 1082 | 901 | 721 | 1622 | 1713 | 1803 |
|  | 0.375 | 70.59 | 1560 | 1300 | 1083 | 867 | 1950 | 2058 | 2167 |
|  | 0.437 | 81.97 | 1818 | 1515 | 1262 | 1010 | 2272 | 2399 | 2525 |
|  | 0.500 | 93.45 | 2080 | 1733 | 1444 | 1156 | 2600 | 2744 | 2889 |
| 20.00 | 0.312 | 65.60 | 1168 | 973 | 811 | 649 | 1460 | 1541 | 1622 |
|  | 0.375 | 78.60 | 1404 | 1170 | 975 | 780 | 1755 | 1853 | 1950 |
|  | 0.437 | 91.30 | 1636 | 1363 | 1136 | 909 | 2045 | 2159 | 2272 |
|  | 0.500 | 104.13 | 1872 | 1560 | 1300 | 1040 | 2340 | 2470 | 2600 |
|  | 0.562 | 116.67 | 2104 | 1753 | 1461 | 1169 | 2630 | 2776 | 2922 |
| 22.00 | 0.375 | 86.61 | 1276 | 1064 | 886 | 709 | 1595 | 1684 | 1773 |
|  | 0.500 | 114.81 | 1702 | 1418 | 1182 | 945 | 2127 | 2245 | 2364 |
|  | 0.625 | 142.68 | 2127 | 1773 | 1477 | 1182 | 2659 | 2807 | 2955 |
|  | 0.750 | 170.21 | 2553 | 2127 | 1773 | 1418 | 3191 | 3368 | 3545 |
| 24.00 | 0.375 | 94.62 | 1170 | 975 | 813 | 650 | 1463 | 1544 | 1625 |
|  | 0.437 | 109.97 | 1363 | 1136 | 947 | 757 | 1704 | 1799 | 1894 |
|  | 0.500 | 125.49 | 1560 | 1300 | 1083 | 867 | 1950 | 2058 | 2167 |
|  | 0.562 | 140.68 | 1753 | 1461 | 1218 | 974 | 2192 | 2314 | 2435 |
|  | 0.625 | 156.03 | 1950 | 1625 | 1354 | 1083 | 2438 | 2573 | 2708 |
|  | 0.750 | 186.23 | 2340 | 1950 | 1625 | 1300 | 2925 | 3088 | 3250 |
| 26.00 | 0.375 | 102.63 | 1080 | 900 | 750 | 600 | 1350 | 1425 | 1500 |
|  | 0.500 | 136.17 | 1440 | 1200 | 1000 | 800 | 1800 | 1900 | 2000 |
|  | 0.625 | 169.38 | 1800 | 1500 | 1250 | 1000 | 2250 | 2375 | 2500 |
|  | 0.750 | 202.25 | 2160 | 1800 | 1500 | 1200 | 2700 | 2850 | 3000 |


| 28.00 | 0.375 | 110.64 | 1003 | 836 | 696 | 557 | 1254 | 1323 | 1393 |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
|  | 0.500 | 146.85 | 1337 | 1114 | 929 | 743 | 1671 | 1764 | 1857 |
|  | 0.625 | 182.73 | 1671 | 1393 | 1161 | 929 | 2089 | 2205 | 2321 |
|  | 0.750 | 218.27 | 2006 | 1671 | 1393 | 1114 | 2507 | 2646 | 2786 |
| 30.00 | 0.375 | 118.65 | 936 | 780 | 650 | 520 | 1170 | 1235 | 1300 |
|  | 0.500 | 157.53 | 1248 | 1040 | 867 | 693 | 1560 | 1647 | 1733 |
|  | 0.625 | 196.08 | 1560 | 1300 | 1083 | 867 | 1950 | 2058 | 2167 |
|  | 0.750 | 234.29 | 1872 | 1560 | 1300 | 1040 | 2340 | 2470 | 2600 |
| 32.00 | 0.375 | 126.66 | 878 | 731 | 609 | 488 | 1097 | 1158 | 1219 |
|  | 0.500 | 168.21 | 1170 | 975 | 813 | 650 | 1463 | 1544 | 1625 |
|  | 0.625 | 209.43 | 1463 | 1219 | 1016 | 813 | 1828 | 1930 | 2031 |
|  | 0.750 | 250.31 | 1755 | 1463 | 1219 | 975 | 2194 | 2316 | 2438 |
| 34.00 | 0.375 | 134.67 | 826 | 688 | 574 | 459 | 1032 | 1090 | 1147 |
|  | 0.500 | 178.89 | 1101 | 918 | 765 | 612 | 1376 | 1453 | 1529 |
|  | 0.625 | 222.78 | 1376 | 1147 | 956 | 765 | 1721 | 1816 | 1912 |
|  | 0.750 | 266.33 | 1652 | 1376 | 1147 | 918 | 2065 | 2179 | 2294 |
| 36.00 | 0.375 | 142.68 | 780 | 650 | 542 | 433 | 975 | 1029 | 1083 |
|  | 0.500 | 189.57 | 1040 | 867 | 722 | 578 | 1300 | 1372 | 1444 |
|  | 0.625 | 236.13 | 1300 | 1083 | 903 | 722 | 1625 | 1715 | 1806 |
|  | 0.750 | 282.35 | 1560 | 1300 | 1083 | 867 | 1950 | 2058 | 2167 |
| 42.00 | 0.375 | 166.71 | 669 | 557 | 464 | 371 | 836 | 882 | 929 |
|  | 0.500 | 221.61 | 891 | 743 | 619 | 495 | 1114 | 1176 | 1238 |
|  | 0.625 | 276.18 | 1114 | 929 | 774 | 619 | 1393 | 1470 | 1548 |
|  | 0.750 | 330.41 | 1337 | 1114 | 929 | 743 | 1671 | 1764 | 1857 |
|  | 1.000 | 437.88 | 1783 | 1486 | 1238 | 990 | 2229 | 2352 | 2476 |

Table 6.8 Pipeline Internal Design Pressures and Test Pressures

| Pipe Material API 5L X56 |  | SMYS <br> Weight lb/ft | 56000 psig |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Diameter | Wall Thickness |  | Internal Design Pressure, psig |  |  |  | Hydrostatic Test Pressure, psig |  |  |
| in. | in. |  | Class 1 | Class 2 | Class 3 | Class 4 | 90\% SMYS | 95\% SMYS | 100\% SMYS |
| 4.5 | 0.237 | 10.79 | 4247 | 3539 | 2949 | 2359 | 5309 | 5604 | 5899 |
|  | 0.337 | 14.98 | 6039 | 5033 | 4194 | 3355 | 7549 | 7968 | 8388 |
|  | 0.437 | 18.96 | 7831 | 6526 | 5438 | 4351 | 9789 | 10333 | 10876 |
|  | 0.531 | 22.51 | 9516 | 7930 | 6608 | 5286 | 11894 | 12555 | 13216 |
| 6.625 | 0.250 | 17.02 | 3043 | 2536 | 2113 | 1691 | 3804 | 4015 | 4226 |
|  | 0.280 | 18.97 | 3408 | 2840 | 2367 | 1893 | 4260 | 4497 | 4734 |
|  | 0.432 | 28.57 | 5258 | 4382 | 3652 | 2921 | 6573 | 6938 | 7303 |
|  | 0.562 | 36.39 | 6841 | 5701 | 4750 | 3800 | 8551 | 9026 | 9501 |
| 8.625 | 0.250 | 22.36 | 2337 | 1948 | 1623 | 1299 | 2922 | 3084 | 3246 |
|  | 0.277 | 24.70 | 2590 | 2158 | 1798 | 1439 | 3237 | 3417 | 3597 |
|  | 0.322 | 28.55 | 3011 | 2509 | 2091 | 1673 | 3763 | 3972 | 4181 |
|  | 0.406 | 35.64 | 3796 | 3163 | 2636 | 2109 | 4745 | 5009 | 5272 |
| 10.75 | 0.250 | 28.04 | 1875 | 1563 | 1302 | 1042 | 2344 | 2474 | 2605 |
|  | 0.307 | 34.24 | 2303 | 1919 | 1599 | 1279 | 2879 | 3039 | 3199 |
|  | 0.365 | 40.48 | 2738 | 2282 | 1901 | 1521 | 3423 | 3613 | 3803 |
|  | 0.500 | 54.74 | 3751 | 3126 | 2605 | 2084 | 4688 | 4949 | 5209 |
| 12.75 | 0.250 | 33.38 | 1581 | 1318 | 1098 | 878 | 1976 | 2086 | 2196 |
|  | 0.330 | 43.77 | 2087 | 1739 | 1449 | 1160 | 2609 | 2754 | 2899 |
|  | 0.375 | 49.56 | 2372 | 1976 | 1647 | 1318 | 2965 | 3129 | 3294 |
|  | 0.406 | 53.52 | 2568 | 2140 | 1783 | 1427 | 3210 | 3388 | 3566 |
|  | 0.500 | 65.42 | 3162 | 2635 | 2196 | 1757 | 3953 | 4173 | 4392 |
| 14.00 | 0.250 | 36.71 | 1440 | 1200 | 1000 | 800 | 1800 | 1900 | 2000 |
|  | 0.312 | 45.61 | 1797 | 1498 | 1248 | 998 | 2246 | 2371 | 2496 |
|  | 0.375 | 54.57 | 2160 | 1800 | 1500 | 1200 | 2700 | 2850 | 3000 |
|  | 0.437 | 63.30 | 2517 | 2098 | 1748 | 1398 | 3146 | 3321 | 3496 |
|  | 0.500 | 72.09 | 2880 | 2400 | 2000 | 1600 | 3600 | 3800 | 4000 |
| 16.00 | 0.250 | 42.05 | 1260 | 1050 | 875 | 700 | 1575 | 1663 | 1750 |
|  | 0.312 | 52.27 | 1572 | 1310 | 1092 | 874 | 1966 | 2075 | 2184 |


|  | 0.375 | 62.58 | 1890 | 1575 | 1313 | 1050 | 2363 | 2494 | 2625 |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
|  | 0.437 | 72.64 | 2202 | 1835 | 1530 | 1224 | 2753 | 2906 | 3059 |
|  | 0.500 | 82.77 | 2520 | 2100 | 1750 | 1400 | 3150 | 3325 | 3500 |
| 18.00 | 0.250 | 47.39 | 1120 | 933 | 778 | 622 | 1400 | 1478 | 1556 |
|  | 0.312 | 58.94 | 1398 | 1165 | 971 | 777 | 1747 | 1844 | 1941 |
|  | 0.375 | 70.59 | 1680 | 1400 | 1167 | 933 | 2100 | 2217 | 2333 |
|  | 0.437 | 81.97 | 1958 | 1631 | 1360 | 1088 | 2447 | 2583 | 2719 |
|  | 0.500 | 93.45 | 2240 | 1867 | 1556 | 1244 | 2800 | 2956 | 3111 |
| 20.00 | 0.312 | 65.60 | 1258 | 1048 | 874 | 699 | 1572 | 1660 | 1747 |
|  | 0.375 | 78.60 | 1512 | 1260 | 1050 | 840 | 1890 | 1995 | 2100 |
|  | 0.437 | 91.30 | 1762 | 1468 | 1224 | 979 | 2202 | 2325 | 2447 |
|  | 0.500 | 104.13 | 2016 | 1680 | 1400 | 1120 | 2520 | 2660 | 2800 |
|  | 0.562 | 116.67 | 2266 | 1888 | 1574 | 1259 | 2832 | 2990 | 3147 |
| 22.00 | 0.375 | 86.61 | 1375 | 1145 | 955 | 764 | 1718 | 1814 | 1909 |
|  | 0.500 | 114.81 | 1833 | 1527 | 1273 | 1018 | 2291 | 2418 | 2545 |
|  | 0.625 | 142.68 | 2291 | 1909 | 1591 | 1273 | 2864 | 3023 | 3182 |
|  | 0.750 | 170.21 | 2749 | 2291 | 1909 | 1527 | 3436 | 3627 | 3818 |
| 24.00 | 0.375 | 94.62 | 1260 | 1050 | 875 | 700 | 1575 | 1663 | 1750 |
|  | 0.437 | 109.97 | 1468 | 1224 | 1020 | 816 | 1835 | 1937 | 2039 |
|  | 0.500 | 125.49 | 1680 | 1400 | 1167 | 933 | 2100 | 2217 | 2333 |
|  | 0.562 | 140.68 | 1888 | 1574 | 1311 | 1049 | 2360 | 2492 | 2623 |
|  | 0.625 | 156.03 | 2100 | 1750 | 1458 | 1167 | 2625 | 2771 | 2917 |
|  | 0.750 | 186.23 | 2520 | 2100 | 1750 | 1400 | 3150 | 3325 | 3500 |
| 26.00 | 0.375 | 102.63 | 1163 | 969 | 808 | 646 | 1454 | 1535 | 1615 |
|  | 0.500 | 136.17 | 1551 | 1292 | 1077 | 862 | 1938 | 2046 | 2154 |
|  | 0.625 | 169.38 | 1938 | 1615 | 1346 | 1077 | 2423 | 2558 | 2692 |
|  | 0.750 | 202.25 | 2326 | 1938 | 1615 | 1292 | 2908 | 3069 | 3231 |
| 28.00 | 0.375 | 110.64 | 1080 | 900 | 750 | 600 | 1350 | 1425 | 1500 |
|  | 0.500 | 146.85 | 1440 | 1200 | 1000 | 800 | 1800 | 1900 | 2000 |
|  | 0.625 | 182.73 | 1800 | 1500 | 1250 | 1000 | 2250 | 2375 | 2500 |
|  | 0.750 | 218.27 | 2160 | 1800 | 1500 | 1200 | 2700 | 2850 | 3000 |

Table 6.8 Pipeline Internal Design Pressures and Test Pressures (Continued)

| Pipe Material API 5L X56 |  | SMYS <br> Weight lb/ft | 56000 psig |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Diameter | Wall Thickness |  | Internal Design Pressure, psig |  |  |  | Hydrostatic Test Pressure, psig |  |  |
| in. | in. |  | Class 1 | Class 2 | Class 3 | Class 4 | 90\% SMYS | 95\% SMYS | 100\% SMYS |
| 30.00 | 0.375 | 118.65 | 1008 | 840 | 700 | 560 | 1260 | 1330 | 1400 |
|  | 0.500 | 157.53 | 1344 | 1120 | 933 | 747 | 1680 | 1773 | 1867 |
|  | 0.625 | 196.08 | 1680 | 1400 | 1167 | 933 | 2100 | 2217 | 2333 |
|  | 0.750 | 234.29 | 2016 | 1680 | 1400 | 1120 | 2520 | 2660 | 2800 |
| 32.00 | 0.375 | 126.66 | 945 | 788 | 656 | 525 | 1181 | 1247 | 1313 |
|  | 0.500 | 168.21 | 1260 | 1050 | 875 | 700 | 1575 | 1663 | 1750 |
|  | 0.625 | 209.43 | 1575 | 1313 | 1094 | 875 | 1969 | 2078 | 2188 |
|  | 0.750 | 250.31 | 1890 | 1575 | 1313 | 1050 | 2363 | 2494 | 2625 |
| 34.00 | 0.375 | 134.67 | 889 | 741 | 618 | 494 | 1112 | 1174 | 1235 |
|  | 0.500 | 178.89 | 1186 | 988 | 824 | 659 | 1482 | 1565 | 1647 |
|  | 0.625 | 222.78 | 1482 | 1235 | 1029 | 824 | 1853 | 1956 | 2059 |
|  | 0.750 | 266.33 | 1779 | 1482 | 1235 | 988 | 2224 | 2347 | 2471 |
| 36.00 | 0.375 | 142.68 | 840 | 700 | 583 | 467 | 1050 | 1108 | 1167 |
|  | 0.500 | 189.57 | 1120 | 933 | 778 | 622 | 1400 | 1478 | 1556 |
|  | 0.625 | 236.13 | 1400 | 1167 | 972 | 778 | 1750 | 1847 | 1944 |
|  | 0.750 | 282.35 | 1680 | 1400 | 1167 | 933 | 2100 | 2217 | 2333 |
| 42.00 | 0.375 | 166.71 | 720 | 600 | 500 | 400 | 900 | 950 | 1000 |
|  | 0.500 | 221.61 | 960 | 800 | 667 | 533 | 1200 | 1267 | 1333 |
|  | 0.625 | 276.18 | 1200 | 1000 | 833 | 667 | 1500 | 1583 | 1667 |
|  | 0.750 | 330.41 | 1440 | 1200 | 1000 | 800 | 1800 | 1900 | 2000 |
|  | 1.000 | 437.88 | 1920 | 1600 | 1333 | 1067 | 2400 | 2533 | 2667 |

Table 6.9 Pipeline Internal Design Pressures and Test Pressures

| Pipe Material API 5L X60 |  | SMYS <br> Weight lb/ft | 60000 psig |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Diameter | Wall Thickness |  | Internal Design Pressure, psig |  |  |  | Hydrostatic Test Pressure, psig |  |  |
| in. | in. |  | Class 1 | Class 2 | Class 3 | Class 4 | 90\% SMYS | 95\% SMYS | 100\% SMYS |
| 4.5 | 0.237 | 10.79 | 4550 | 3792 | 3160 | 2528 | 5688 | 6004 | 6320 |
|  | 0.337 | 14.98 | 6470 | 5392 | 4493 | 3595 | 8088 | 8537 | 8987 |
|  | 0.437 | 18.96 | 8390 | 6992 | 5827 | 4661 | 10488 | 11071 | 11653 |
|  | 0.531 | 22.51 | 10195 | 8496 | 7080 | 5664 | 12744 | 13452 | 14160 |
| 6.625 | 0.250 | 17.02 | 3260 | 2717 | 2264 | 1811 | 4075 | 4302 | 4528 |
|  | 0.280 | 18.97 | 3652 | 3043 | 2536 | 2029 | 4565 | 4818 | 5072 |
|  | 0.432 | 28.57 | 5634 | 4695 | 3912 | 3130 | 7042 | 7434 | 7825 |
|  | 0.562 | 36.39 | 7329 | 6108 | 5090 | 4072 | 9162 | 9671 | 10180 |
| 8.625 | 0.250 | 22.36 | 2504 | 2087 | 1739 | 1391 | 3130 | 3304 | 3478 |
|  | 0.277 | 24.70 | 2775 | 2312 | 1927 | 1542 | 3469 | 3661 | 3854 |
|  | 0.322 | 28.55 | 3226 | 2688 | 2240 | 1792 | 4032 | 4256 | 4480 |
|  | 0.406 | 35.64 | 4067 | 3389 | 2824 | 2259 | 5084 | 5366 | 5649 |
| 10.75 | 0.250 | 28.04 | 2009 | 1674 | 1395 | 1116 | 2512 | 2651 | 2791 |
|  | 0.307 | 34.24 | 2467 | 2056 | 1713 | 1371 | 3084 | 3256 | 3427 |
|  | 0.365 | 40.48 | 2934 | 2445 | 2037 | 1630 | 3667 | 3871 | 4074 |
|  | 0.500 | 54.74 | 4019 | 3349 | 2791 | 2233 | 5023 | 5302 | 5581 |
| 12.75 | 0.250 | 33.38 | 1694 | 1412 | 1176 | 941 | 2118 | 2235 | 2353 |
|  | 0.330 | 43.77 | 2236 | 1864 | 1553 | 1242 | 2795 | 2951 | 3106 |
|  | 0.375 | 49.56 | 2541 | 2118 | 1765 | 1412 | 3176 | 3353 | 3529 |
|  | 0.406 | 53.52 | 2751 | 2293 | 1911 | 1528 | 3439 | 3630 | 3821 |
|  | 0.500 | 65.42 | 3388 | 2824 | 2353 | 1882 | 4235 | 4471 | 4706 |
| 14.00 | 0.250 | 36.71 | 1543 | 1286 | 1071 | 857 | 1929 | 2036 | 2143 |
|  | 0.312 | 45.61 | 1925 | 1605 | 1337 | 1070 | 2407 | 2541 | 2674 |
|  | 0.375 | 54.57 | 2314 | 1929 | 1607 | 1286 | 2893 | 3054 | 3214 |
|  | 0.437 | 63.30 | 2697 | 2247 | 1873 | 1498 | 3371 | 3558 | 3746 |
|  | 0.500 | 72.09 | 3086 | 2571 | 2143 | 1714 | 3857 | 4071 | 4286 |

Table 6.9 Pipeline Internal Design Pressures and Test Pressures (Continued)

| Pipe Material API 5L X60 |  | SMYS <br> Weight lb/ft | 60000 psig |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Diameter | Wall Thickness |  | Internal Design Pressure, psig |  |  |  | Hydrostatic Test Pressure, psig |  |  |
| in. | in. |  | Class 1 | Class 2 | Class 3 | Class 4 | 90\% SMYS | 95\% SMYS | 100\% SMYS |
| 16.00 | 0.250 | 42.05 | 1350 | 1125 | 938 | 750 | 1688 | 1781 | 1875 |
|  | 0.312 | 52.27 | 1685 | 1404 | 1170 | 936 | 2106 | 2223 | 2340 |
|  | 0.375 | 62.58 | 2025 | 1688 | 1406 | 1125 | 2531 | 2672 | 2813 |
|  | 0.437 | 72.64 | 2360 | 1967 | 1639 | 1311 | 2950 | 3114 | 3278 |
|  | 0.500 | 82.77 | 2700 | 2250 | 1875 | 1500 | 3375 | 3563 | 3750 |
| 18.00 | 0.250 | 47.39 | 1200 | 1000 | 833 | 667 | 1500 | 1583 | 1667 |
|  | 0.312 | 58.94 | 1498 | 1248 | 1040 | 832 | 1872 | 1976 | 2080 |
|  | 0.375 | 70.59 | 1800 | 1500 | 1250 | 1000 | 2250 | 2375 | 2500 |
|  | 0.437 | 81.97 | 2098 | 1748 | 1457 | 1165 | 2622 | 2768 | 2913 |
|  | 0.500 | 93.45 | 2400 | 2000 | 1667 | 1333 | 3000 | 3167 | 3333 |
| 20.00 | 0.312 | 65.60 | 1348 | 1123 | 936 | 749 | 1685 | 1778 | 1872 |
|  | 0.375 | 78.60 | 1620 | 1350 | 1125 | 900 | 2025 | 2138 | 2250 |
|  | 0.437 | 91.30 | 1888 | 1573 | 1311 | 1049 | 2360 | 2491 | 2622 |
|  | 0.500 | 104.13 | 2160 | 1800 | 1500 | 1200 | 2700 | 2850 | 3000 |
|  | 0.562 | 116.67 | 2428 | 2023 | 1686 | 1349 | 3035 | 3203 | 3372 |
| 22.00 | 0.375 | 86.61 | 1473 | 1227 | 1023 | 818 | 1841 | 1943 | 2045 |
|  | 0.500 | 114.81 | 1964 | 1636 | 1364 | 1091 | 2455 | 2591 | 2727 |
|  | 0.625 | 142.68 | 2455 | 2045 | 1705 | 1364 | 3068 | 3239 | 3409 |
|  | 0.750 | 170.21 | 2945 | 2455 | 2045 | 1636 | 3682 | 3886 | 4091 |
| 24.00 | 0.375 | 94.62 | 1350 | 1125 | 938 | 750 | 1688 | 1781 | 1875 |
|  | 0.437 | 109.97 | 1573 | 1311 | 1093 | 874 | 1967 | 2076 | 2185 |
|  | 0.500 | 125.49 | 1800 | 1500 | 1250 | 1000 | 2250 | 2375 | 2500 |
|  | 0.562 | 140.68 | 2023 | 1686 | 1405 | 1124 | 2529 | 2670 | 2810 |
|  | 0.625 | 156.03 | 2250 | 1875 | 1563 | 1250 | 2813 | 2969 | 3125 |
|  | 0.750 | 186.23 | 2700 | 2250 | 1875 | 1500 | 3375 | 3563 | 3750 |
| 26.00 | 0.375 | 102.63 | 1246 | 1038 | 865 | 692 | 1558 | 1644 | 1731 |
|  | 0.500 | 136.17 | 1662 | 1385 | 1154 | 923 | 2077 | 2192 | 2308 |
|  | 0.625 | 169.38 | 2077 | 1731 | 1442 | 1154 | 2596 | 2740 | 2885 |
|  | 0.750 | 202.25 | 2492 | 2077 | 1731 | 1385 | 3115 | 3288 | 3462 |


| 28.00 | 0.375 | 110.64 | 1157 | 964 | 804 | 643 | 1446 | 1527 | 1607 |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
|  | 0.500 | 146.85 | 1543 | 1286 | 1071 | 857 | 1929 | 2036 | 2143 |
|  | 0.625 | 182.73 | 1929 | 1607 | 1339 | 1071 | 2411 | 2545 | 2679 |
|  | 0.750 | 218.27 | 2314 | 1929 | 1607 | 1286 | 2893 | 3054 | 3214 |
| 30.00 | 0.375 | 118.65 | 1080 | 900 | 750 | 600 | 1350 | 1425 | 1500 |
|  | 0.500 | 157.53 | 1440 | 1200 | 1000 | 800 | 1800 | 1900 | 2000 |
|  | 0.625 | 196.08 | 1800 | 1500 | 1250 | 1000 | 2250 | 2375 | 2500 |
|  | 0.750 | 234.29 | 2160 | 1800 | 1500 | 1200 | 2700 | 2850 | 3000 |
| 32.00 | 0.375 | 126.66 | 1013 | 844 | 703 | 563 | 1266 | 1336 | 1406 |
|  | 0.500 | 168.21 | 1350 | 1125 | 938 | 750 | 1688 | 1781 | 1875 |
|  | 0.625 | 209.43 | 1688 | 1406 | 1172 | 938 | 2109 | 2227 | 2344 |
|  | 0.750 | 250.31 | 2025 | 1688 | 1406 | 1125 | 2531 | 2672 | 2813 |
| 34.00 | 0.375 | 134.67 | 953 | 794 | 662 | 529 | 1191 | 1257 | 1324 |
|  | 0.500 | 178.89 | 1271 | 1059 | 882 | 706 | 1588 | 1676 | 1765 |
|  | 0.625 | 222.78 | 1588 | 1324 | 1103 | 882 | 1985 | 2096 | 2206 |
|  | 0.750 | 266.33 | 1906 | 1588 | 1324 | 1059 | 2382 | 2515 | 2647 |
| 36.00 | 0.375 | 142.68 | 900 | 750 | 625 | 500 | 1125 | 1188 | 1250 |
|  | 0.500 | 189.57 | 1200 | 1000 | 833 | 667 | 1500 | 1583 | 1667 |
|  | 0.625 | 236.13 | 1500 | 1250 | 1042 | 833 | 1875 | 1979 | 2083 |
|  | 0.750 | 282.35 | 1800 | 1500 | 1250 | 1000 | 2250 | 2375 | 2500 |
| 42.00 | 0.375 | 166.71 | 771 | 643 | 536 | 429 | 964 | 1018 | 1071 |
|  | 0.500 | 221.61 | 1029 | 857 | 714 | 571 | 1286 | 1357 | 1429 |
|  | 0.625 | 276.18 | 1286 | 1071 | 893 | 714 | 1607 | 1696 | 1786 |
|  | 0.750 | 330.41 | 1543 | 1286 | 1071 | 857 | 1929 | 2036 | 2143 |
|  | 1.000 | 437.88 | 2057 | 1714 | 1429 | 1143 | 2571 | 2714 | 2857 |

Table 6.10 Pipeline Internal Design Pressures and Test Pressures

| Pipe Material API 5L X65 |  | SMYS <br> Weight lb/ft | 65000 psig |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Diameter | Wall Thickness |  | Internal Design Pressure, psig |  |  |  | Hydrostatic Test Pressure, psig |  |  |
| in. | in. |  | Class 1 | Class 2 | Class 3 | Class 4 | 90\% SMYS | 95\% SMYS | 100\% SMYS |
| 4.5 | 0.237 | 10.79 | 4930 | 4108 | 3423 | 2739 | 6162 | 6504 | 6847 |
|  | 0.337 | 14.98 | 7010 | 5841 | 4868 | 3894 | 8762 | 9249 | 9736 |
|  | 0.437 | 18.96 | 9090 | 7575 | 6312 | 5050 | 11362 | 11993 | 12624 |
|  | 0.531 | 22.51 | 11045 | 9204 | 7670 | 6136 | 13806 | 14573 | 15340 |
| 6.625 | 0.250 | 17.02 | 3532 | 2943 | 2453 | 1962 | 4415 | 4660 | 4906 |
|  | 0.280 | 18.97 | 3956 | 3297 | 2747 | 2198 | 4945 | 5220 | 5494 |
|  | 0.432 | 28.57 | 6103 | 5086 | 4238 | 3391 | 7629 | 8053 | 8477 |
|  | 0.562 | 36.39 | 7940 | 6617 | 5514 | 4411 | 9925 | 10477 | 11028 |
| 8.625 | 0.250 | 22.36 | 2713 | 2261 | 1884 | 1507 | 3391 | 3580 | 3768 |
|  | 0.277 | 24.70 | 3006 | 2505 | 2088 | 1670 | 3758 | 3966 | 4175 |
|  | 0.322 | 28.55 | 3494 | 2912 | 2427 | 1941 | 4368 | 4611 | 4853 |
|  | 0.406 | 35.64 | 4406 | 3672 | 3060 | 2448 | 5507 | 5813 | 6119 |
| 10.75 | 0.250 | 28.04 | 2177 | 1814 | 1512 | 1209 | 2721 | 2872 | 3023 |
|  | 0.307 | 34.24 | 2673 | 2228 | 1856 | 1485 | 3341 | 3527 | 3713 |
|  | 0.365 | 40.48 | 3178 | 2648 | 2207 | 1766 | 3973 | 4193 | 4414 |
|  | 0.500 | 54.74 | 4353 | 3628 | 3023 | 2419 | 5442 | 5744 | 6047 |
| 12.75 | 0.250 | 33.38 | 1835 | 1529 | 1275 | 1020 | 2294 | 2422 | 2549 |
|  | 0.330 | 43.77 | 2423 | 2019 | 1682 | 1346 | 3028 | 3196 | 3365 |
|  | 0.375 | 49.56 | 2753 | 2294 | 1912 | 1529 | 3441 | 3632 | 3824 |
|  | 0.406 | 53.52 | 2981 | 2484 | 2070 | 1656 | 3726 | 3933 | 4140 |
|  | 0.500 | 65.42 | 3671 | 3059 | 2549 | 2039 | 4588 | 4843 | 5098 |
| 14.00 | 0.250 | 36.71 | 1671 | 1393 | 1161 | 929 | 2089 | 2205 | 2321 |
|  | 0.312 | 45.61 | 2086 | 1738 | 1449 | 1159 | 2607 | 2752 | 2897 |
|  | 0.375 | 54.57 | 2507 | 2089 | 1741 | 1393 | 3134 | 3308 | 3482 |
|  | 0.437 | 63.30 | 2922 | 2435 | 2029 | 1623 | 3652 | 3855 | 4058 |
|  | 0.500 | 72.09 | 3343 | 2786 | 2321 | 1857 | 4179 | 4411 | 4643 |
| 16.00 | 0.250 | 42.05 | 1463 | 1219 | 1016 | 813 | 1828 | 1930 | 2031 |
|  | 0.312 | 52.27 | 1825 | 1521 | 1268 | 1014 | 2282 | 2408 | 2535 |


|  | 0.375 | 62.58 | 2194 | 1828 | 1523 | 1219 | 2742 | 2895 | 3047 |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
|  | 0.437 | 72.64 | 2556 | 2130 | 1775 | 1420 | 3196 | 3373 | 3551 |
|  | 0.500 | 82.77 | 2925 | 2438 | 2031 | 1625 | 3656 | 3859 | 4063 |
| 18.00 | 0.250 | 47.39 | 1300 | 1083 | 903 | 722 | 1625 | 1715 | 1806 |
|  | 0.312 | 58.94 | 1622 | 1352 | 1127 | 901 | 2028 | 2141 | 2253 |
|  | 0.375 | 70.59 | 1950 | 1625 | 1354 | 1083 | 2438 | 2573 | 2708 |
|  | 0.437 | 81.97 | 2272 | 1894 | 1578 | 1262 | 2841 | 2998 | 3156 |
|  | 0.500 | 93.45 | 2600 | 2167 | 1806 | 1444 | 3250 | 3431 | 3611 |
| 20.00 | 0.312 | 65.60 | 1460 | 1217 | 1014 | 811 | 1825 | 1927 | 2028 |
|  | 0.375 | 78.60 | 1755 | 1463 | 1219 | 975 | 2194 | 2316 | 2438 |
|  | 0.437 | 91.30 | 2045 | 1704 | 1420 | 1136 | 2556 | 2698 | 2841 |
|  | 0.500 | 104.13 | 2340 | 1950 | 1625 | 1300 | 2925 | 3088 | 3250 |
|  | 0.562 | 116.67 | 2630 | 2192 | 1827 | 1461 | 3288 | 3470 | 3653 |
| 22.00 | 0.375 | 86.61 | 1595 | 1330 | 1108 | 886 | 1994 | 2105 | 2216 |
|  | 0.500 | 114.81 | 2127 | 1773 | 1477 | 1182 | 2659 | 2807 | 2955 |
|  | 0.625 | 142.68 | 2659 | 2216 | 1847 | 1477 | 3324 | 3509 | 3693 |
|  | 0.750 | 170.21 | 3191 | 2659 | 2216 | 1773 | 3989 | 4210 | 4432 |
| 24.00 | 0.375 | 94.62 | 1463 | 1219 | 1016 | 813 | 1828 | 1930 | 2031 |
|  | 0.437 | 109.97 | 1704 | 1420 | 1184 | 947 | 2130 | 2249 | 2367 |
|  | 0.500 | 125.49 | 1950 | 1625 | 1354 | 1083 | 2438 | 2573 | 2708 |
|  | 0.562 | 140.68 | 2192 | 1827 | 1522 | 1218 | 2740 | 2892 | 3044 |
|  | 0.625 | 156.03 | 2438 | 2031 | 1693 | 1354 | 3047 | 3216 | 3385 |
|  | 0.750 | 186.23 | 2925 | 2438 | 2031 | 1625 | 3656 | 3859 | 4063 |
| 26.00 | 0.375 | 102.63 | 1350 | 1125 | 938 | 750 | 1688 | 1781 | 1875 |
|  | 0.500 | 136.17 | 1800 | 1500 | 1250 | 1000 | 2250 | 2375 | 2500 |
|  | 0.625 | 169.38 | 2250 | 1875 | 1563 | 1250 | 2813 | 2969 | 3125 |
|  | 0.750 | 202.25 | 2700 | 2250 | 1875 | 1500 | 3375 | 3563 | 3750 |
| 28.00 | 0.375 | 110.64 | 1254 | 1045 | 871 | 696 | 1567 | 1654 | 1741 |
|  | 0.500 | 146.85 | 1671 | 1393 | 1161 | 929 | 2089 | 2205 | 2321 |
|  | 0.625 | 182.73 | 2089 | 1741 | 1451 | 1161 | 2612 | 2757 | 2902 |
|  | 0.750 | 218.27 | 2507 | 2089 | 1741 | 1393 | 3134 | 3308 | 3482 |

Table 6.10 Pipeline Internal Design Pressures and Test Pressures (Continued)

| Pipe Material API 5L X65 |  | SMYS <br> Weight lb/ft | 65000 psig |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Diameter | Wall Thickness |  | Internal Design Pressure, psig |  |  |  | Hydrostatic Test Pressure, psig |  |  |
| in. | in. |  | Class 1 | Class 2 | Class 3 | Class 4 | 90\% SMYS | 95\% SMYS | 100\% SMYS |
| 30.00 | 0.375 | 118.65 | 1170 | 975 | 813 | 650 | 1463 | 1544 | 1625 |
|  | 0.500 | 157.53 | 1560 | 1300 | 1083 | 867 | 1950 | 2058 | 2167 |
|  | 0.625 | 196.08 | 1950 | 1625 | 1354 | 1083 | 2438 | 2573 | 2708 |
|  | 0.750 | 234.29 | 2340 | 1950 | 1625 | 1300 | 2925 | 3088 | 3250 |
| 32.00 | 0.375 | 126.66 | 1097 | 914 | 762 | 609 | 1371 | 1447 | 1523 |
|  | 0.500 | 168.21 | 1463 | 1219 | 1016 | 813 | 1828 | 1930 | 2031 |
|  | 0.625 | 209.43 | 1828 | 1523 | 1270 | 1016 | 2285 | 2412 | 2539 |
|  | 0.750 | 250.31 | 2194 | 1828 | 1523 | 1219 | 2742 | 2895 | 3047 |
| 34.00 | 0.375 | 134.67 | 1032 | 860 | 717 | 574 | 1290 | 1362 | 1434 |
|  | 0.500 | 178.89 | 1376 | 1147 | 956 | 765 | 1721 | 1816 | 1912 |
|  | 0.625 | 222.78 | 1721 | 1434 | 1195 | 956 | 2151 | 2270 | 2390 |
|  | 0.750 | 266.33 | 2065 | 1721 | 1434 | 1147 | 2581 | 2724 | 2868 |
| 36.00 | 0.375 | 142.68 | 975 | 813 | 677 | 542 | 1219 | 1286 | 1354 |
|  | 0.500 | 189.57 | 1300 | 1083 | 903 | 722 | 1625 | 1715 | 1806 |
|  | 0.625 | 236.13 | 1625 | 1354 | 1128 | 903 | 2031 | 2144 | 2257 |
|  | 0.750 | 282.35 | 1950 | 1625 | 1354 | 1083 | 2438 | 2573 | 2708 |
| 42.00 | 0.375 | 166.71 | 836 | 696 | 580 | 464 | 1045 | 1103 | 1161 |
|  | 0.500 | 221.61 | 1114 | 929 | 774 | 619 | 1393 | 1470 | 1548 |
|  | 0.625 | 276.18 | 1393 | 1161 | 967 | 774 | 1741 | 1838 | 1935 |
|  | 0.750 | 330.41 | 1671 | 1393 | 1161 | 929 | 2089 | 2205 | 2321 |
|  | 1.000 | 437.88 | 2229 | 1857 | 1548 | 1238 | 2786 | 2940 | 3095 |

Table 6.11 Pipeline Internal Design Pressures and Test Pressures

| Pipe Material API 5L X70 |  | SMYS <br> Weight lb/ft | 70000 psig |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Diameter | Wall Thickness |  | Internal Design Pressure, psig |  |  |  | Hydrostatic Test Pressure, psig |  |  |
| in. | in. |  | Class 1 | Class 2 | Class 3 | Class 4 | 90\% SMYS | 95\% SMYS | 100\% SMYS |
| 4.5 | 0.237 | 10.79 | 5309 | 4424 | 3687 | 2949 | 6636 | 7005 | 7373 |
|  | 0.337 | 14.98 | 7549 | 6291 | 5242 | 4194 | 9436 | 9960 | 10484 |
|  | 0.437 | 18.96 | 9789 | 8157 | 6798 | 5438 | 12236 | 12916 | 13596 |
|  | 0.531 | 22.51 | 11894 | 9912 | 8260 | 6608 | 14868 | 15694 | 16520 |
| 6.625 | 0.250 | 17.02 | 3804 | 3170 | 2642 | 2113 | 4755 | 5019 | 5283 |
|  | 0.280 | 18.97 | 4260 | 3550 | 2958 | 2367 | 5325 | 5621 | 5917 |
|  | 0.432 | 28.57 | 6573 | 5477 | 4565 | 3652 | 8216 | 8673 | 9129 |
|  | 0.562 | 36.39 | 8551 | 7126 | 5938 | 4750 | 10689 | 11282 | 11876 |
| 8.625 | 0.250 | 22.36 | 2922 | 2435 | 2029 | 1623 | 3652 | 3855 | 4058 |
|  | 0.277 | 24.70 | 3237 | 2698 | 2248 | 1798 | 4047 | 4271 | 4496 |
|  | 0.322 | 28.55 | 3763 | 3136 | 2613 | 2091 | 4704 | 4965 | 5227 |
|  | 0.406 | 35.64 | 4745 | 3954 | 3295 | 2636 | 5931 | 6261 | 6590 |
| 10.75 | 0.250 | 28.04 | 2344 | 1953 | 1628 | 1302 | 2930 | 3093 | 3256 |
|  | 0.307 | 34.24 | 2879 | 2399 | 1999 | 1599 | 3598 | 3798 | 3998 |
|  | 0.365 | 40.48 | 3423 | 2852 | 2377 | 1901 | 4278 | 4516 | 4753 |
|  | 0.500 | 54.74 | 4688 | 3907 | 3256 | 2605 | 5860 | 6186 | 6512 |
| 12.75 | 0.250 | 33.38 | 1976 | 1647 | 1373 | 1098 | 2471 | 2608 | 2745 |
|  | 0.330 | 43.77 | 2609 | 2174 | 1812 | 1449 | 3261 | 3442 | 3624 |
|  | 0.375 | 49.56 | 2965 | 2471 | 2059 | 1647 | 3706 | 3912 | 4118 |
|  | 0.406 | 53.52 | 3210 | 2675 | 2229 | 1783 | 4012 | 4235 | 4458 |
|  | 0.500 | 65.42 | 3953 | 3294 | 2745 | 2196 | 4941 | 5216 | 5490 |
| 14.00 | 0.250 | 36.71 | 1800 | 1500 | 1250 | 1000 | 2250 | 2375 | 2500 |
|  | 0.312 | 45.61 | 2246 | 1872 | 1560 | 1248 | 2808 | 2964 | 3120 |
|  | 0.375 | 54.57 | 2700 | 2250 | 1875 | 1500 | 3375 | 3563 | 3750 |
|  | 0.437 | 63.30 | 3146 | 2622 | 2185 | 1748 | 3933 | 4152 | 4370 |
|  | 0.500 | 72.09 | 3600 | 3000 | 2500 | 2000 | 4500 | 4750 | 5000 |

Table 6.11 Pipeline Internal Design Pressures and Test Pressures (Continued)

| Pipe Material API 5L X70 |  | SMYS <br> Weight lb/ft | 70000 psig |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Diameter | Wall Thickness |  | Internal Design Pressure, psig |  |  |  | Hydrostatic Test Pressure, psig |  |  |
| in. | in. |  | Class 1 | Class 2 | Class 3 | Class 4 | 90\% SMYS | 95\% SMYS | 100\% SMYS |
| 16.00 | 0.250 | 42.05 | 1575 | 1313 | 1094 | 875 | 1969 | 2078 | 2188 |
|  | 0.312 | 52.27 | 1966 | 1638 | 1365 | 1092 | 2457 | 2594 | 2730 |
|  | 0.375 | 62.58 | 2363 | 1969 | 1641 | 1313 | 2953 | 3117 | 3281 |
|  | 0.437 | 72.64 | 2753 | 2294 | 1912 | 1530 | 3441 | 3633 | 3824 |
|  | 0.500 | 82.77 | 3150 | 2625 | 2188 | 1750 | 3938 | 4156 | 4375 |
| 18.00 | 0.250 | 47.39 | 1400 | 1167 | 972 | 778 | 1750 | 1847 | 1944 |
|  | 0.312 | 58.94 | 1747 | 1456 | 1213 | 971 | 2184 | 2305 | 2427 |
|  | 0.375 | 70.59 | 2100 | 1750 | 1458 | 1167 | 2625 | 2771 | 2917 |
|  | 0.437 | 81.97 | 2447 | 2039 | 1699 | 1360 | 3059 | 3229 | 3399 |
|  | 0.500 | 93.45 | 2800 | 2333 | 1944 | 1556 | 3500 | 3694 | 3889 |
| 20.00 | 0.312 | 65.60 | 1572 | 1310 | 1092 | 874 | 1966 | 2075 | 2184 |
|  | 0.375 | 78.60 | 1890 | 1575 | 1313 | 1050 | 2363 | 2494 | 2625 |
|  | 0.437 | 91.30 | 2202 | 1835 | 1530 | 1224 | 2753 | 2906 | 3059 |
|  | 0.500 | 104.13 | 2520 | 2100 | 1750 | 1400 | 3150 | 3325 | 3500 |
|  | 0.562 | 116.67 | 2832 | 2360 | 1967 | 1574 | 3541 | 3737 | 3934 |
| 22.00 | 0.375 | 86.61 | 1718 | 1432 | 1193 | 955 | 2148 | 2267 | 2386 |
|  | 0.500 | 114.81 | 2291 | 1909 | 1591 | 1273 | 2864 | 3023 | 3182 |
|  | 0.625 | 142.68 | 2864 | 2386 | 1989 | 1591 | 3580 | 3778 | 3977 |
|  | 0.750 | 170.21 | 3436 | 2864 | 2386 | 1909 | 4295 | 4534 | 4773 |
| 24.00 | 0.375 | 94.62 | 1575 | 1313 | 1094 | 875 | 1969 | 2078 | 2188 |
|  | 0.437 | 109.97 | 1835 | 1530 | 1275 | 1020 | 2294 | 2422 | 2549 |
|  | 0.500 | 125.49 | 2100 | 1750 | 1458 | 1167 | 2625 | 2771 | 2917 |
|  | 0.562 | 140.68 | 2360 | 1967 | 1639 | 1311 | 2951 | 3114 | 3278 |
|  | 0.625 | 156.03 | 2625 | 2188 | 1823 | 1458 | 3281 | 3464 | 3646 |
|  | 0.750 | 186.23 | 3150 | 2625 | 2188 | 1750 | 3938 | 4156 | 4375 |
| 26.00 | 0.375 | 102.63 | 1454 | 1212 | 1010 | 808 | 1817 | 1918 | 2019 |
|  | 0.500 | 136.17 | 1938 | 1615 | 1346 | 1077 | 2423 | 2558 | 2692 |
|  | 0.625 | 169.38 | 2423 | 2019 | 1683 | 1346 | 3029 | 3197 | 3365 |
|  | 0.750 | 202.25 | 2908 | 2423 | 2019 | 1615 | 3635 | 3837 | 4038 |


| 28.00 | 0.375 | 110.64 | 1350 | 1125 | 938 | 750 | 1688 | 1781 | 1875 |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
|  | 0.500 | 146.85 | 1800 | 1500 | 1250 | 1000 | 2250 | 2375 | 2500 |
|  | 0.625 | 182.73 | 2250 | 1875 | 1563 | 1250 | 2813 | 2969 | 3125 |
|  | 0.750 | 218.27 | 2700 | 2250 | 1875 | 1500 | 3375 | 3563 | 3750 |
| 30.00 | 0.375 | 118.65 | 1260 | 1050 | 875 | 700 | 1575 | 1663 | 1750 |
|  | 0.500 | 157.53 | 1680 | 1400 | 1167 | 933 | 2100 | 2217 | 2333 |
|  | 0.625 | 196.08 | 2100 | 1750 | 1458 | 1167 | 2625 | 2771 | 2917 |
|  | 0.750 | 234.29 | 2520 | 2100 | 1750 | 1400 | 3150 | 3325 | 3500 |
| 32.00 | 0.375 | 126.66 | 1181 | 984 | 820 | 656 | 1477 | 1559 | 1641 |
|  | 0.500 | 168.21 | 1575 | 1313 | 1094 | 875 | 1969 | 2078 | 2188 |
|  | 0.625 | 209.43 | 1969 | 1641 | 1367 | 1094 | 2461 | 2598 | 2734 |
|  | 0.750 | 250.31 | 2363 | 1969 | 1641 | 1313 | 2953 | 3117 | 3281 |
| 34.00 | 0.375 | 134.67 | 1112 | 926 | 772 | 618 | 1390 | 1467 | 1544 |
|  | 0.500 | 178.89 | 1482 | 1235 | 1029 | 824 | 1853 | 1956 | 2059 |
|  | 0.625 | 222.78 | 1853 | 1544 | 1287 | 1029 | 2316 | 2445 | 2574 |
|  | 0.750 | 266.33 | 2224 | 1853 | 1544 | 1235 | 2779 | 2934 | 3088 |
| 36.00 | 0.375 | 142.68 | 1050 | 875 | 729 | 583 | 1313 | 1385 | 1458 |
|  | 0.500 | 189.57 | 1400 | 1167 | 972 | 778 | 1750 | 1847 | 1944 |
|  | 0.625 | 236.13 | 1750 | 1458 | 1215 | 972 | 2188 | 2309 | 2431 |
|  | 0.750 | 282.35 | 2100 | 1750 | 1458 | 1167 | 2625 | 2771 | 2917 |
| 42.00 | 0.375 | 166.71 | 900 | 750 | 625 | 500 | 1125 | 1188 | 1250 |
|  | 0.500 | 221.61 | 1200 | 1000 | 833 | 667 | 1500 | 1583 | 1667 |
|  | 0.625 | 276.18 | 1500 | 1250 | 1042 | 833 | 1875 | 1979 | 2083 |
|  | 0.750 | 330.41 | 1800 | 1500 | 1250 | 1000 | 2250 | 2375 | 2500 |
|  | 1.000 | 437.88 | 2400 | 2000 | 1667 | 1333 | 3000 | 3167 | 3333 |

Table 6.12 Pipeline Internal Design Pressures and Test Pressures

| Pipe Material API 5L X80 |  | SMYS <br> Weight lb/ft | 80000 psig |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Diameter | Wall Thickness |  | Internal Design Pressure, psig |  |  |  | Hydrostatic Test Pressure, psig |  |  |
| in. | in. |  | Class 1 | Class 2 | Class 3 | Class 4 | 90\% SMYS | 95\% SMYS | 100\% SMYS |
| 4.5 | 0.237 | 10.79 | 6067 | 5056 | 4213 | 3371 | 7584 | 8005 | 8427 |
|  | 0.337 | 14.98 | 8627 | 7189 | 5991 | 4793 | 10784 | 11383 | 11982 |
|  | 0.437 | 18.96 | 11187 | 9323 | 7769 | 6215 | 13984 | 14761 | 15538 |
|  | 0.531 | 22.51 | 13594 | 11328 | 9440 | 7552 | 16992 | 17936 | 18880 |
| 6.625 | 0.250 | 17.02 | 4347 | 3623 | 3019 | 2415 | 5434 | 5736 | 6038 |
|  | 0.280 | 18.97 | 4869 | 4057 | 3381 | 2705 | 6086 | 6424 | 6762 |
|  | 0.432 | 28.57 | 7512 | 6260 | 5217 | 4173 | 9390 | 9912 | 10433 |
|  | 0.562 | 36.39 | 9772 | 8144 | 6786 | 5429 | 12216 | 12894 | 13573 |
| 8.625 | 0.250 | 22.36 | 3339 | 2783 | 2319 | 1855 | 4174 | 4406 | 4638 |
|  | 0.277 | 24.70 | 3700 | 3083 | 2569 | 2055 | 4625 | 4882 | 5139 |
|  | 0.322 | 28.55 | 4301 | 3584 | 2987 | 2389 | 5376 | 5675 | 5973 |
|  | 0.406 | 35.64 | 5423 | 4519 | 3766 | 3013 | 6778 | 7155 | 7532 |
| 10.75 | 0.250 | 28.04 | 2679 | 2233 | 1860 | 1488 | 3349 | 3535 | 3721 |
|  | 0.307 | 34.24 | 3290 | 2742 | 2285 | 1828 | 4112 | 4341 | 4569 |
|  | 0.365 | 40.48 | 3911 | 3260 | 2716 | 2173 | 4889 | 5161 | 5433 |
|  | 0.500 | 54.74 | 5358 | 4465 | 3721 | 2977 | 6698 | 7070 | 7442 |
| 12.75 | 0.250 | 33.38 | 2259 | 1882 | 1569 | 1255 | 2824 | 2980 | 3137 |
|  | 0.330 | 43.77 | 2982 | 2485 | 2071 | 1656 | 3727 | 3934 | 4141 |
|  | 0.375 | 49.56 | 3388 | 2824 | 2353 | 1882 | 4235 | 4471 | 4706 |
|  | 0.406 | 53.52 | 3668 | 3057 | 2547 | 2038 | 4585 | 4840 | 5095 |
|  | 0.500 | 65.42 | 4518 | 3765 | 3137 | 2510 | 5647 | 5961 | 6275 |
| 14.00 | 0.250 | 36.71 | 2057 | 1714 | 1429 | 1143 | 2571 | 2714 | 2857 |
|  | 0.312 | 45.61 | 2567 | 2139 | 1783 | 1426 | 3209 | 3387 | 3566 |
|  | 0.375 | 54.57 | 3086 | 2571 | 2143 | 1714 | 3857 | 4071 | 4286 |
|  | 0.437 | 63.30 | 3596 | 2997 | 2497 | 1998 | 4495 | 4745 | 4994 |
|  | 0.500 | 72.09 | 4114 | 3429 | 2857 | 2286 | 5143 | 5429 | 5714 |
| 16.00 | 0.250 | 42.05 | 1800 | 1500 | 1250 | 1000 | 2250 | 2375 | 2500 |
|  | 0.312 | 52.27 | 2246 | 1872 | 1560 | 1248 | 2808 | 2964 | 3120 |


|  | 0.375 | 62.58 | 2700 | 2250 | 1875 | 1500 | 3375 | 3563 | 3750 |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
|  | 0.437 | 72.64 | 3146 | 2622 | 2185 | 1748 | 3933 | 4152 | 4370 |
|  | 0.500 | 82.77 | 3600 | 3000 | 2500 | 2000 | 4500 | 4750 | 5000 |
| 18.00 | 0.250 | 47.39 | 1600 | 1333 | 1111 | 889 | 2000 | 2111 | 2222 |
|  | 0.312 | 58.94 | 1997 | 1664 | 1387 | 1109 | 2496 | 2635 | 2773 |
|  | 0.375 | 70.59 | 2400 | 2000 | 1667 | 1333 | 3000 | 3167 | 3333 |
|  | 0.437 | 81.97 | 2797 | 2331 | 1942 | 1554 | 3496 | 3690 | 3884 |
|  | 0.500 | 93.45 | 3200 | 2667 | 2222 | 1778 | 4000 | 4222 | 4444 |
| 20.00 | 0.312 | 65.60 | 1797 | 1498 | 1248 | 998 | 2246 | 2371 | 2496 |
|  | 0.375 | 78.60 | 2160 | 1800 | 1500 | 1200 | 2700 | 2850 | 3000 |
|  | 0.437 | 91.30 | 2517 | 2098 | 1748 | 1398 | 3146 | 3321 | 3496 |
|  | 0.500 | 104.13 | 2880 | 2400 | 2000 | 1600 | 3600 | 3800 | 4000 |
|  | 0.562 | 116.67 | 3237 | 2698 | 2248 | 1798 | 4046 | 4271 | 4496 |
| 22.00 | 0.375 | 86.61 | 1964 | 1636 | 1364 | 1091 | 2455 | 2591 | 2727 |
|  | 0.500 | 114.81 | 2618 | 2182 | 1818 | 1455 | 3273 | 3455 | 3636 |
|  | 0.625 | 142.68 | 3273 | 2727 | 2273 | 1818 | 4091 | 4318 | 4545 |
|  | 0.750 | 170.21 | 3927 | 3273 | 2727 | 2182 | 4909 | 5182 | 5455 |
| 24.00 | 0.375 | 94.62 | 1800 | 1500 | 1250 | 1000 | 2250 | 2375 | 2500 |
|  | 0.437 | 109.97 | 2098 | 1748 | 1457 | 1165 | 2622 | 2768 | 2913 |
|  | 0.500 | 125.49 | 2400 | 2000 | 1667 | 1333 | 3000 | 3167 | 3333 |
|  | 0.562 | 140.68 | 2698 | 2248 | 1873 | 1499 | 3372 | 3559 | 3747 |
|  | 0.625 | 156.03 | 3000 | 2500 | 2083 | 1667 | 3750 | 3958 | 4167 |
|  | 0.750 | 186.23 | 3600 | 3000 | 2500 | 2000 | 4500 | 4750 | 5000 |
| 26.00 | 0.375 | 102.63 | 1662 | 1385 | 1154 | 923 | 2077 | 2192 | 2308 |
|  | 0.500 | 136.17 | 2215 | 1846 | 1538 | 1231 | 2769 | 2923 | 3077 |
|  | 0.625 | 169.38 | 2769 | 2308 | 1923 | 1538 | 3462 | 3654 | 3846 |
|  | 0.750 | 202.25 | 3323 | 2769 | 2308 | 1846 | 4154 | 4385 | 4615 |
| 28.00 | 0.375 | 110.64 | 1543 | 1286 | 1071 | 857 | 1929 | 2036 | 2143 |
|  | 0.500 | 146.85 | 2057 | 1714 | 1429 | 1143 | 2571 | 2714 | 2857 |
|  | 0.625 | 182.73 | 2571 | 2143 | 1786 | 1429 | 3214 | 3393 | 3571 |
|  | 0.750 | 218.27 | 3086 | 2571 | 2143 | 1714 | 3857 | 4071 | 4286 |

Table 6.12 Pipeline Internal Design Pressures and Test Pressures (Continued)

| Pipe Material API 5L X80 |  | SMYS <br> Weight lb/ft | 80000 psig |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Diameter | Wall Thickness |  | Internal Design Pressure, psig |  |  |  | Hydrostatic Test Pressure, psig |  |  |
| in. | in. |  | Class 1 | Class 2 | Class 3 | Class 4 | 90\% SMYS | 95\% SMYS | 100\% SMYS |
| 30.00 | 0.375 | 118.65 | 1440 | 1200 | 1000 | 800 | 1800 | 1900 | 2000 |
|  | 0.500 | 157.53 | 1920 | 1600 | 1333 | 1067 | 2400 | 2533 | 2667 |
|  | 0.625 | 196.08 | 2400 | 2000 | 1667 | 1333 | 3000 | 3167 | 3333 |
|  | 0.750 | 234.29 | 2880 | 2400 | 2000 | 1600 | 3600 | 3800 | 4000 |
| 32.00 | 0.375 | 126.66 | 1350 | 1125 | 938 | 750 | 1688 | 1781 | 1875 |
|  | 0.500 | 168.21 | 1800 | 1500 | 1250 | 1000 | 2250 | 2375 | 2500 |
|  | 0.625 | 209.43 | 2250 | 1875 | 1563 | 1250 | 2813 | 2969 | 3125 |
|  | 0.750 | 250.31 | 2700 | 2250 | 1875 | 1500 | 3375 | 3563 | 3750 |
| 34.00 | 0.375 | 134.67 | 1271 | 1059 | 882 | 706 | 1588 | 1676 | 1765 |
|  | 0.500 | 178.89 | 1694 | 1412 | 1176 | 941 | 2118 | 2235 | 2353 |
|  | 0.625 | 222.78 | 2118 | 1765 | 1471 | 1176 | 2647 | 2794 | 2941 |
|  | 0.750 | 266.33 | 2541 | 2118 | 1765 | 1412 | 3176 | 3353 | 3529 |
| 36.00 | 0.375 | 142.68 | 1200 | 1000 | 833 | 667 | 1500 | 1583 | 1667 |
|  | 0.500 | 189.57 | 1600 | 1333 | 1111 | 889 | 2000 | 2111 | 2222 |
|  | 0.625 | 236.13 | 2000 | 1667 | 1389 | 1111 | 2500 | 2639 | 2778 |
|  | 0.750 | 282.35 | 2400 | 2000 | 1667 | 1333 | 3000 | 3167 | 3333 |
| 42.00 | 0.375 | 166.71 | 1029 | 857 | 714 | 571 | 1286 | 1357 | 1429 |
|  | 0.500 | 221.61 | 1371 | 1143 | 952 | 762 | 1714 | 1810 | 1905 |
|  | 0.625 | 276.18 | 1714 | 1429 | 1190 | 952 | 2143 | 2262 | 2381 |
|  | 0.750 | 330.41 | 2057 | 1714 | 1429 | 1143 | 2571 | 2714 | 2857 |
|  | 1.000 | 437.88 | 2743 | 2286 | 1905 | 1524 | 3429 | 3619 | 3810 |

Table 6.13 Pipeline Internal Design Pressures and Test Pressures

| Pipe Material API 5L X90 |  | SMYS <br> Weight lb/ft | 90000 psig |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Diameter | Wall Thickness |  | Internal Design Pressure, psig |  |  |  | Hydrostatic Test Pressure, psig |  |  |
| in. | in. |  | Class 1 | Class 2 | Class 3 | Class 4 | 90\% SMYS | 95\% SMYS | 100\% SMYS |
| 4.5 | 0.237 | 10.79 | 6826 | 5688 | 4740 | 3792 | 8532 | 9006 | 9480 |
|  | 0.337 | 14.98 | 9706 | 8088 | 6740 | 5392 | 12132 | 12806 | 13480 |
|  | 0.437 | 18.96 | 12586 | 10488 | 8740 | 6992 | 15732 | 16606 | 17480 |
|  | 0.531 | 22.51 | 15293 | 12744 | 10620 | 8496 | 19116 | 20178 | 21240 |
| 6.625 | 0.250 | 17.02 | 4891 | 4075 | 3396 | 2717 | 6113 | 6453 | 6792 |
|  | 0.280 | 18.97 | 5477 | 4565 | 3804 | 3043 | 6847 | 7227 | 7608 |
|  | 0.432 | 28.57 | 8451 | 7042 | 5869 | 4695 | 10564 | 11150 | 11737 |
|  | 0.562 | 36.39 | 10994 | 9162 | 7635 | 6108 | 13742 | 14506 | 15269 |
| 8.625 | 0.250 | 22.36 | 3757 | 3130 | 2609 | 2087 | 4696 | 4957 | 5217 |
|  | 0.277 | 24.70 | 4162 | 3469 | 2890 | 2312 | 5203 | 5492 | 5781 |
|  | 0.322 | 28.55 | 4838 | 4032 | 3360 | 2688 | 6048 | 6384 | 6720 |
|  | 0.406 | 35.64 | 6101 | 5084 | 4237 | 3389 | 7626 | 8049 | 8473 |
| 10.75 | 0.250 | 28.04 | 3014 | 2512 | 2093 | 1674 | 3767 | 3977 | 4186 |
|  | 0.307 | 34.24 | 3701 | 3084 | 2570 | 2056 | 4626 | 4883 | 5140 |
|  | 0.365 | 40.48 | 4400 | 3667 | 3056 | 2445 | 5500 | 5806 | 6112 |
|  | 0.500 | 54.74 | 6028 | 5023 | 4186 | 3349 | 7535 | 7953 | 8372 |
| 12.75 | 0.250 | 33.38 | 2541 | 2118 | 1765 | 1412 | 3176 | 3353 | 3529 |
|  | 0.330 | 43.77 | 3354 | 2795 | 2329 | 1864 | 4193 | 4426 | 4659 |
|  | 0.375 | 49.56 | 3812 | 3176 | 2647 | 2118 | 4765 | 5029 | 5294 |
|  | 0.406 | 53.52 | 4127 | 3439 | 2866 | 2293 | 5159 | 5445 | 5732 |
|  | 0.500 | 65.42 | 5082 | 4235 | 3529 | 2824 | 6353 | 6706 | 7059 |
| 14.00 | 0.250 | 36.71 | 2314 | 1929 | 1607 | 1286 | 2893 | 3054 | 3214 |
|  | 0.312 | 45.61 | 2888 | 2407 | 2006 | 1605 | 3610 | 3811 | 4011 |
|  | 0.375 | 54.57 | 3471 | 2893 | 2411 | 1929 | 4339 | 4580 | 4821 |
|  | 0.437 | 63.30 | 4045 | 3371 | 2809 | 2247 | 5057 | 5338 | 5619 |
|  | 0.500 | 72.09 | 4629 | 3857 | 3214 | 2571 | 5786 | 6107 | 6429 |

Table 6.13 Pipeline Internal Design Pressures and Test Pressures (Continued)

| Pipe Material API 5L X90 |  | SMYS <br> Weight lb/ft | 90000 psig |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Diameter | Wall Thickness |  | Internal Design Pressure, psig |  |  |  | Hydrostatic Test Pressure, psig |  |  |
| in. | in. |  | Class 1 | Class 2 | Class 3 | Class 4 | 90\% SMYS | 95\% SMYS | 100\% SMYS |
| 16.00 | 0.250 | 42.05 | 2025 | 1688 | 1406 | 1125 | 2531 | 2672 | 2813 |
|  | 0.312 | 52.27 | 2527 | 2106 | 1755 | 1404 | 3159 | 3335 | 3510 |
|  | 0.375 | 62.58 | 3038 | 2531 | 2109 | 1688 | 3797 | 4008 | 4219 |
|  | 0.437 | 72.64 | 3540 | 2950 | 2458 | 1967 | 4425 | 4670 | 4916 |
|  | 0.500 | 82.77 | 4050 | 3375 | 2813 | 2250 | 5063 | 5344 | 5625 |
| 18.00 | 0.250 | 47.39 | 1800 | 1500 | 1250 | 1000 | 2250 | 2375 | 2500 |
|  | 0.312 | 58.94 | 2246 | 1872 | 1560 | 1248 | 2808 | 2964 | 3120 |
|  | 0.375 | 70.59 | 2700 | 2250 | 1875 | 1500 | 3375 | 3563 | 3750 |
|  | 0.437 | 81.97 | 3146 | 2622 | 2185 | 1748 | 3933 | 4152 | 4370 |
|  | 0.500 | 93.45 | 3600 | 3000 | 2500 | 2000 | 4500 | 4750 | 5000 |
| 20.00 | 0.312 | 65.60 | 2022 | 1685 | 1404 | 1123 | 2527 | 2668 | 2808 |
|  | 0.375 | 78.60 | 2430 | 2025 | 1688 | 1350 | 3038 | 3206 | 3375 |
|  | 0.437 | 91.30 | 2832 | 2360 | 1967 | 1573 | 3540 | 3736 | 3933 |
|  | 0.500 | 104.13 | 3240 | 2700 | 2250 | 1800 | 4050 | 4275 | 4500 |
|  | 0.562 | 116.67 | 3642 | 3035 | 2529 | 2023 | 4552 | 4805 | 5058 |
| 22.00 | 0.375 | 86.61 | 2209 | 1841 | 1534 | 1227 | 2761 | 2915 | 3068 |
|  | 0.500 | 114.81 | 2945 | 2455 | 2045 | 1636 | 3682 | 3886 | 4091 |
|  | 0.625 | 142.68 | 3682 | 3068 | 2557 | 2045 | 4602 | 4858 | 5114 |
|  | 0.750 | 170.21 | 4418 | 3682 | 3068 | 2455 | 5523 | 5830 | 6136 |
| 24.00 | 0.375 | 94.62 | 2025 | 1688 | 1406 | 1125 | 2531 | 2672 | 2813 |
|  | 0.437 | 109.97 | 2360 | 1967 | 1639 | 1311 | 2950 | 3114 | 3278 |
|  | 0.500 | 125.49 | 2700 | 2250 | 1875 | 1500 | 3375 | 3563 | 3750 |
|  | 0.562 | 140.68 | 3035 | 2529 | 2108 | 1686 | 3794 | 4004 | 4215 |
|  | 0.625 | 156.03 | 3375 | 2813 | 2344 | 1875 | 4219 | 4453 | 4688 |
|  | 0.750 | 186.23 | 4050 | 3375 | 2813 | 2250 | 5063 | 5344 | 5625 |
| 26.00 | 0.375 | 102.63 | 1869 | 1558 | 1298 | 1038 | 2337 | 2466 | 2596 |
|  | 0.500 | 136.17 | 2492 | 2077 | 1731 | 1385 | 3115 | 3288 | 3462 |
|  | 0.625 | 169.38 | 3115 | 2596 | 2163 | 1731 | 3894 | 4111 | 4327 |
|  | 0.750 | 202.25 | 3738 | 3115 | 2596 | 2077 | 4673 | 4933 | 5192 |


| 28.00 | 0.375 | 110.64 | 1736 | 1446 | 1205 | 964 | 2170 | 2290 | 2411 |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
|  | 0.500 | 146.85 | 2314 | 1929 | 1607 | 1286 | 2893 | 3054 | 3214 |
|  | 0.625 | 182.73 | 2893 | 2411 | 2009 | 1607 | 3616 | 3817 | 4018 |
|  | 0.750 | 218.27 | 3471 | 2893 | 2411 | 1929 | 4339 | 4580 | 4821 |
| 30.00 | 0.375 | 118.65 | 1620 | 1350 | 1125 | 900 | 2025 | 2138 | 2250 |
|  | 0.500 | 157.53 | 2160 | 1800 | 1500 | 1200 | 2700 | 2850 | 3000 |
|  | 0.625 | 196.08 | 2700 | 2250 | 1875 | 1500 | 3375 | 3563 | 3750 |
|  | 0.750 | 234.29 | 3240 | 2700 | 2250 | 1800 | 4050 | 4275 | 4500 |
| 32.00 | 0.375 | 126.66 | 1519 | 1266 | 1055 | 844 | 1898 | 2004 | 2109 |
|  | 0.500 | 168.21 | 2025 | 1688 | 1406 | 1125 | 2531 | 2672 | 2813 |
|  | 0.625 | 209.43 | 2531 | 2109 | 1758 | 1406 | 3164 | 3340 | 3516 |
|  | 0.750 | 250.31 | 3038 | 2531 | 2109 | 1688 | 3797 | 4008 | 4219 |
| 34.00 | 0.375 | 134.67 | 1429 | 1191 | 993 | 794 | 1787 | 1886 | 1985 |
|  | 0.500 | 178.89 | 1906 | 1588 | 1324 | 1059 | 2382 | 2515 | 2647 |
|  | 0.625 | 222.78 | 2382 | 1985 | 1654 | 1324 | 2978 | 3143 | 3309 |
|  | 0.750 | 266.33 | 2859 | 2382 | 1985 | 1588 | 3574 | 3772 | 3971 |
| 36.00 | 0.375 | 142.68 | 1350 | 1125 | 938 | 750 | 1688 | 1781 | 1875 |
|  | 0.500 | 189.57 | 1800 | 1500 | 1250 | 1000 | 2250 | 2375 | 2500 |
|  | 0.625 | 236.13 | 2250 | 1875 | 1563 | 1250 | 2813 | 2969 | 3125 |
|  | 0.750 | 282.35 | 2700 | 2250 | 1875 | 1500 | 3375 | 3563 | 3750 |
| 42.00 | 0.375 | 166.71 | 1157 | 964 | 804 | 643 | 1446 | 1527 | 1607 |
|  | 0.500 | 221.61 | 1543 | 1286 | 1071 | 857 | 1929 | 2036 | 2143 |
|  | 0.625 | 276.18 | 1929 | 1607 | 1339 | 1071 | 2411 | 2545 | 2679 |
|  | 0.750 | 330.41 | 2314 | 1929 | 1607 | 1286 | 2893 | 3054 | 3214 |
|  | 1.000 | 437.88 | 3086 | 2571 | 2143 | 1714 | 3857 | 4071 | 4286 |

For class 1 , the range of hydrotest pressures is

$$
(1.25 \times 1872) \text { psig to }(1.3194 \times 1872) \mathrm{psig}=2340 \mathrm{psig} \text { to } 2470 \mathrm{psig}
$$

where 1.3194 is equal to the factor $1.25 \times 95 / 90$, representing the upper limit of the hydrotest envelope.

For class 2, the range of hydrotest pressures is

$$
(1.25 \times 1560) \mathrm{psig} \text { to }(1.3194 \times 1560) \mathrm{psig}=1950 \mathrm{psig} \text { to } 2058 \mathrm{psig}
$$

For class 3, the range of hydrotest pressures is

$$
(1.25 \times 1300) \mathrm{psig} \text { to }(1.3194 \times 1300) \mathrm{psig}=1625 \mathrm{psig} \text { to } 1715 \mathrm{psig}
$$

For class 4, the range of hydrotest pressures is

$$
(1.25 \times 1040) \mathrm{psig} \text { to }(1.3194 \times 1040) \mathrm{psig}=1300 \mathrm{psig} \text { to } 1372 \mathrm{psig}
$$

### 6.10 BLOWDOWN CALCULATIONS

Blowdown valves and piping systems are installed around the mainline valve in a gas transmission piping system in order to evacuate gas from sections of pipeline in the event of an emergency or for maintenance purposes. The objective of the blowdown assembly is to remove gas from the pipeline once the pipe section is isolated by closing the mainline block valves in a reasonable period of time. The pipe size required to blow down a section of pipe will depend on the gas gravity, pipe diameter, length of pipe section, pressure in the pipeline, and blowdown time. AGA recommends the following equation to estimate the blowdown time:

$$
\begin{equation*}
T=\frac{0.0588 P_{1}^{\frac{1}{3}} G^{\frac{1}{2}} D^{2} L F_{c}}{d^{2}} \quad \text { (USCS units) } \tag{6.9}
\end{equation*}
$$

where
$T$ = blowdown time, min
$P_{1}=$ initial pressure, psia
$G=$ gas gravity (air $=1.00$ )
$D=$ pipe inside diameter, in.
$L=$ length of pipe section, mi
$d$ = inside diameter of blowdown pipe, in.
$F_{c}=$ choke factor (as follows)
Choke factor list
Ideal nozzle $=1.0$
Through gate $=1.6$
Regular gate $=1.8$
Regular lube plug $=2.0$
Venturi lube plug $=3.2$

In SI units,

$$
\begin{equation*}
T=\frac{0.0192 P_{1}^{\frac{1}{3}} G^{\frac{1}{2}} D^{2} L F_{c}}{d^{2}} \quad \text { (SI units) } \tag{6.10}
\end{equation*}
$$

where
$P_{1}=$ initial pressure, kPa
$D=$ pipe inside diameter, mm
$L=$ length of pipe section, km
$d=$ pipe inside diameter of blowdown, mm

Other symbols are as defined before.

## Example 4

Calculate the blowdown time required for an NPS 6, 0.250 in. wall thickness, blowdown assembly on an NPS 24 pipe, 0.500 in . wall thickness, considering a 5 mi pipe section starting at a pressure of 1000 psia . The gas gravity is 0.6 and choke factor $=1.8$.

Solution
Pipe inside diameter $=24-2 \times 0.500=23 \mathrm{in}$.
Blowdown pipe inside diameter $=6.625-2 \times 0.250=6.125$ in.
Using Equation 6.9, we get

$$
T=\frac{0.0588 \times(1000)^{\frac{1}{3}}(0.6)^{\frac{1}{2}}(23)^{2} \times 5 \times 1.8}{6.125^{2}}=58 \text { min, approximately }
$$

### 6.11 DETERMINING PIPE TONNAGE

Frequently in pipeline design, we are interested in knowing the amount of pipe used so that we can determine the total cost of pipe. A convenient formula for calculating the weight per unit length of pipe used by pipe vendors is given in Equation 6.11.

In USCS units, pipe weight in lb/ft is calculated for a given diameter and wall thickness as follows:

$$
\begin{equation*}
w=10.68 \times t \times(D-t) \quad \text { (USCS units) } \tag{6.11}
\end{equation*}
$$

where
$w=$ pipe weight, $\mathrm{lb} / \mathrm{ft}$
$D=$ pipe outside diameter, in.
$t=$ pipe wall thickness, in.

The constant 10.68 in Equation 6.11 includes the density of steel and, therefore, the equation is only applicable to steel pipe. For other pipe material, we can ratio the densities to obtain the pipe weight for nonsteel pipe.

In SI units, the pipe weight in $\mathrm{kg} / \mathrm{m}$ is found from

$$
\begin{equation*}
w=0.0246 \times t \times(D-t) \quad \text { (SI units) } \tag{6.12}
\end{equation*}
$$

where
$w=$ pipe weight, $\mathrm{kg} / \mathrm{m}$
$D=$ pipe outside diameter, mm
$t$ = pipe wall thickness, mm

## Example 5

Calculate the total amount of pipe in a 10 mi pipeline, NPS 20, 0.500 in wall thickness. If pipe costs $\$ 700$ per ton, determine the total pipeline cost.

## Solution

Using Equation 6.11, the weight per foot of pipe is

$$
w=10.68 \times 0.500 \times(20-0.500)=101.46 \mathrm{lb} / \mathrm{ft}
$$

Therefore, the total pipe tonnage in 10 miles of pipe is

$$
\begin{gathered}
\text { Tonnage }=101.46 \times 5280 \times 10 / 2000=2679 \text { tons } \\
\text { Total pipeline cost }=2679 \times 700=\$ 1,875,300
\end{gathered}
$$

## Example 6

A 60 km pipeline consists of 20 km of DN $500,12 \mathrm{~mm}$ wall thickness pipe connected to a 40 km length of DN $400,10 \mathrm{~mm}$ wall thickness pipe. What are the total metric tons of pipe?

Solution

Using Equation 6.12, the weight per meter of DN 500 pipe is

$$
w=0.0246 \times 12 \times(500-12)=144.06 \mathrm{~kg} / \mathrm{m}
$$

and the weight per meter of DN 400 pipe is

$$
w=0.0246 \times 10 \times(400-10)=95.94 \mathrm{~kg} / \mathrm{m}
$$

Therefore, the total pipe weight for 20 km of DN 500 pipe and 40 km of DN 400 pipe is

$$
\begin{gathered}
\text { Weight }=(20 \times 144.06)+(40 \times 95.94)=6719 \text { tons } \\
\text { Total metric tons }=6719
\end{gathered}
$$

## Example 7

Calculate the MOP for NPS 16 pipeline, 0.250 in wall thickness, constructed of API 5LX-52 steel. What minimum wall thickness is required for an internal working pressure of 1440 psi? Use class 2 construction with design factor $\mathrm{F}=0.60$ and for an operating temperature below $250^{\circ} \mathrm{F}$.

## Solution

Using Equation 6.8, the internal design pressure is

$$
P=\frac{2 \times 0.250 \times 52,000 \times 0.60 \times 1.0 \times 1.0}{16}=975 \mathrm{psig}
$$

For an internal working pressure of 1440 psi , the wall thickness required is

$$
1440=\frac{2 \times t \times 52,000 \times 0.6 \times 1.0}{16}
$$

Solving for $t$, we get

$$
\text { Wall thickness } t=0.369 \mathrm{in} \text {. }
$$

The nearest standard pipe wall thickness is 0.375 in .

## Example 8

A natural gas pipeline, 600 km long, is constructed of DN 800 pipe and has a required operating pressure of 9 MPa . Compare the cost of using X-60 and X-70 steel pipe. The material costs of the two grades of pipe are as follows:

| Pipe Grade | Material Cost-\$/tonne |
| :---: | :---: |
| $\mathrm{X}-60$ | 800 |
| $\mathrm{X}-70$ | 900 |

Use a class 1 design factor and temperature deration factor of 1.00 .

## Solution

We will first determine the wall thickness of pipe required to withstand the operating pressure of 9 MPa .

Using Equation 6.8, the pipe wall thickness required for X-60 pipe ( $60,000 \mathrm{psi}=$ 414 MPa ) is

$$
t=\frac{9 \times 800}{2 \times 414 \times 1.0 \times 0.72 \times 1.0}=12.08 \mathrm{~mm} . \text { Use } 13 \mathrm{~mm} \text { wall thickness. }
$$

Similarly, the pipe wall thickness required for X-70 pipe $(70,000 \mathrm{psi}=483 \mathrm{MPa})$ is

$$
t=\frac{9 \times 800}{2 \times 483 \times 1.0 \times 0.72 \times 1.0}=10.35 \mathrm{~mm} . \text { Use } 11 \mathrm{~mm} \text { wall thickness. }
$$

The pipe weight in $\mathrm{kg} / \mathrm{m}$ will be calculated using Equation 6.12 . For X-60 pipe,

$$
\text { Weight per meter }=0.0246 \times 13 \times(800-13)=251.68 \mathrm{~kg} / \mathrm{m}
$$

Therefore, the total cost of 600 km pipeline at $\$ 800$ per ton of X-60 pipe is

$$
\text { Total cost }=600 \times 251.68 \times 800=\$ 120.81 \text { million }
$$

Similarly, the pipe weight in $\mathrm{kg} / \mathrm{m}$ for $\mathrm{X}-70$ pipe is

$$
\text { Weight per meter }=0.0246 \times 11 \times(800-11)=213.50 \mathrm{~kg} / \mathrm{m}
$$

Therefore, the total cost of 600 km pipeline at $\$ 900$ per ton of $\mathrm{X}-70$ pipe is

$$
\text { Total cost }=600 \times 213.50 \times 900=\$ 115.29 \text { million }
$$

Therefore, the X-70 pipe will cost less than the X-60 pipe. The difference in cost is

$$
\$ 120.81-\$ 115.29=\$ 5.52 \text { million. }
$$

### 6.12 SUMMARY

In this chapter we discussed how to calculate the pipe wall thickness required to withstand an internal pressure in a gas pipeline using Barlow's equation. The influence of the population density in the vicinity of the pipeline on the required pipe wall thickness by reducing the allowable hoop stress in the high population areas was explained by way of class locations. We explored the range of pressures required to hydrotest pipeline sections to ensure safe operation of the pipeline. The effect of pipeline elevations on determining a testing plan by sectioning the pipeline was covered. The need for isolating portions of the pipeline by properly spaced mainline valves and the method of calculating the time required for evacuating gas from the pipeline sections were also discussed. Finally, a simple method of calculating the pipe tonnage was explained.

## PROBLEMS

1. A gas pipeline is constructed of API 5L X70 steel, NPS 24, 0.375 in. wall thickness. Calculate the MAOP of this pipeline for a class 1 design factor and a temperature deration factor of 1.00 .
2. A gas pipeline, DN $500,12 \mathrm{~mm}$ wall thickness, is constructed of API 5L X65 pipe.
(a) Calculate the design pressures for class 1 and class 2 locations.
(b) What is the range of hydrotest pressures for each of these class locations?

Assume the joint factor $=1.00$ and temperature deration factor $=1.00$.
3. Calculate the total tonnage of pipe material for a 24 mi pipeline, NPS $16,0.375 \mathrm{in}$. wall thickness. If pipe costs $\$ 700$ per ton, determine the total pipeline cost.
4. A 50 km pipeline consists of 15 km of $\mathrm{DN} 400,10 \mathrm{~mm}$ wall thickness pipe connected to a 35 km length of DN $300,8 \mathrm{~mm}$ wall thickness pipe.
(a) Calculate the total metric tons of pipe.
(b) If this pipeline were replaced with a single 50 km long pipeline, DN 400 , API 5LX-65 material, what minimum wall thickness would be required for a class 1 design at an MOP of 9 MPa ?
5. Calculate the minimum wall thickness for NPS 16 pipeline constructed of API 5LX-60 steel to withstand an internal pressure of 1440 psi. Use a class 1 design and temperature deration factor of 1.00 .
6. A natural gas pipeline, 240 mi long, is constructed of NPS 24 pipe and has a required operating pressure of 1200 psig . Compare the cost of using X-70 or X-80 steel pipe. The material cost of X-70 pipe is $\$ 850 /$ ton, and for $\mathrm{X}-80$ pipe it is $\$ 1000 /$ ton. Use a class 1 design factor and temperature deration factor of 1.00 .
7. A natural gas pipeline, NPS 24, traverses a hilly terrain with elevations ranging from 300 ft at Norwalk to 4500 ft at the Fulton summit (milepost 50), followed by an elevation of 500 ft at the pipeline terminus at Danby. The pipeline is 100 mi long and is constructed of API 5L X-65 pipe as follows:

| Section | Wall Thickness (in.) | Class |
| :--- | :---: | :---: |
| Norwalk to mp 30 | 0.500 | 2 |
| mp 30 to Fulton | 0.500 | 1 |
| Fulton to mp 70 | 0.375 | 2 |
| mp 70 to Danby | 0.375 | 3 |

Determine a hydrostatic test plan for this pipeline, considering a test pressure envelope of 90 to $95 \%$ yield. What is the minimum number of test sections required?

## REFERENCES

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4. Mohitpour, M., Golshan, H., and Murray, A., Pipeline Design and Construction, 2nd ed., ASME Press, New York, 2003.
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## CHAPTER 7

## Thermal Hydraulics

In this chapter we will further discuss thermal hydraulics, which was briefly reviewed in Chapter 3. The importance of taking into account the gas temperature variation along a gas transmission pipeline and its impact on pressure drop and flow rate will be explained. Isothermal hydraulics, which formed the majority of calculations in Chapter 2 and Chapter 3, will be compared with thermal hydraulics. Since manual calculation of gas pipeline hydraulics, considering thermal effects, is quite laborious and time consuming, we will use examples of pipeline simulation cases using a popular gas pipeline hydraulics software application.

### 7.1 ISOTHERMAL VERSUS THERMAL HYDRAULICS

In the previous chapters the hydraulic analysis of gas flow through pipelines was mainly done based upon isothermal or constant temperature flow. This assumption is fairly good in long-distance pipelines where the gas temperature reaches a constant value equal to or close to the surrounding soil (or ambient) temperature at large distances from the compressor stations. However, upon compressing the gas, depending on the compression ratio, the outlet temperature of the gas from the compressor station can be considerably higher than that of the ambient air or surrounding soil. In Chapter 4, Example 8, we found that when gas is compressed adiabatically from a $60^{\circ} \mathrm{F}$ suction temperature and a compression ratio of 2.0 , the discharge temperature is $278.3^{\circ} \mathrm{F}$. Since pipe coating limitations restrict temperatures to about 140 to $150^{\circ} \mathrm{F}$, cooling of the compressed gas is necessary at the downstream side of the compressor station. In this example, assuming gas cooling results in a discharge temperature of $140^{\circ} \mathrm{F}$ as gas enters the pipeline, we find that the temperature difference between the gas at $140^{\circ} \mathrm{F}$ and the surrounding soil at $70^{\circ} \mathrm{F}$ will cause heat transfer to take place between the pipeline gas and the surrounding soil. It is found that the gas temperature drops off rapidly for the first few miles and eventually reaches a temperature close to the soil temperature. Additionally, in a long transmission pipeline, the soil temperature can vary along the pipeline as well, causing different heat transfer rates at locations along the pipeline. This is illustrated in Figure 7.1.


Figure 7.1 Temperature variation in a gas pipeline.

In some instances, the expansion of gas as it flows along a pipeline can result in gas temperature reaching a slightly lower temperature than the surrounding soil. This is called the Joule-Thompson cooling effect. Thus, if the soil temperature is fairly constant at $70^{\circ} \mathrm{F}$, due to the Joule-Thompson effect, the final temperature of the gas at the terminus of the pipeline can drop to 60 or $65^{\circ} \mathrm{F}$. This is illustrated in Figure 7.2.

This cooling of gas below the surrounding soil temperature depends on the pressure differential and the Joule-Thompson coefficient. Ignoring this cooling effect will result in a more conservative (lower flow rate for a given pressure drop) flow rate calculation, since cooler temperature means less pressure drop in a gas pipeline and, hence, higher flow rate.

Thermal hydraulics is the study of gas pipeline pressures, temperatures, and flow rates, taking into account the thermal properties of the soil, pipe, and pipe insulation, if any. Due to such variation in gas temperature, calculation of pressure drop must be made by considering short lengths of pipe that make up the total pipeline. For example, if the pipeline is 50 mi long, we subdivide it into short segments of 1 or 2 mi lengths and apply the General Flow equation for each pipe segment, considering


Figure 7.2 Joule-Thompson cooling effect in a gas pipeline.
an average gas temperature and an average ambient soil temperature. Starting with the upstream pressure of segment 1 , the downstream pressure will be calculated assuming an average temperature for segment 1 . Next, using the calculated downstream pressure as the upstream pressure for segment 2 , we calculate the downstream pressure for segment 2 . The process is continued until all segments of the pipeline are covered. It must be noted that the variation of temperature from segment to segment must be taken into account to calculate the compressibility factor used in the General Flow equation. The following equation is the General Flow equation that we used frequently in Chapter 2 and Chapter 3:

$$
\begin{equation*}
Q=38.77 F\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G T_{f} L_{e} Z}\right)^{0.5} D^{2.5} \tag{7.1}
\end{equation*}
$$

After examining Equation 7.1, we see that if the gas flowing temperature $T_{f}$ is constant (isothermal flow) throughout the length of the pipeline $L_{e}$, the compressibility factor $Z$ of the gas will depend on the average pressure of the pipe segment. If the gas temperature also varies along the pipeline, based on the preceding discussions, the compressibility factor $Z$ will change in a different manner, since it is a function of both the gas temperature and gas pressure. Therefore, it is seen that calculation of $P_{1}$ or $P_{2}$ for a given flow rate $Q$ from Equation 7.1 will yield different results if isothermal conditions do not exist.

The calculation of gas temperature at any point along the pipeline, taking into account the heat transfer between the gas and surrounding soil, is quite complicated. It does not lend itself easily to manual calculations. We will discuss the method of calculation for thermal hydraulics in this section. To accurately take into account the temperature variations, a suitable gas pipeline hydraulics simulation program must be used since, as indicated earlier, manual calculation is quite laborious and time consuming. Several commercial simulation programs are available to model steady-state gas pipeline hydraulics. These programs calculate the gas temperatures and pressures by taking into consideration variations of soil temperature, pipe burial depth, and thermal conductivities of pipe, insulation, and soil. One such software program is GASMOD, marketed by SYSTEK Technologies, Inc. (www.systek.us). Appendix D includes a sample simulation of a gas pipeline using the GASMOD software. In this chapter we will use GASMOD to illustrate thermal hydraulics analysis.

### 7.2 TEMPERATURE VARIATION AND GAS PIPELINE MODELING

Consider a buried pipeline transporting gas from point A to point B . We will analyze a short segment of length $\Delta L$ of this pipe, as shown in Figure 7.3, and apply the principles of heat transfer to determine how the gas temperature varies along the pipeline.

The upstream end of the pipe segment of length $\Delta L$ is at temperature $T_{1}$ and the downstream end at temperature $T_{2}$. The average gas temperature in this segment is represented by $T$. The outside soil temperature at this location is $T_{s}$. Assume steadystate conditions and the mass flow rate of gas to be $m$. The gas flow from the upstream


Figure 7.3 Analysis of temperature variation.
end to the downstream end of the segment causes a temperature drop of $\Delta T$. The heat loss from the gas can be represented by

$$
\begin{equation*}
\Delta H=-m C p \Delta T \tag{7.2}
\end{equation*}
$$

where
$\Delta H=$ heat transfer rate, Btu/h
$m=$ mass flow rate of gas, $\mathrm{lb} / \mathrm{h}$
$C p=$ average specific heat of gas, $\mathrm{Btu} / \mathrm{lb} /{ }^{\circ} \mathrm{F}$
$\Delta T=$ temperature difference $=T_{1}-T_{2},{ }^{\circ} \mathrm{F}$
The negative sign in Equation 7.2 indicates loss of heat from upstream temperature $T_{1}$ to downstream temperature $T_{2}$.

Next, we consider the heat transfer from the gas to the surrounding soil in terms of the overall heat transfer coefficient $U$ and the difference in temperature between the gas and surrounding soil represented by $\left(T-T_{s}\right)$. Therefore, we can write the following equation for heat transfer:

$$
\begin{equation*}
\Delta H=U \Delta A\left(T-T_{s}\right) \tag{7.3}
\end{equation*}
$$

where
$U=$ overall heat transfer coefficient, Btu/h $/ \mathrm{ft}^{2} /{ }^{\circ} \mathrm{F}$
$\Delta A=$ surface area of pipe for heat transfer $=\pi D \Delta L$
$T=$ average gas temperature in pipe segment, ${ }^{\circ} \mathrm{F}$
$T_{s}=$ average soil temperature surrounding pipe segment, ${ }^{\circ} \mathrm{F}$
$D=$ pipe inside diameter, ft
Equating the two values of the heat transfer rate $\Delta H$ from Equation 7.2 and Equation 7.3, we get

$$
-m C p \Delta T=U \Delta A\left(T-T_{s}\right)
$$

Simplifying, we get

$$
\begin{equation*}
\frac{\Delta T}{T-T_{s}}=-\left(\frac{\pi U D}{m C p}\right) \Delta L \tag{7.4}
\end{equation*}
$$

Rewriting Equation 7.4 in differential form and integrating, we get

$$
\begin{equation*}
\int_{1}^{2} \frac{d T}{T-T_{s}}=\int_{2}^{1}-\left(\frac{\pi U D}{m C p}\right) d L \tag{7.5}
\end{equation*}
$$

Integrating and simplifying, we get

$$
\begin{equation*}
\frac{T_{2}-T_{s}}{T_{1}-T_{s}}=e^{-\theta} \tag{7.6}
\end{equation*}
$$

where $e=$ base of natural logarithms $(e=2.718 \ldots)$ and

$$
\begin{equation*}
\theta=\frac{\pi U D \Delta L}{m C p} \tag{7.7}
\end{equation*}
$$

Simplifying Equation 7.6 further, we get the downstream temperature of the pipe segment of length $\Delta L$ as

$$
\begin{equation*}
T_{2}=T_{s}+\left(T_{1}-T_{s}\right) e^{-\theta} \tag{7.8}
\end{equation*}
$$

It can be seen from Equation 7.8 that as the pipe length increases, the term $e^{-\theta}$ approaches zero and the temperature $T_{2}$ becomes equal to soil temperature $T_{\mathrm{s}}$. Therefore, in a long gas pipeline, the gas temperature ultimately equals the surrounding soil temperature. This is illustrated in Figure 7.1.

In the preceding analysis, we made several simplifying assumptions. We assumed that the soil temperature and the overall heat transfer coefficient remained constant and ignored the Joule-Thompson effect as gas expands through a pipeline. In a long pipeline, the soil temperature may actually vary along the pipeline and, therefore, must be taken into account in these calculations. One approach would be to subdivide the pipeline into segments that have constant soil temperatures and perform calculations for each segment separately. The Joule-Thompson effect causes the gas to cool slightly due to expansion. Therefore, in a long pipeline, the gas temperature at the delivery point can fall below that of the ground or soil temperature, as indicated in Figure 7.2.

### 7.3 REVIEW OF SIMULATION MODEL REPORTS

To illustrate thermal effects in a gas pipeline, we will analyze a gas transmission pipeline, first using the method outlined in Chapter 3. Next, we will analyze the same pipeline, taking into account the thermal conductivity and soil temperatures. The latter method requires some form of computer simulation models. To do this, we have chosen the commercially available software known as GASMOD. We will compare the results of the isothermal hydraulics of Chapter 3 with the thermal hydraulics using GASMOD. Examples will be used to illustrate the comparison.

## Example 1

A natural gas pipeline system is being built from Rockport to Concord, a distance of 240 miles. The pipeline is constructed of NPS 30, 0.500 in . wall thickness, API 5L-X60 pipe. The MOP is 1400 psig. Gas enters the Rockport compressor station at $70^{\circ} \mathrm{F}$ and 800 psig pressure. The soil temperature can be assumed to be $60^{\circ} \mathrm{F}$ throughout. The gas flow rate is 420 MMSCFD, and the gas specific gravity and viscosity are 0.6 and $0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{s}$, respectively. The contract delivery pressure required at Concord is 500 psig . Assume an isothermal flow at $70^{\circ} \mathrm{F}$ and a gas specific heat ratio of 1.29 . Use a compressor adiabatic efficiency of $80 \%$ and mechanical efficiency of $98 \%$. Use the General Flow equation with a Colebrook friction factor, assuming a pipe internal roughness of $700 \mu \mathrm{in}$. Calculate the pressure profile and the compressor horsepower required at Rockport. Compare these results with thermal hydraulic analysis using GASMOD. Assume a base pressure of 14.7 psia and base temperature of $60^{\circ} \mathrm{F}$. The pipeline elevation profile is essentially flat.

## Solution

Inside diameter of pipe $D=30-2 \times 0.500=29 \mathrm{in}$.
First, we calculate the Reynolds number from Equation 2.34:

$$
R=0.0004778\left(\frac{14.7}{60+460}\right)\left(\frac{0.6 \times 420 \times 10^{6}}{0.000008 \times 29}\right)=14,671,438
$$

Next, using Colebrook Equation 2.39, we calculate the friction factor as

$$
\frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{0.0007}{3.7 \times 29}+\frac{2.51}{14,671,438 \sqrt{f}}\right)
$$

Solving by trial and error, we get

$$
f=0.0097
$$

Therefore, the transmission factor is, using Equation 2.42,

$$
F=\frac{2}{\sqrt{0.0097}}=20.33
$$

To calculate the compressibility factor $Z$, the average pressure is required. Since the inlet pressure is unknown, we will calculate an approximate value of $Z$ using a value of $110 \%$ of the delivery pressure for the average pressure.

The average pressure is

$$
P_{\mathrm{avg}}=1.1 \times(500+14.7)=566.17 \mathrm{psia}=551.47 \mathrm{psig}
$$

Using CNGA Equation 1.34, we calculate the value of compressibility factor as

$$
Z=\frac{1}{\left[1+\left(\frac{(566.17-14.7) \times 344,400(10)^{1.785 \times 0.6}}{530^{3.825}}\right)\right]}=0.9217
$$

Since there is no elevation difference between the beginning of the pipeline and the end of the pipeline, the elevation component in Equation 2.7 can be neglected and $e^{s}=1$.

The outlet pressure is

$$
P_{2}=500+14.7=514.7 \text { psia. }
$$

From General Flow Equation 2.4, substituting given values, we get

$$
420 \times 10^{6}=38.77 \times 20.33\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-514.7^{2}}{0.6 \times 530 \times 240 \times 0.9217}\right)^{0.5}(29.0)^{2.5}
$$

Solving for the upstream pressure, we get

$$
P_{1}=1021.34 \mathrm{psia}=1006.64 \mathrm{psig}
$$

Using this value of $P_{1}$, we calculate the new average pressure using Equation 2.14:

$$
P_{\mathrm{avg}}=\frac{2}{3}\left(1021.34+514.7-\frac{1021.34 \times 514.7}{1021.34+514.7}\right)=795.87 \mathrm{psia}
$$

This compares with the value of 566.17 psia we assumed initially for calculating $Z$. Obviously, we were way off. Recalculating $Z$ using the recently calculated value of $P_{\text {avg }}$, we get

$$
Z=\frac{1}{\left[1+\left(\frac{(795.87-14.7) \times 344,400(10)^{1.785 \times 0.6}}{530^{3.825}}\right)\right]}=0.8926
$$

We will now recalculate the inlet pressure using this value of $Z$. From General Flow Equation 2.4, we get

$$
420 \times 10^{6}=38.77 \times 20.33\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-514.7^{2}}{0.6 \times 530 \times 240 \times 0.8926}\right)^{0.5}(29.0)^{2.5}
$$

Solving for the upstream pressure, we get

$$
P_{1}=1009.24 \mathrm{psia}=994.54 \mathrm{psig}
$$

This compares with 1021.34 psia calculated earlier. This is almost $1 \%$ different. We could repeat the process and get a better approximation.

Using this recently calculated value of $P_{1}$, we calculate the new average pressure using Equation 2.14 as

$$
P_{\text {avg }}=\frac{2}{3}\left(1009.24+514.7-\frac{1009.24 \times 514.7}{1009.24+514.7}\right)=788.72 \mathrm{psia}
$$

This compares with the previous approximation of 795.87 psia . The error is less than $1 \%$.

Recalculating $Z$ using this value of $P_{\text {avg }}$, we get

$$
Z=\frac{1}{\left[1+\left(\frac{(788.72-14.7) \times 344,400(10)^{1.785 \times 0.6}}{530^{3.825}}\right)\right]}=0.8935
$$

We will now recalculate the inlet pressure using this value of $Z$. From General Flow Equation 2.4, we get

$$
420 \times 10^{6}=38.77 \times 20.33\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-514.7^{2}}{0.6 \times 530 \times 240 \times 0.8935}\right)^{0.5}(29.0)^{2.5}
$$

Solving for the upstream pressure, we get

$$
P_{1}=1009.62 \mathrm{psia}=994.92 \mathrm{psig}
$$

This compares with 1009.24 psia calculated earlier. The difference is less than $0.04 \%$; therefore, we can stop iterating any further.

The HP required at the Rockport compressor station will be calculated using Equation 4.15 as follows:
$H P=0.0857 \times 420\left(\frac{1.29}{0.29}\right)(70+460)\left(\frac{1+0.8935}{2}\right)\left(\frac{1}{0.8}\right)\left[\left(\frac{1009.62}{814.7}\right)^{\frac{0.29}{1.29}}-1\right]=4962$
Using Equation 4.17, we calculate the driver horsepower required, based on a mechanical efficiency of 0.98 .

$$
\text { BHP required }=\frac{4962}{0.98}=5063
$$

The final results are:
Inlet pressure at Rockport $=994.92 \mathrm{psig}$

Delivery pressure at Concord $=500 \mathrm{psig}$ at a flow rate of 420 MMSCFD
BHP required at Rockport compressor station $=5063$ HP

It must be noted that the preceding calculations ignored any elevation changes along the pipeline. If we had considered the pipe elevations at Rockport and Concord, the result would have been different.

This pressure of 994.92 psig required at the Rockport compressor station was calculated assuming a constant gas flowing temperature of $70^{\circ} \mathrm{F}$ and considering the pipeline as one single segment 240 mi long. As explained in earlier chapters, the calculation accuracy is improved if we subdivide the pipeline into short segments. By doing so, we calculate the upstream pressure of each segment starting with the last segment near Concord. If the pipeline is divided into 100 equal pipe segments of 2.4 mi each, the pressure $P_{100}$ at the upstream end of the last segment is calculated using the General Flow equation, considering a 500 psig downstream pressure. Next, using this calculated pressure, $P_{100}$, we calculate the upstream pressure $P_{99}$ of the 99th segment. The process is repeated until all segments are covered and the value of the pressure $P_{1}$ at Rockport is calculated. This is illustrated in Figure 7.4.

By subdividing the pipeline in this fashion, we are improving the accuracy of calculations. Of course, manual calculation in this manner is going to be quite laborious and time consuming, and we should use some form of a computer program to perform this task.

Next, we will compare the isothermal calculation results with thermal hydraulics using the GASMOD program. Given next is the output report from the GASMOD program.


Figure 7.4 Subdividing pipe into segments.

```
******** GASMOD - GAS PIPELINE HYDRAULIC SIMULATION
************ 32-bit Version 5.00.100
DATE: 16-September-2004 TIME: 21:30:37
PROJECT DESCRIPTION:
Pipeline from Rockport to Concord
Case Number:
Pipeline data file: C:\GASMOD32\RockportPipeline.TOT
Pressure drop formula: Colebrook-White
Pipeline efficiency: 1.00
Compressibility factor method:
Inlet gas gravity(air=1.0):
Inlet gas viscosity:
CALCULATION OPTIONS:
Branch pipe calculations: NO
Loop pipe calculations: NO
Compressor fuel calculated: NO
Joule-Thompson effect included: No
Customized output:
NO
Holding delivery pressure
at terminus
```1139

C: \GASMOD32\RockportPipeline.TOT

Colebrook-White
1.00

CNGA
0.60000
\(0.0000080(1 \mathrm{~b} / \mathrm{ft}-\mathrm{sec})\)

NO
NO
NO
NO
NO
```

Holding delivery pressure
at terminus
$* * * * * * * *$ Calculations Based on Specified Thermal
Conductivities of Pipe, Soil, and Insulation

Origin suction temperature:
Base temperature:
Base pressure:
Origin suction pressure:
Delivery pressure:
Minimum pressure:
Gas specific heat ratio:
Maximum gas velocity:

Inlet flow rate:
Outlet flow rate:

| Distance (mi) | Elevation <br> (ft) | Diameter (in.) | Thickness (in.) | Roughness (in.) |
| :---: | :---: | :---: | :---: | :---: |
| 0.00 | 250.00 | 30.000 | 0.500 | 0.000700 |
| 10.00 | 250.00 | 30.000 | 0.500 | 0.000700 |
| 20.00 | 250.00 | 30.000 | 0.500 | 0.000700 |
| 30.00 | 250.00 | 30.000 | 0.500 | 0.000700 |

```
\begin{tabular}{rrrrr}
40.00 & 250.00 & 30.000 & 0.500 & 0.000700 \\
50.00 & 250.00 & 30.000 & 0.500 & 0.000700 \\
60.00 & 250.00 & 30.000 & 0.500 & 0.000700 \\
70.00 & 250.00 & 30.000 & 0.500 & 0.000700 \\
100.00 & 250.00 & 30.000 & 0.500 & 0.000700 \\
120.00 & 250.00 & 30.000 & 0.500 & 0.000700 \\
140.00 & 250.00 & 30.000 & 0.500 & 0.000700 \\
150.00 & 250.00 & 30.000 & 0.500 & 0.000700 \\
170.00 & 250.00 & 30.000 & 0.500 & 0.000700 \\
190.00 & 250.00 & 30.000 & 0.500 & 0.000700 \\
200.00 & 250.00 & 30.000 & 0.500 & 0.000700 \\
220.00 & 250.00 & 30.000 & 0.500 & 0.000700 \\
240.00 & 250.00 & 30.000 & 0.500 & 0.000700
\end{tabular}
\begin{tabular}{|c|c|c|c|c|c|c|}
\hline \multirow[b]{2}{*}{Distance (mi)} & \multirow[b]{2}{*}{\begin{tabular}{l}
Cover \\
(in)
\end{tabular}} & \multicolumn{3}{|l|}{\begin{tabular}{l}
Thermal Conductivity \\
(Btu/hr/ft/degF)
\end{tabular}} & \multirow[b]{2}{*}{\[
\begin{aligned}
& \text { Insul.Thk } \\
& \text { (in) }
\end{aligned}
\]} & \multirow[b]{2}{*}{Soil Temp (degF)} \\
\hline & & Pipe & Soil & Insulation & & \\
\hline 0.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
\hline 10.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
\hline 20.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
\hline 30.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
\hline 40.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
\hline 50.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
\hline 60.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
\hline 70.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
\hline 100.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
\hline 120.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
\hline 140.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
\hline 150.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
\hline 170.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
\hline 190.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
\hline 200.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
\hline 220.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
\hline 240.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
\hline
\end{tabular}
**************** COMPRESSOR STATION DATA
FLOW RATES, PRESSURES, AND TEMPERATURES:
\begin{tabular}{|c|c|c|c|c|c|c|c|c|c|}
\hline Name & \begin{tabular}{l}
Flow \\
Rate \\
(MMSCFD)
\end{tabular} & \begin{tabular}{l}
Suct. \\
Press. \\
(psig)
\end{tabular} & \begin{tabular}{l}
Disch. \\
Press. \\
(psig)
\end{tabular} & \begin{tabular}{l}
Compr. \\
Ratio \\
(psig)
\end{tabular} & \begin{tabular}{l}
Suct. \\
Loss. \\
(psig)
\end{tabular} & \begin{tabular}{l}
Disch. \\
Loss. \\
(degF)
\end{tabular} & \begin{tabular}{l}
Suct. \\
Temp \\
(degF)
\end{tabular} & \begin{tabular}{l}
Disch. \\
Temp \\
(degF)
\end{tabular} & MaxPipe Temp \\
\hline Rockport & 420.00 & 800.00 & 996.17 & 1.2408 & 0.00 & 0.00 & 70.00 & 102.92 & 140.00 \\
\hline
\end{tabular}
\(* * * * * * * * * ~ C O M P R E S S O R ~ E F F I C I E N C Y, ~ H P, ~ A N D ~ F U E L ~ U S E D ~\) ********

\begin{tabular}{|c|c|c|c|c|c|}
\hline \[
\begin{aligned}
& \text { Distance } \\
& \text { (mi) }
\end{aligned}
\] & Reynold'sNum. & \begin{tabular}{l}
FrictFactor \\
(Darcy)
\end{tabular} & Transmission Factor & HeatTransCoeff (Btu/hr/ ft2/degF) & Compressibility Factor (CNGA) \\
\hline 0.000 & 14,671,438. & 0.0099 & 20.13 & 0.3361 & 0.8865 \\
\hline 10.000 & 14,671,438. & 0.0099 & 20.13 & 0.3361 & 0.8791 \\
\hline 20.000 & 14,671,438. & 0.0099 & 20.13 & 0.3361 & 0.8747 \\
\hline 30.000 & 14,671,438. & 0.0099 & 20.13 & 0.3361 & 0.8725 \\
\hline 40.000 & 14,671,438. & 0.0099 & 20.13 & 0.3361 & 0.8718 \\
\hline 50.000 & 14,671,438. & 0.0099 & 20.13 & 0.3361 & 0.8722 \\
\hline 60.000 & 14,671,438. & 0.0099 & 20.13 & 0.3361 & 0.8733 \\
\hline 70.000 & 14,671,438. & 0.0099 & 20.13 & 0.3361 & 0.8767 \\
\hline 100.000 & 14,671,438. & 0.0099 & 20.13 & 0.3361 & 0.8821 \\
\hline 120.000 & 14,671,438. & 0.0099 & 20.13 & 0.3361 & 0.8870 \\
\hline 140.000 & 14,671,438. & 0.0099 & 20.13 & 0.3361 & 0.8909 \\
\hline 150.000 & 14,671,438. & 0.0099 & 20.13 & 0.3361 & 0.8952 \\
\hline 170.000 & 14,671,438. & 0.0099 & 20.13 & 0.3361 & 0.9012 \\
\hline 190.000 & 14,671,438. & 0.0099 & 20.13 & 0.3361 & 0.9061 \\
\hline 200.000 & 14,671,438. & 0.0099 & 20.13 & 0.3361 & 0.9114 \\
\hline 220.000 & 14,671,438. & 0.0099 & 20.13 & 0.3361 & 0.9171 \\
\hline 240.000 & 14,671,438. & 0.0099 & 20.13 & 0.3361 & 0.9171 \\
\hline
\end{tabular}

\section*{PIPELINE TEMPERATURE AND PRESSURE PROFILE ******}
\begin{tabular}{rcccccccc}
\begin{tabular}{c} 
Distance \\
(mi)
\end{tabular} & \begin{tabular}{c} 
Diameter \\
(in)
\end{tabular} & \begin{tabular}{c} 
Flow \\
(MMSCFD)
\end{tabular} & \begin{tabular}{c} 
Velocity \\
(ft/sec)
\end{tabular} & \begin{tabular}{c} 
Press. \\
(psig)
\end{tabular} & \begin{tabular}{c} 
GasTemp. \\
\((\) degF)
\end{tabular} & \begin{tabular}{c} 
SoilTemp. \\
(degF)
\end{tabular} & \begin{tabular}{c} 
MAOP \\
\((\) psig)
\end{tabular} & Location \\
0.00 & 30.000 & 420.0000 & 15.44 & 996.17 & 102.92 & 60.00 & 1400.00 Rockport \\
10.00 & 30.000 & 420.0000 & 15.44 & 979.59 & 87.37 & 60.00 & 1400.00 \\
20.00 & 30.000 & 420.0000 & 15.44 & 963.25 & 77.22 & 60.00 & 1400.00 \\
30.00 & 30.000 & 420.0000 & 15.44 & 946.96 & 70.74 & 60.00 & 1400.00 \\
40.00 & 30.000 & 420.0000 & 15.44 & 930.60 & 66.66 & 60.00 & 1400.00 \\
50.00 & 30.000 & 420.0000 & 15.44 & 914.07 & 64.11 & 60.00 & 1400.00 \\
60.00 & 30.000 & 420.0000 & 15.44 & 897.30 & 62.54 & 60.00 & 1400.00 \\
70.00 & 30.000 & 420.0000 & 15.44 & 880.23 & 61.56 & 60.00 & 1400.00 \\
100.00 & 30.000 & 420.0000 & 15.44 & 826.86 & 60.36 & 60.00 & 1400.00
\end{tabular}
\begin{tabular}{|c|c|c|c|c|c|c|c|c|}
\hline 120.00 & 30.000 & 420.0000 & 15.44 & 789.12 & 60.14 & 60.00 & 1400.00 & \\
\hline 140.00 & 30.000 & 420.0000 & 15.44 & 749.30 & 60.05 & 60.00 & 1400.00 & \\
\hline 150.00 & 30.000 & 420.0000 & 15.44 & 728.51 & 60.03 & 60.00 & 1400.00 & \\
\hline 170.00 & 30.000 & 420.0000 & 15.44 & 684.85 & 60.01 & 60.00 & 1400.00 & \\
\hline 190.00 & 30.000 & 420.0000 & 15.44 & 637.95 & 60.00 & 60.00 & 1400.00 & \\
\hline 200.00 & 30.000 & 420.0000 & 15.44 & 613.06 & 60.00 & 60.00 & 1400.00 & \\
\hline 220.00 & 30.000 & 420.0000 & 15.44 & 559.72 & 60.00 & 60.00 & 1400.00 & \\
\hline 240.00 & 30.000 & 420.0000 & 15.44 & 500.35 & 60.00 & 60.00 & 1400.00 & Concord \\
\hline \multicolumn{3}{|l|}{*************} & \multicolumn{2}{|l|}{PACK VOLUMES AND} & \multicolumn{2}{|l|}{PRESSURES *} & \multicolumn{2}{|l|}{*************} \\
\hline
\end{tabular}
\begin{tabular}{|c|c|c|}
\hline Distance (mi) & Pressure (psig) & \begin{tabular}{l}
Line Pack \\
(million std.cu.ft)
\end{tabular} \\
\hline 0.00 & 996.17 & 0.0000 \\
\hline 10.00 & 979.59 & 17.8138 \\
\hline 20.00 & 963.25 & 17.9714 \\
\hline 30.00 & 946.96 & 17.9566 \\
\hline 40.00 & 930.60 & 17.8193 \\
\hline 50.00 & 914.07 & 17.5990 \\
\hline 60.00 & 897.30 & 17.3235 \\
\hline 70.00 & 880.23 & 17.0114 \\
\hline 100.00 & 826.86 & 48.9787 \\
\hline 120.00 & 789.12 & 30.7496 \\
\hline 140.00 & 749.30 & 29.1424 \\
\hline 150.00 & 728.51 & 13.9438 \\
\hline 170.00 & 684.85 & 26.5765 \\
\hline 190.00 & 637.95 & 24.7442 \\
\hline 200.00 & 613.06 & 11.6489 \\
\hline 220.00 & 559.72 & 21.7584 \\
\hline 240.00 & 500.35 & 19.5576 \\
\hline \multicolumn{3}{|l|}{Total line pack in main pipeline \(=350.5950\) (million std.cu.ft)} \\
\hline ********* & f GASMOD & t ************* \\
\hline
\end{tabular}

It can be seen from the GASMOD thermal hydraulic analysis report that the inlet pressure at Rockport is 996.17 psig, whereas the manual calculation considering isothermal flow yielded an inlet pressure of 994.92 psig at the Rockport compressor station. Thus, taking into account the temperature variation of the gas along the pipeline, the pressure required at Rockport is approximately 4 psig higher. This does not seem to be very significant. However, in many cases the temperature variation along the pipeline will cause pressures calculated to be significantly different. To recap, the manual calculations were based on an isothermal gas flow temperature of \(70^{\circ} \mathrm{F}\), whereas the thermal hydraulics shows variation of the gas temperature ranging from \(102.92^{\circ} \mathrm{F}\) at the Rockport compressor discharge to \(60^{\circ} \mathrm{F}\) at Concord. The gas temperature reaches the soil temperature of \(60^{\circ} \mathrm{F}\) at approximately milepost 190 , after which it remains constant at \(60^{\circ} \mathrm{F}\). Figure 7.5 shows the temperature variation in this case.

Next, we will illustrate the calculation of the pressure and temperature profile considering the pipeline elevation difference between Rockport and Concord and considering a branch pipeline bringing in an additional 200 MMSCFD.


Figure 7.5 Gas temperature variation-Rockport to Concord pipeline.

\section*{Example 2}

Consider a natural gas pipeline system from Rockport (elevation 250 ft ) to Concord (elevation 800 ft ), a distance of 240 mi . The pipeline is constructed of NPS 30, 0.500 in . wall thickness, API 5L-X60 pipe. The MOP is 1400 psig.

The pipeline elevation profile is listed below:
\begin{tabular}{rcc}
\hline Milepost & Elevation & Location \\
\hline 0.00 & 250.00 & Rockport \\
10.00 & 300.00 & \\
20.00 & 200.00 & \\
30.00 & 320.00 & \\
40.00 & 400.00 & \\
50.00 & 375.00 & \\
60.00 & 410.00 & \\
70.00 & 430.00 & \\
100.00 & 450.00 & \\
120.00 & 500.00 & \\
140.00 & 400.00 & \\
150.00 & 600.00 & \\
170.00 & 700.00 & \\
190.00 & 710.00 & \\
200.00 & 720.00 & \\
220.00 & 750.00 & \\
240.00 & 800.00 & Concord \\
\hline
\end{tabular}

Gas enters the Rockport compressor station at \(70^{\circ} \mathrm{F}\) and 800 psig pressure. The soil temperature can be assumed to be \(60^{\circ} \mathrm{F}\) throughout. The gas flow rate is 420 MMSCFD , and the gas specific gravity and viscosity are 0.6 and \(0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{s}\), respectively. At Vale (milepost 100, elevation 450 ft ), a branch pipeline 80 mi long, NPS 24, 0.375 in . wall thickness, brings in an additional 200 MMSCFD gas from a gathering facility at Drake. The elevation at Drake is 300 ft and that at Vale is 450 ft .

The pipeline elevation profile for the branch pipe from Drake to Vale is as follows:
\begin{tabular}{ccc}
\hline Milepost & Elevation & Location \\
\hline 0.00 & 300.00 & Drake \\
10.00 & 100.00 & \\
20.00 & 125.00 & \\
40.00 & 200.00 & \\
50.00 & 250.00 & \\
70.00 & 300.00 & Vale \\
80.00 & 450.00 & \\
\hline
\end{tabular}

The inlet temperature at the beginning of the branch is \(70^{\circ} \mathrm{F}\) in Figure 7.6.

The contract delivery pressure required at Concord is 500 psig . Assume an isothermal flow at \(70^{\circ} \mathrm{F}\) and gas specific heat ratio of 1.29 . Use a compressor adiabatic efficiency of \(80 \%\) and mechanical efficiency of \(98 \%\). Use the General Flow equation with a Colebrook friction factor, assuming a pipe internal roughness of \(700 \mu\) in. Calculate the pressure profile and the compressor horsepower required at Rockport. Compare these results with thermal hydraulic analysis using GASMOD. Assume a base pressure of 14.7 psia and base temperature of \(60^{\circ} \mathrm{F}\).

\section*{Solution}

Inside diameter of pipe \(D=30-2 \times 0.500=29 \mathrm{in}\).

The calculation of the pressure at milepost 100 will be done first. This is because we know the delivery pressure at Concord and the 140 mi segment from Vale (at milepost 100 to the pipeline terminus at Concord) flows 620 MMSCFD . In comparison, the first 100 mi from Rockport to Vale, carries only 420 MMSCFD , and both upstream and downstream pressures are unknown. After finding the pressure at Vale, we can


Figure 7.6 Rockport to Concord pipeline with branch from Drake.
calculate the upstream pressure at Rockport, considering the 100 mi segment at the lower flow rate.

First, we calculate the Reynolds number from Equation 2.34 for a 620 MMSCFD flow rate:
\[
R=0.0004778\left(\frac{14.7}{60+460}\right)\left(\frac{0.6 \times 620 \times 10^{6}}{0.000008 \times 29}\right)=21,657,837
\]

Next, using Colebrook Equation 2.39, we calculate the friction factor as
\[
\frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{0.0007}{3.7 \times 29}+\frac{2.51}{21,657,837 \sqrt{f}}\right)
\]

Solving by trial and error, we get
\[
f=0.0096
\]

Therefore, the transmission factor is, using Equation 2.42,
\[
F=\frac{2}{\sqrt{0.0096}}=20.41
\]

To calculate the compressibility factor \(Z\), the average pressure is required. Since the inlet pressure at Vale is unknown, we will calculate an approximate value of \(Z\) using a value of \(110 \%\) of the delivery pressure for the average pressure.

The average pressure is
\[
P_{\text {avg }}=1.1 \times(500+14.7)=566.17 \mathrm{psia}=551.47 \mathrm{psig}
\]

Using CNGA Equation 1.34, we calculate the value of the compressibility factor as
\[
Z=\frac{1}{\left[1+\left(\frac{(566.17-14.7) \times 344,400(10)^{1.785 \times 0.6}}{530^{3.825}}\right)\right]}=0.9217
\]

Since there is an elevation difference of \(350(800-450) \mathrm{ft}\) between Vale and Concord, we must apply the elevation correction according to Equation 2.7.

Using Equation 2.10, the elevation adjustment parameter is
\[
s=0.0375 \times 0.6\left(\frac{800-450}{530 \times 0.9217}\right)=0.0161
\]

The equivalent length \(L_{e}\) from Equation 2.9 is
\[
L_{e}=\frac{140(1.0163-1)}{0.0161}=141.74 \mathrm{mi}
\]

Next, using Equation 2.7, we calculate the pressure at Vale, milepost 100, as follows:
\[
620 \times 10^{6}=38.77 \times 20.41\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-1.0163 \times 514.7^{2}}{0.6 \times 530 \times 141.74 \times 0.9217}\right)^{0.5} 29^{2.5}
\]

Solving for the upstream pressure at Vale, we get
\[
P_{1}=1122.49 \mathrm{psia}=1107.79 \mathrm{psig}
\]

Using this calculated value of \(P_{1}\), we calculate the new average pressure using Equation 2.14 as
\[
P_{\mathrm{avg}}=\frac{2}{3}\left(1122.49+514.7-\frac{1122.49 \times 514.7}{1122.49+514.7}\right)=856.2 \mathrm{psia}=841.5 \mathrm{psig}
\]

This compares with the previous approximation of 551.47 psig. Obviously, the assumed value was way off. Recalculating the compressibility factor, using the recently calculated average pressure,
\[
Z=\frac{1}{\left[1+\left(\frac{(856.2-14.7) \times 344,400(10)^{1.785 \times 0.6}}{530^{3.825}}\right)\right]}=0.8852
\]

Recalculating the pressure at Vale, we get
\[
620 \times 10^{6}=38.77 \times 20.41\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-1.0163 \times 514.7^{2}}{0.6 \times 530 \times 141.74 \times 0.8852}\right)^{0.5} 29^{2.5}
\]

Solving for the upstream pressure at Vale, we get
\[
P_{1}=1104.88 \mathrm{psia}=1090.18 \mathrm{psig}
\]

Compared to the last calculated value, the difference is: \(1090.18-1107.79=-17.61\) psig or \(1.6 \%\). One more iteration would get us closer to the correct value. We recalculate the new average pressure using Equation 2.14 as
\[
P_{\mathrm{avg}}=\frac{2}{3}\left(1104.88+514.7-\frac{1104.88 \times 514.7}{1104.88+514.7}\right)=845.63 \mathrm{psia}=830.93 \mathrm{psig}
\]

We recalculate the compressibility factor, using the recently calculated average pressure, as
\[
Z=\frac{1}{\left[1+\left(\frac{(845.63-14.7) \times 344,400(10)^{1.785 \times 0.6}}{530^{3.825}}\right)\right]}=0.8865
\]

Recalculating the pressure at Vale, we get
\[
620 \times 10^{6}=38.77 \times 20.41\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-1.0163 \times 514.7^{2}}{0.6 \times 530 \times 141.74 \times 0.8865}\right)^{0.5} 29^{2.5}
\]

Solving for the upstream pressure at Vale, we get
\[
P_{1}=1106.15 \mathrm{psia}=1091.45 \mathrm{psig}
\]

Compared to the last calculated value, the difference is: \(1090.18-1091.45=-1.27\) psig or \(0.12 \%\). This is close enough, and no further iteration is needed.

Therefore, the pressure at Vale \(=1106.15 \mathrm{psia}=1091.45 \mathrm{psig}\).
Next, using this as the downstream pressure for the pipe segment from Rockport to Vale, we calculate the upstream pressure at Rockport as follows, for a flow rate of 420 MMSCFD.

From previous calculations at 420 MMSCFD (Example 1), the Reynolds number is
\[
R=14,671,438
\]

The friction factor was calculated as
\[
f=0.0097
\]
and the transmission factor was
\[
F=\frac{2}{\sqrt{0.0097}}=20.33
\]

To calculate the compressibility factor \(Z\), the average pressure is required. Since the inlet pressure at Rockport is unknown, we will calculate an approximate value of \(Z\) using a value of \(110 \%\) of the downstream pressure at Vale for the average pressure.

The average pressure is
\[
P_{\text {avg }}=1.1 \times 1106.15=1216.77 \mathrm{psia}=1202.07 \mathrm{psig}
\]

Using this, we calculate the compressibility factor as
\[
Z=\frac{1}{\left[1+\left(\frac{(1216.77-14.7) \times 344,400(10)^{1.785 \times 0.6}}{530^{3.825}}\right)\right]}=0.8437
\]

Since there is an elevation difference of \(200(450-250)\) ft between Rockport and Vale, the elevation correction according to Equation 2.7 must be applied.

Using Equation 2.10, the elevation adjustment parameter is
\[
s=0.0375 \times 0.6\left(\frac{450-250}{530 \times 0.8437}\right)=0.0101
\]

The equivalent length \(L_{e}\) from Equation 2.9 is
\[
L_{e}=\frac{100(1.0101-1)}{0.0101}=100 \mathrm{mi}
\]

Calculating the pressure at Rockport, we get
\[
420 \times 10^{6}=38.77 \times 20.33\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-1.0101 \times 1106.15^{2}}{0.6 \times 530 \times 100 \times 0.8437}\right)^{0.5} 29^{2.5}
\]

Solving for the upstream pressure at Rockport, we get
\[
P_{1}=1238.04 \mathrm{psia}=1223.34 \mathrm{psig}
\]

Next, we will compare the isothermal calculation results with thermal hydraulics using the GASMOD program. Given below is the output report from the GASMOD program.
```

******** GASMOD - GAS PIPELINE HYDRAULIC SIMULATION
************ 32-bit Version 5.00.100 *************

```
```

DATE: 17-September-2004 TIME: 07:16:19
PROJECT DESCRIPTION:
Pipeline from Rockport to Concord
Case number:
1143
Pipeline data file: C:\GASMOD32\RockportPipeline.TOT
Pressure drop formula: Colebrook-White
Pipeline efficiency: 1.00
Compressibility factor method: CNGA
Inlet gas gravity (air=1.0): 0.60000
Inlet gas viscosity:
0.0000080(lb/ft-sec)
CALCULATION OPTIONS:
Branch pipe calculations: YES
Loop pipe calculations: NO
Compressor fuel calculated: NO
Joule-Thompson effect included: NO
Customized output: No
Holding delivery pressure
at terminus

```
**** Calculations Based on Specified Thermal Conductivities of Pipe, Soil, and Insulation ****
\begin{tabular}{ll} 
Origin suction temperature: & \(70.00(\mathrm{degF})\) \\
Base temperature: & \(60.00(\mathrm{degF})\) \\
Base pressure: & \(14.700(\mathrm{psig})\) \\
Origin suction pressure: & \(800.00(\mathrm{psig})\) \\
Delivery pressure: & \(499.66(\mathrm{psig})\) \\
Minimum pressure: & \(100.00(\mathrm{psig})\) \\
Gas specific heat ratio: & 1.29 \\
Maximum gas velocity: & \(50.00(\mathrm{ft} / \mathrm{sec})\) \\
& \\
Inlet flow rate: & \(420.0000(\) MMSCFD \\
Outlet flow rate: & \(620.0000(\) MMSCFD
\end{tabular}
\begin{tabular}{ccccc}
\(* * * * * * * * * * * * * * * * ~ P I P E L I N E ~\) & PROFILE DATA & ********** \\
\begin{tabular}{c} 
Distance \\
(mi)
\end{tabular} & \begin{tabular}{c} 
Elevation \\
\((\mathrm{ft})\)
\end{tabular} & \begin{tabular}{c} 
Diameter \\
(in)
\end{tabular} & \begin{tabular}{c} 
Thickness \\
(in)
\end{tabular} & \begin{tabular}{c} 
Roughness \\
(in)
\end{tabular} \\
0.00 & 250.00 & 30.000 & 0.500 & 0.000700 \\
10.00 & 300.00 & 30.000 & 0.500 & 0.000700 \\
20.00 & 200.00 & 30.000 & 0.500 & 0.000700 \\
30.00 & 320.00 & 30.000 & 0.500 & 0.000700 \\
40.00 & 400.00 & 30.000 & 0.500 & 0.000700 \\
50.00 & 375.00 & 30.000 & 0.500 & 0.000700 \\
60.00 & 410.00 & 30.000 & 0.500 & 0.000700 \\
70.00 & 430.00 & 30.000 & 0.500 & 0.000700 \\
100.00 & 450.00 & 30.000 & 0.500 & 0.000700 \\
120.00 & 500.00 & 30.000 & 0.500 & 0.000700 \\
140.00 & 400.00 & 30.000 & 0.500 & 0.000700 \\
150.00 & 600.00 & 30.000 & 0.500 & 0.000700 \\
170.00 & 700.00 & 30.000 & 0.500 & 0.000700 \\
190.00 & 710.00 & 30.000 & 0.500 & 0.000700 \\
200.00 & 720.00 & 30.000 & 0.500 & 0.000700 \\
220.00 & 750.00 & 30.000 & 0.500 & 0.000700 \\
240.00 & 800.00 & 30.000 & 0.500 & 0.000700
\end{tabular}
\begin{tabular}{rcccccc} 
Distance & Cover & \multicolumn{5}{c}{\begin{tabular}{c} 
Thermal Conductivity \\
(mi)
\end{tabular}} \\
& (in) & Pipe & Soil & Insulation & (in) & (degF) \\
0.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
10.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
20.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
30.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
40.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
50.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
60.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
70.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
100.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
120.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
140.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
150.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
170.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
190.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
200.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
220.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00 \\
240.000 & 36.000 & 29.000 & 0.800 & 0.020 & 0.000 & 60.00
\end{tabular}
\begin{tabular}{|c|c|c|c|c|c|c|c|c|c|}
\hline \multicolumn{3}{|l|}{****************} & COMPRESSOR & \multicolumn{2}{|l|}{STATION DATA} & \multicolumn{3}{|l|}{A \(* * * * * * * * * * * * * * ~\)} & \\
\hline FLOW RAT & S, PRESS & RES, & TEMPER & TURES : & & & & & \\
\hline Name & \begin{tabular}{l}
Flow \\
Rate \\
(MMSCFD)
\end{tabular} & \begin{tabular}{l}
Suct. \\
Press. \\
(psig)
\end{tabular} & \begin{tabular}{l}
Disch. \\
Press. \\
(psig)
\end{tabular} & \begin{tabular}{l}
Compr . \\
Ratio
\end{tabular} & \begin{tabular}{l}
Suct. \\
Loss. \\
(psig)
\end{tabular} & \begin{tabular}{l}
Disch. \\
Loss. \\
(psig)
\end{tabular} & \begin{tabular}{l}
Suct. \\
Temp. \\
(degF)
\end{tabular} & \begin{tabular}{l}
Disch. \\
Temp \\
(degF)
\end{tabular} & \begin{tabular}{l}
MaxPipe \\
Temp \\
(degF)
\end{tabular} \\
\hline Rockport & 420.00 & 800.00 & 1223.97 & 1.5204 & 0.00 & 0.00 & 70.00 & 132.24 & 140.00 \\
\hline
\end{tabular}
\begin{tabular}{|c|c|c|c|c|c|c|c|}
\hline \multirow[t]{2}{*}{******} & \multirow[t]{2}{*}{COMPRESS} & \multicolumn{3}{|l|}{R EFFICIENCY,} & \multirow[t]{2}{*}{AND FUEL} & \multicolumn{2}{|l|}{USED *********} \\
\hline & & Compr & Mech. & Overall & & Fuel & Fuel \\
\hline Name & \begin{tabular}{l}
Distance \\
(mi)
\end{tabular} & \begin{tabular}{l}
Effy. \\
(\%)
\end{tabular} & \begin{tabular}{l}
Effy. \\
(\%)
\end{tabular} & \begin{tabular}{l}
Effy. \\
(\%)
\end{tabular} & Horse Power & \begin{tabular}{l}
Factor \\
(MCF/day/HP)
\end{tabular} & \begin{tabular}{l}
Used \\
(MMSCFD)
\end{tabular} \\
\hline Rockport & 0.00 & 80.00 & 98.00 & 78.40 & 9,513.40 & 0.2000 & ------ \\
\hline
\end{tabular}

****** REYNOLD'S NUMBER AND HEAT TRANSFER COEFFICIENT
\begin{tabular}{cccccc}
\begin{tabular}{c} 
Distance \\
(mi)
\end{tabular} & Reynold'sNum. & \begin{tabular}{c} 
FrictFactor \\
(Darcy)
\end{tabular} & \begin{tabular}{c} 
Transmission \\
Factor
\end{tabular} & \begin{tabular}{c} 
HeatTransCoeff \\
\((\) Btu/hr/ \\
ft2/degF)
\end{tabular} & \begin{tabular}{c} 
Compressibility \\
Factor \\
\((\) CNGA)
\end{tabular} \\
0.000 & \(14,671,438\). & 0.0099 & 20.13 & 0.3361 & \\
10.000 & \(14,671,438\). & 0.0099 & 20.13 & 0.3361 & 0.8820 \\
20.000 & \(14,671,438\). & 0.0099 & 20.13 & 0.3361 & 0.8678 \\
30.000 & \(14,671,438\). & 0.0099 & 20.13 & 0.3361 & 0.8578 \\
40.000 & \(14,671,438\). & 0.0099 & 20.13 & 0.3361 & 0.8516 \\
50.000 & \(14,671,438\). & 0.0099 & 20.13 & 0.3361 & 0.8479 \\
60.000 & \(14,671,438\). & 0.0099 & 20.13 & 0.3361 & 0.8460 \\
70.000 & \(14,671,438\). & 0.0099 & 20.13 & 0.3361 & 0.8454 \\
100.000 & \(21,657,838\). & 0.0098 & 20.25 & 0.3365 & 0.8465 \\
120.000 & \(21,657,838\). & 0.0098 & 20.25 & 0.3365 & 0.8516 \\
140.000 & \(21,657,838\). & 0.0098 & 20.25 & 0.3365 & 0.8589 \\
150.000 & \(21,657,838\). & 0.0098 & 20.25 & 0.3365 & 0.8650 \\
170.000 & \(21,657,838\). & 0.0098 & 20.25 & 0.3365 & 0.8721 \\
190.000 & \(21,657,838\). & 0.0098 & 20.25 & 0.3365 & 0.8821 \\
200.000 & \(21,657,838\). & 0.0098 & 20.25 & 0.3365 & 0.8903 \\
220.000 & \(21,657,838\). & 0.0098 & 20.25 & 0.3365 & 0.8998 \\
240.000 & \(21,657,838\). & 0.0098 & 20.25 & 0.3365 & 0.9148
\end{tabular}

\section*{PIPELINE TEMPERATURE AND PRESSURE PROFILE}
\begin{tabular}{|c|c|c|c|c|c|c|c|c|}
\hline Distance (mi) & \[
\begin{gathered}
\text { Diameter } \\
\text { (in) }
\end{gathered}
\] & \begin{tabular}{l}
Flow \\
(MMSCFD)
\end{tabular} & \begin{tabular}{l}
Velocity \\
(ft/sec)
\end{tabular} & Press. (psig) & GasTemp. (degF) & SoilTemp. (degF) & \begin{tabular}{l}
MAOP \\
(psig)
\end{tabular} & Location \\
\hline 0.00 & 30.000 & 420.0000 & 12.60 & 1223.97 & 132.24 & 60.00 & 1400.00 & Rockport \\
\hline 10.00 & 30.000 & 420.0000 & 12.60 & 1208.59 & 107.14 & 60.00 & 1400.00 & \\
\hline 20.00 & 30.000 & 420.0000 & 12.60 & 1197.97 & 90.16 & 60.00 & 1400.00 & \\
\hline 30.00 & 30.000 & 420.0000 & 12.60 & 1181.38 & 79.02 & 60.00 & 1400.00 & \\
\hline 40.00 & 30.000 & 420.0000 & 12.60 & 1166.06 & 71.88 & 60.00 & 1400.00 & \\
\hline 50.00 & 30.000 & 420.0000 & 12.60 & 1153.87 & 67.38 & 60.00 & 1400.00 & \\
\hline 60.00 & 30.000 & 420.0000 & 12.60 & 1139.88 & 64.56 & 60.00 & 1400.00 & \\
\hline 70.00 & 30.000 & 420.0000 & 12.60 & 1126.25 & 62.81 & 60.00 & 1400.00 & \\
\hline 100.00 & 30.000 & 620.0000 & 18.60 & 1085.71 & 60.65 & 60.00 & 1400.00 & \\
\hline 120.00 & 30.000 & 620.0000 & 18.60 & 1023.96 & 60.34 & 60.00 & 1400.00 & \\
\hline 140.00 & 30.000 & 620.0000 & 18.60 & 961.88 & 60.17 & 60.00 & 1400.00 & \\
\hline 150.00 & 30.000 & 620.0000 & 18.60 & 922.84 & 60.13 & 60.00 & 1400.00 & \\
\hline 170.00 & 30.000 & 620.0000 & 18.60 & 847.15 & 60.06 & 60.00 & 1400.00 & \\
\hline 190.00 & 30.000 & 620.0000 & 18.60 & 765.54 & 60.03 & 60.00 & 1400.00 & \\
\hline 200.00 & 30.000 & 620.0000 & 18.60 & 720.84 & 60.02 & 60.00 & 1400.00 & \\
\hline 220.00 & 30.000 & 620.0000 & 18.60 & 620.90 & 60.01 & 60.00 & 1400.00 & \\
\hline 240.00 & 30.000 & 620.0000 & 18.60 & 499.66 & 60.01 & 60.00 & 1400.00 & Concord \\
\hline
\end{tabular}

\section*{************ LINE PACK VOLUMES AND PRESSURES}
\begin{tabular}{rcc}
\begin{tabular}{c} 
Distance \\
(mi)
\end{tabular} & \begin{tabular}{c} 
Pressure \\
(psig)
\end{tabular} & \begin{tabular}{c} 
Line Pack \\
(million std.cu.ft)
\end{tabular} \\
0.00 & 1223.97 & 0.0000 \\
10.00 & 1208.59 & 21.2972 \\
20.00 & 1197.97 & 22.0280 \\
30.00 & 1181.38 & 22.4450 \\
40.00 & 1166.06 & 22.5706 \\
50.00 & 1153.87 & 22.5714 \\
60.00 & 1139.88 & 22.4716 \\
70.00 & 1126.25 & 22.2810 \\
100.00 & 1085.71 & 65.4581 \\
120.00 & 1023.96 & 41.3973 \\
140.00 & 961.88 & 38.6813 \\
150.00 & 922.84 & 18.2370 \\
170.00 & 847.15 & 34.0291 \\
190.00 & 765.54 & 30.7118 \\
200.00 & 720.84 & 14.0358 \\
220.00 & 620.90 & 25.1637 \\
240.00 & 499.66 & 20.7997
\end{tabular}

Total line pack in main pipeline \(=444.1785\) (million std.cu.ft)
\begin{tabular}{|c|c|c|c|c|c|c|c|c|}
\hline \multicolumn{9}{|l|}{\multirow[t]{2}{*}{\begin{tabular}{l}
NUMBER OF PIPE BRANCHES = 1 \\
BRANCH TEMPERATURE AND PRESSURE PROFILE:
\end{tabular}}} \\
\hline & & & & & & & & \\
\hline \multicolumn{9}{|l|}{\multirow[t]{2}{*}{Incoming Branch File: VALEBRANCH.TOT Branch Location: at 100 (mi)}} \\
\hline & & & & & & & & \\
\hline \multicolumn{9}{|l|}{Distance Elevation Diameter Flow Velocity Press. Gas Temp. Amb Temp. Location (mi) (ft) (in) (MMSCFD) (ft/sec) (psig) (degF) (degF)} \\
\hline 0.00 & 150.00 & 24.000 & 200.000 & 9.81 & 1163.79 & 70.00 & 60.00 & Drake \\
\hline 10.00 & 100.00 & 24.000 & 200.000 & 9.81 & 1156.06 & 63.97 & 60.00 & \\
\hline 20.00 & 125.00 & 24.000 & 200.000 & 9.96 & 1146.13 & 61.56 & 60.00 & \\
\hline 40.00 & 200.00 & 24.000 & 200.000 & 10.14 & 1125.37 & 60.24 & 60.00 & \\
\hline 50.00 & 200.00 & 24.000 & 200.000 & 10.23 & 1115.97 & 60.09 & 60.00 & \\
\hline 70.00 & 200.00 & 24.000 & 200.000 & 10.40 & 1096.89 & 60.01 & 60.00 & \\
\hline 80.00 & 250.00 & 24.000 & 200.000 & 10.51 & 1085.79 & 60.01 & 60.00 & Vale \\
\hline \multicolumn{9}{|l|}{*************** End of GASMOD Output Report \(* * * * * * * * * * * * * ~\)} \\
\hline
\end{tabular}

It can be seen from the GASMOD thermal hydraulic analysis report that the inlet pressure at Rockport is approximately 1224 psig, whereas the manual calculation considering isothermal flow yielded an inlet pressure of approximately 1223 psig at the Rockport compressor station. This difference is not very significant. However, in many cases the temperature variation along the pipeline will cause pressures calculated to be significantly different, especially in short pipelines. As an example, if we had a pipeline 100 mi long similar to the pipe section between Rockport and Vale, the thermal hydraulics will show a drastic temperature variation, from 132.24 to \(60.65^{\circ} \mathrm{F}\). Therefore, an isothermal analysis at \(70^{\circ} \mathrm{F}\) for the entire 100 mi length will show considerable discrepancy in pressures. This is left as an exercise for the reader.

To recap, the manual calculations were based on an isothermal gas flow temperature of \(70^{\circ} \mathrm{F}\), whereas the thermal hydraulics shows variation of the gas temperature ranging from \(132.24^{\circ} \mathrm{F}\) at the Rockport compressor discharge to \(60.01^{\circ} \mathrm{F}\) at Concord, which is very close to the surrounding soil temperature of \(60^{\circ} \mathrm{F}\).

The compression ratio is 1.52 at the Rockport compressor station, where the 800 psig inlet pressure of the gas is increased to the discharge pressure of 1224 psig . This, in accordance with our previous analysis in Chapter 4 under compressors, causes the discharge temperature of the gas to increase to \(132.24^{\circ} \mathrm{F}\). If the compression ratio were higher, the discharge temperature of the gas due to compression would have been still higher. The pipeline coating temperature limitation is \(140^{\circ} \mathrm{F}\) and would then require gas cooling in order to avoid damage to the pipe coating. It can be seen from the GASMOD report that the gas flow temperature starts off at \(132.24^{\circ} \mathrm{F}\) at milepost 0 (Rockport) and quickly drops to \(67.38^{\circ} \mathrm{F}\) at milepost 50 . This is the exponential temperature decay we discussed in an earlier section. Beginning at milepost 50, the gas temperature starts dropping off more gradually until it almost attains the soil temperature at milepost 240 (Concord). Also, the section of pipe between milepost 170 and Concord is at a fairly constant temperature, close to the soil temperature of \(60^{\circ} \mathrm{F}\). Therefore, the 70 mi pipeline section between milepost 170 and Concord can be considered to be in isothermal flow, for all practical purposes. A manual calculation of this last 70 mi pipe segment flowing
at 620 MMSCFD will yield a pressure profile very close to the pressures shown in the GASMOD report. This will be demonstrated next.

First, we calculate the Reynolds number from Equation 2.34:
\[
R=0.0004778\left(\frac{14.7}{60+460}\right)\left(\frac{0.6 \times 620 \times 10^{6}}{0.000008 \times 29}\right)=21,657,837
\]

Next, using Colebrook Equation 2.39, we calculate the friction factor as
\[
\frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{0.0007}{3.7 \times 29}+\frac{2.51}{21,657,837 \sqrt{f}}\right)
\]

Solving by trial and error, we get
\[
f=0.0096
\]

Therefore, the transmission factor is, using Equation 2.42,
\[
F=\frac{2}{\sqrt{0.0096}}=20.41
\]

To calculate the compressibility factor \(Z\), the average pressure is required. Since the pressure at milepost 170 is unknown, we will calculate an approximate value of \(Z\) using a value of 850 psig for the average pressure.

Using CNGA Equation 1.34, we calculate the value of the compressibility factor as
\[
Z=\frac{1}{\left[1+\left(\frac{850 \times 344,400(10)^{1.785 \times 0.6}}{520^{3.825}}\right)\right]}=0.8765
\]

Since there is an elevation difference of \(100(800-700) \mathrm{ft}\) between milepost 170 and Concord, we must apply the elevation correction according to Equation 2.7.

Using Equation 2.10, the elevation adjustment parameter is
\[
s=0.0375 \times 0.6\left(\frac{800-700}{520 \times 0.8765}\right)=0.0049
\]

The equivalent length \(L_{e}\) from Equation 2.9 is
\[
L_{e}=\frac{70(1.0049-1)}{0.0049}=70.17 \mathrm{mi}
\]

Next, using Equation 2.7, we calculate the pressure at milepost 70 as follows:
\[
\begin{gathered}
620 \times 10^{6}=38.77 \times 20.41\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-1.0049 \times 514.7^{2}}{0.6 \times 520 \times 70.17 \times 0.8765}\right)^{0.5} 29^{2.5} \\
P_{1}=851.59 \mathrm{psia}=836.89 \mathrm{psig}
\end{gathered}
\]

This compares with 850 psig we assumed earlier. Recalculating the average pressure based on \(P_{1}=851.59 \mathrm{psia}\) and \(P_{2}=514.7 \mathrm{psia}\), we get, using Equation 2.14,
\[
P_{\text {avg }}=\frac{2}{3}\left(851.59+514.7-\frac{851.59 \times 514.7}{851.59+514.7}\right)=696.99 \mathrm{psia}
\]

Next, using CNGA Equation 1.34, we recalculate the compressibility factor as
\[
Z=\frac{1}{\left[1+\left(\frac{682.29 \times 344,400\left(100^{1.785 \times 0.6}\right.}{520^{3.825}}\right)\right]}=0.8984
\]

Next, using Equation 2.7, we calculate the pressure at milepost 70 as follows:
\[
\begin{gathered}
620 \times 10^{6}=38.77 \times 20.41\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-1.0049 \times 514.7^{2}}{0.6 \times 520 \times 70.17 \times 0.8984}\right)^{0.5} 29^{2.5} \\
P_{1}=858.30 \mathrm{psia}=843.6 \mathrm{psig}
\end{gathered}
\]

This value is less than \(1 \%\) different from the previously calculated value of 836.39 psig. Therefore, we do not have to iterate any further. Comparing the value of 843.6 psig with the pressure of approximately 847 psig from the GASMOD report, we see that we are less than \(0.5 \%\) apart. Thus, the assumption of isothermal flow in the last 70 mi section of the pipeline is a valid one. We would have been closer still if the 70 mi section had been subdivided into two or more segments and the upstream pressures calculated as discussed in an earlier section.

In conclusion, we can state that calculating the pressures and HP in a gas pipeline based on the assumption of constant temperature throughout the pipeline will yield satisfactory answers if the pipeline is long. For shorter pipelines, calculations must be performed by subdividing the pipeline into short segments and taking into account heat transfer between the pipeline gas and the surrounding soil.

\subsection*{7.4 SUMMARY}

In this chapter we reviewed the thermal effects of pressure drop and horsepower required in a natural gas pipeline system. We pointed out the differences between the results obtained from isothermal and thermal hydraulic analysis. This was illustrated with example problems using an isothermal analysis compared to a more rigorous approach considering heat transfer between the pipeline gas and the surrounding soil. A popular gas pipeline hydraulic simulation software application was used to illustrate the calculation methodology.

\section*{PROBLEMS}
1. Apply the technique discussed in the temperature variation calculation section to calculate the temperature profile of a gas pipeline 4 mi long, NPS 20, with 0.375 in wall thickness, at a flow rate of 200 MMSCFD.
2. A 200 mile, NPS 24, 0.500 inch wall thickness pipeline from Mobile to Savannah is used for transporting 300 MMSCFD of natural gas (gravity \(=0.65\) and viscosity \(=0.000008 \mathrm{lb} / \mathrm{ft}-\mathrm{s}\) ). The MOP is 1400 psig . The gas inlet temperature and pressure at Mobile are \(80^{\circ} \mathrm{F}\) and 1200 psig , respectively. The soil temperature can be assumed to be \(60^{\circ} \mathrm{F}\) throughout. The delivery pressure required at Savannah is 900 psig. Assume isothermal flow at \(70^{\circ} \mathrm{F}\). Using the Panhandle B equation with an efficiency of 0.95 , calculate the free flow volume with no compressor stations. Compare these results with thermal hydraulic analysis using subdivided pipe segments and heat transfer calculations. The base pressure is 14.7 psia and base temperature is \(60^{\circ} \mathrm{F}\).

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\section*{CHAPTER 8}

\section*{Transient Analysis and Case Studies}

\begin{abstract}
This chapter is devoted to transient pressure analysis and case studies of typical longdistance gas transmission pipelines. The subject of transient pressure analysis is quite complex, and understanding the theory behind it requires delving into differential equations and solution by the method of characteristics. Further, these calculations require some form of computer simulation to arrive at meaningful results. Nevertheless, we will discuss several scenarios that are typical of unsteady flow in gas pipelines that cause transient conditions. The objective is to determine how the pressure varies along the pipeline due to disturbances caused by transient conditions, such as a mainline valve closure and compressor station shutdown. If these transient conditions cause the pipeline pressures at some points to exceed the MOP, measures must be provided to ensure that the pressures do not violate the limits allowed by design codes. For detailed analysis of transient pressures in gas pipelines, the reader should refer to the texts listed in the Reference section of this chapter. We will also be reviewing several real-world pipeline transportation scenarios that encompass the concepts covered in the preceding chapters.
\end{abstract}

\subsection*{8.1 UNSTEADY FLOW}

In the preceding chapters, we analyzed pipelines that were in steady-state flow. This means that, at any point in time, the pipelines were operating at constant flow rates with a constant pressure and temperature profile. In other words, if the pressures, temperatures, and flow rates were measured at some point in time, say at 10 a.m. on a certain day, these parameters persisted in values throughout the period under investigation. Therefore, at some other time, such as 12 noon or 5 p.m., the pressure profile, the temperature profile, and the gas flow rates were all the same as that at 10 a.m. In reality, this does not happen. Due to one reason or another, the flow rates and pressures tend to change. This might be due to changes in delivery conditions, such as variation in the amount of gas being received at delivery stations due to changes in operation of facilities that require the gas. Further environmental conditions, such as
atmospheric temperatures, can cause variation in compressor performance, resulting in flow and pressure changes at the compressor discharge. The latter will, in turn, cause changes in pressure and temperature of the pipeline gas. Another reason for unsteady flow condition can be a result of switching or closing valves to divert flow to different customers or shutdown or startup of compressor stations. Such unsteady operation causes transient pressures in the pipeline. Under unsteady or transient flow, the pressures, temperatures, and flow rates become time dependent. This means that it will no longer be correct to use results based upon steady-state flow calculations.

\subsection*{8.1.1 Transient Due to Mainline Valve Closure}

To illustrate a transient condition, let us review a simple pipeline system with a head compressor station and a mainline valve at the end of the pipeline, as shown in Figure 8.1.

The pipeline has been operating in steady-state condition for a long time. The valve at the end of the pipeline is suddenly closed due to malfunction or human error. Immediately, the pressure at the valve and at points upstream of it starts to rise, as shown by the dashed lines in Figure 8.1. Since gas is compressible, the compressor at the upstream end continues to pump gas without sensing the pressure rise downstream. This will result in an increase in line pack in the downstream section of the pipe, which will progress toward the upstream end. The transient pressure waves moving upstream will eventually reach the discharge of the compressor, causing the discharge pressure to rise. If the increased pressure attains the discharge shutdown setting, the compressor will trip and shut down, producing no further pumping pressure. The blocked-in gas in the pipeline will continue to undergo pressure variation from upstream to downstream as the pressure waves go back and forth at the speed of sound in gas. Eventually, the pressure surge dies out because of friction and loss of inertia resulting from reduction in gas velocity. This is illustrated in another


Figure 8.1 Transient due to valve closure.


Figure 8.2 Compressor performance curve vs. pipeline system head curve.
way in Figure 8.2, using the compressor performance curve and the pipeline system curve.

The steady-state system head curve is represented by AB and the compressor performance curve by CD . The steady-state operating point is therefore at E , where the compressor head H matches the pipeline system head required at the flow rate \(Q\). As the valve at the end of the pipeline is closed, the system head curve shifts to the left, indicating reduction in gas flow due to increased pipeline resistance caused by the constriction in the valve.

\subsection*{8.1.2 Transient Due to Compressor Shutdown}

Another transient condition that can occur in a simple pipeline described in Figure 8.2 is that of a compressor station shutdown from a steady-state operating condition. Suppose the compressor shuts down in 30 seconds after a long period of steady-state flow. Since there is no pressure being generated at the upstream end of the pipeline, but gas continues to be delivered at the downstream end, the line pack in the pipeline starts reducing starting at the downstream end. The pressures continue to fall along the pipeline and eventually stabilize at some blocked-in pressure.

Another slightly complicated compressor station shutdown scenario that causes transient pressures is illustrated in Figure 8.3. In this case, a pipeline with two compressor stations is shown with a hydraulic pressure gradient under steady-state conditions. If the intermediate compressor station shuts down and the gas continues to be pumped from the first compressor station bypassing the second compressor station, the hydraulic pressure gradient will eventually be as indicated in Figure 8.3. However, before steady-state conditions are achieved with only one compressor operating, transient pressures are developed from the point of shutdown of the intermediate compressor station. Suppose the initial flow rate with both compressors operating under steady-state conditions is indicated by a flow rate of \(Q\). The compressor


NPS 16 pipeline 200 mi long

Figure 8.3 Transient due to compressor station shutdown.
performance curve superimposed on the pipeline system head curve for the pipe segment between the two compressor stations is shown in Figure 8.4.

Initially, the system head curve \(A B\) for the pipe segment 1 between the two compressor stations results in a flow rate of \(Q\) with the operating point at E . At this point, the compressor head H of the first compressor station matches the pipeline system head required at the flow rate \(Q\). When the second compressor station shuts down, it no longer provides the discharge pressure to boost the gas in pipe segment 2 . Therefore, the first compressor station has to push the gas all the way to the end of the pipeline. It therefore has to contend with a longer pipe segment, which has a system head curve FG as shown in Figure 8.4. It can be seen that the new operating point K is at


Figure 8.4 Transient due to intermediate compressor shutdown.
a reduced flow rate \(Q_{1}\). If the point K on the compressor head curve is at too high a discharge pressure, the control mechanism will signal the compressor to slow down in speed. Thus, if the original compressor curve was based on \(15,000 \mathrm{rpm}\), the compressor would slow down to a speed such as \(12,000 \mathrm{rpm}\). This results in a new operating point, L, corresponding to a flow rate \(Q_{2}\), as shown in Figure 8.4. In summary, shutting down the second compressor station causes the operating point to move from point E on the compressor head curve at \(15,000 \mathrm{rpm}\) down to point L on the compressor head curve at \(12,000 \mathrm{rpm}\). Correspondingly, the flow rate will decrease from \(Q\) at point E to \(Q_{2}\) at point L .

\subsection*{8.2 CASE STUDIES}

In the next few pages of this chapter, we are going to look at some real-life gas transmission pipeline systems. We will be applying the concepts learned in the previous chapters to determine the pressures and flow rates required in various scenarios.

\subsection*{8.2.1 Offshore Pipeline Case}

Consider a gas production facility located offshore. The gas is compressed from the offshore platform through submarine pipelines that go ashore and subsequently connect to onshore pipelines for transportation of gas to industrial consumers. We will look at sizing such an offshore and onshore piping system for transporting a given quantity of gas. Calculations will be performed considering different options such as the AGA equation and Panhandle equations. We will illustrate this using an example.

\section*{Case Study 1—Offshore/Onshore Pipeline}

A natural gas pipeline system originates at an offshore facility that compresses the gas through 200 mi of NPS \(30,0.625 \mathrm{in}\). wall thickness submarine pipelines to an onshore location, as depicted in Figure 8.5.

A compressor station located onshore is used to compress the gas through a 120 mi , NPS 24, 0.500 inch wall thickness onshore buried pipeline for eventual delivery to a power plant. Determine the maximum flow rate possible under the following conditions. Neglect elevation effects. The compression ratio is 1.5 . Use the Weymouth equation with \(95 \%\) efficiency. Assume a base pressure of 14.7 psia and base temperature of \(60^{\circ} \mathrm{F}\). The gas flowing temperature is \(60^{\circ} \mathrm{F}\) and the compressibility factor is 0.88 . The gas gravity is 0.65 .


Figure 8.5 Offshore/onshore pipeline.
(a) Gas pressure at the platform equals 1480 psig and free flow occurs without use of any compression offshore or onshore. The delivery pressure at the power plant is 500 psig.
(b) Considering the MOP at the platform and onshore equal to 1480 psig , determine the maximum throughput possible with compression offshore and onshore.

Solution
(a) Free flow with 1480 psig at the offshore platform.

Using Weymouth Equation 2.52, neglecting elevation effects, calculate the pressure at the beginning of the NPS 24 onshore pipeline. The pipe inside diameter \(=24-\) \(2 \times 0.500=23 \mathrm{in}\).
\[
\begin{equation*}
Q=433.5 \times 0.95\left(\frac{60+460}{14.7}\right)\left(\frac{P_{1}^{2}-514.7^{2}}{0.65 \times 520 \times 120 \times 0.88}\right)^{0.5} 23^{2.667} \tag{8.1}
\end{equation*}
\]

Similarly, considering the NPS 30 pipeline 200 mi long,

Pipe inside diameter \(=30-2 \times 0.625=28.75\) in.
\[
\begin{equation*}
Q=433.5 \times 0.95\left(\frac{60+460}{14.7}\right)\left(\frac{1494.7^{2}-P_{1}^{2}}{0.65 \times 520 \times 200 \times 0.88}\right)^{0.5} 28.75^{2.667} \tag{8.2}
\end{equation*}
\]

Eliminating \(Q\) from both Equation 8.1 and Equation 8.2 by division, we get
\[
\begin{equation*}
1=\left(\frac{P_{1}^{2}-514.7^{2}}{1494.7^{2}-P_{1}^{2}}\right)^{0.5}\left(\frac{200}{120}\right)^{0.5}\left(\frac{23.0}{28.75}\right)^{2.667} \tag{8.3}
\end{equation*}
\]

Solving for the pressure \(P_{1}\) at the junction of the two pipes onshore,
\[
P_{1}=1253.7 \mathrm{psia}=1239 \mathrm{psig}
\]

Next, substituting this value of \(P_{1}\) in Equation 8.1, we calculate the free flow volume flow rate as
\[
Q=433.5 \times 0.95\left(\frac{520}{14.7}\right)\left(\frac{1253.7^{2}-514.7^{2}}{0.65 \times 520 \times 120 \times 0.88}\right)^{0.5} 23^{2.667}
\]
or
\[
Q=377.53 \mathrm{MMSCFD}
\]

Therefore, without any compression, the free flow possible is 377.53 MMSCFD.
(b) With compressors installed at onshore and offshore locations, each location will be delivering at an MOP of 1480 psig . With a compression ratio of 1.5 , the suction pressure at the onshore compressor is
\[
P_{s}=\frac{1480+14.7}{1.5}=996.47 \mathrm{psia}=981.77 \mathrm{psig}
\]

First, calculate the capacity of the NPS 24 onshore pipeline, considering 1480 psig at the upstream end and 500 psig at the downstream end 120 mi away.

Using Weymouth Equation 2.52,
\[
\begin{aligned}
& Q=433.5 \times 0.95 \times \frac{520}{14.7}\left(\frac{1494.7^{2}-514.7^{2}}{0.65 \times 520 \times 120 \times 0.88}\right)^{0.5} 23^{2.667} \\
& Q=463.43 \text { MMSCFD }
\end{aligned}
\]

Next, we must determine if the offshore NPS 30 pipeline can transmit this flow starting at 1480 psig at the offshore platform and with a downstream pressure of 981.77 psig calculated earlier.
\[
Q=433.5 \times 0.95 \times \frac{520}{14.7}\left(\frac{1494.7^{2}-996.47^{2}}{0.65 \times 520 \times 200 \times 0.88}\right)^{0.5}(28.75)^{2.667}
\]

Solving for \(Q\), we get
\[
Q=516.76 \mathrm{MMSCFD}
\]

Thus, the NPS 30 submarine pipeline has a capacity of 516.76 MMSCFD, whereas the onshore NPS 24 pipeline has a capacity of only 463.43 MMSCFD. Picking the lower of the two flow rates, the maximum throughput possible with the onshore compressor is 463.43 MMSCFD.

\section*{Case Study 2-Gas Gathering System and Trunk Line to Power Plant}

Natural gas gathered from the San Juan gas fields is collected at Chico and transported through a DN \(800,15 \mathrm{~mm}\) wall thickness pipeline system, 420 km long, that ties into another DN \(800,15 \mathrm{~mm}\) wall thickness gas transmission pipeline at Rio for eventual delivery to a power plant at Madera, as shown in Figure 8.6.

Chico is at an elevation of 2100 m , whereas Rio and Madera are at 1650 m and 3100 m , respectively. The length of the pipeline from Rio to Madera is 280 km . The required delivery pressure at Madera is 35 Bar gauge. The gas gravity and viscosity are 0.65 and 0.012 cP , respectively. The gas inlet temperature at Chico is \(20^{\circ} \mathrm{C}\), and the pressure is 40 Bar gauge. Assume a constant gas flow temperature of \(20^{\circ} \mathrm{C}\). The pipeline MOP is 100 Bar gauge. The base temperature and base pressure are \(15^{\circ} \mathrm{C}\) and 1 Bar absolute, respectively. Use the Panhandle B equation with a pipeline efficiency of \(95 \%\). Assume a gas compressibility factor of 0.85 throughout.


Figure 8.6 Chico-Rio pipeline to Madera power plant.
(a) Determine the compressor station power required to deliver \(6 \mathrm{Mm}^{3} /\) day at Madera. Use \(80 \%\) isentropic efficiency and a specific heat ratio of 1.4.
(b) What modifications are required to provide gas volumes of \(1.5 \mathrm{Mm}^{3} /\) day for an industrial consumer at Rio in addition to that required at Madera?
(c) What pipeline capacity can be expected if all compressor stations are shut down and free flow occurs from Chico to Rio and Madera? Ignore deliveries at Rio and assume all gas flows to Madera.

\section*{Solution}

Assume initially that one compressor station at Chico will be able to transport \(6 \mathrm{Mm}^{3} /\) day to Madera.
\[
\begin{gathered}
1 \mathrm{Bar}=100 \mathrm{kPa} \\
\text { Pipe inside diameter }=800-2 \times 15=770 \mathrm{~mm}
\end{gathered}
\]

The elevation adjustment parameter from Equation 2.11 for Rio to Madera is
\[
\begin{aligned}
s & =0.0684 \times 0.65\left(\frac{3100-1650}{(20+273) 0.85}\right)=0.2589 \\
e^{s} & =e^{0.2589}=1.2954
\end{aligned}
\]

The equivalent length from Equation 2.9 is
\[
L_{e}=280 \times \frac{(1.2954-1)}{0.2589}=319.52 \mathrm{~km}
\]

Using Panhandle B Equation 2.60, considering elevation difference, first for the Rio to Madera pipe segment, we get
\[
\begin{aligned}
6 \times 10^{6}= & 1.002 \times 10^{-2} \times 0.95 \times\left(\frac{15+273}{100}\right)^{1.02} \\
& \times\left(\frac{P_{1}^{2}-1.2954(3600)^{2}}{0.65^{0.961} \times 293 \times 319.52 \times 0.85}\right)^{0.51} 770^{2.53}
\end{aligned}
\]

Solving for the pressure at Rio,
\[
P_{1}=4818 \mathrm{kPa}
\]

Next, using this pressure as the downstream pressure for the 420 km pipe segment from Chico to Rio, we get
\[
\text { Elevation adjustment parameter } s=0.0684 \times 0.65\left(\frac{1650-2100}{(20+273) 0.85}\right)=-0.0803
\]
\[
e^{s}=e^{-0.0803}=0.9228
\]

The equivalent length is
\[
\begin{aligned}
L_{e}= & 420 \times \frac{(0.9228-1)}{-0.0803}=403.74 \mathrm{~km} \\
6 \times 10^{6}= & 1.002 \times 10^{-2} \times 0.95 \times\left(\frac{15+273}{100}\right)^{1.02} \\
& \times\left(\frac{P_{1}^{2}-0.9228 \times 4818^{2}}{0.65^{0.961} \times 293 \times 403.74 \times 0.85}\right)^{0.51} 770^{2.53}
\end{aligned}
\]

Solving for the upstream pressure at Chico, we get
\[
P_{1}=5435 \mathrm{kPa}=54.35 \text { Bar absolute }
\]

Since the inlet pressure at Chico is 40 Bar gauge, the compression ratio required at Chico is
\[
\text { Compression ratio }=\frac{54.35}{40+1}=1.33
\]

The compressor station power required is calculated from Equation 4.16 as follows:
\[
\text { Power }=4.0639 \times 6\left(\frac{1.40}{0.4}\right)(293)\left(\frac{1+0.85}{2}\right)\left(\frac{1}{0.8}\right)\left[(1.33)^{\frac{0.4}{1.4}}-1\right]
\]
or
\[
\text { Power }=2455 \mathrm{KW}
\]

When the Rio delivery of \(1.5 \mathrm{Mm}^{3} / \mathrm{day}\) is included, we calculate the upstream pressure at Chico for the 420 km segment as follows:
\[
\begin{aligned}
7.5 \times 10^{6}= & 1.002 \times 10^{-2} \times 0.95 \times\left(\frac{288}{100}\right)^{1.02} \\
& \times\left(\frac{P_{1}^{2}-0.9228 \times 4818^{2}}{0.65^{0.961} \times 293 \times 403.74 \times 0.85}\right)^{0.51} 770^{2.53}
\end{aligned}
\]

By proportion, we get
\[
\frac{7.5}{6.0}=\frac{P_{1}^{2}-0.9228 \times 4818^{2}}{5435^{2}-0.9228 \times 4818^{2}}
\]

Solving for \(P_{1}\), we get
\[
P_{1}=5831 \mathrm{kPa}=58.31 \mathrm{Bar} \text { absolute }
\]

The new compression ratio becomes
\[
\text { Compression ratio }=\frac{58.31}{40+1}=1.42
\]

The new power required at the Chico compressor station is
\[
\text { Power }=4.0639 \times 7.5\left(\frac{1.40}{0.4}\right)(293)\left(\frac{1+0.85}{2}\right)\left(\frac{1}{0.8}\right)\left[(1.42)^{\frac{0.4}{1.4}}-1\right]=3808 \mathrm{KW}
\]
(c) When the compressor station at Chico is shut down, the pressure available is only 40 Bar or 4000 kPa . Using this upstream pressure and considering the entire \((420+\) 280) \(\mathrm{km}=700 \mathrm{~km}\) pipeline from Chico to Madera, the free flow capability is calculated using the Panhandle B equation by considering the elevation changes in two steps.

From Equation 2.12 and Equation 2.13, for the 420 km segment the elevation falls from 2100 m to 1650 m and
\[
s=-0.0803 \text { and } e^{s}=0.9228 \text { (as calculated earlier) }
\]

From Equation 2.12,
\[
\begin{aligned}
j_{1} & =\frac{0.9228-1}{-0.0803}=0.9614 \\
L_{1} & =420 \mathrm{~km}
\end{aligned}
\]

Similarly, for the 280 km second segment of the pipeline, the elevation rises from 2100 m to 3100 m , measured from Chico.
\[
s=0.1785
\]
and
\[
\begin{aligned}
& e^{s}=1.1954 \\
& j_{2}=\frac{1.1954-1}{0.1785}=1.095 \\
& L_{2}=280 \mathrm{~km}
\end{aligned}
\]

From Equation 2.13, the equivalent length is
\[
L_{e}=0.9614 \times 420+1.095 \times 280 \times 0.9228=686.72 \mathrm{~km}
\]

For the entire line,
\[
\begin{aligned}
s & =0.0684 \times 0.65\left(\frac{3100-2100}{293 \times 0.85}\right)=0.1785 \\
e^{s} & =e^{0.1785}=1.1954
\end{aligned}
\]

Applying the Panhandle B equation for the entire pipeline, we get
\[
\begin{gathered}
Q=1.002 \times 10^{-2} \times 0.95 \times\left(\frac{288}{100}\right)^{1.02}\left(\frac{4100^{2}-1.1954 \times 3600^{2}}{0.65^{0.961} \times 293 \times 686.72 \times 0.85}\right)^{0.51} 770^{2.53} \\
Q=1,926,314 \mathrm{~m}^{3} / \text { day }=1.93 \mathrm{Mm}^{3} / \text { day }
\end{gathered}
\]

Thus, with the Chico compressor station shut down, the free flow throughput is
\[
Q=1.93 \mathrm{Mm}^{3} / \mathrm{day}
\]

Obviously, this is inadequate to feed the Madera power plant that requires \(6 \mathrm{Mm}^{3} /\) day.

\section*{Case Study 3—Fairfield to Beaumont and Travis Pipeline}

A natural gas pipeline, NPS 24, is being built from the gas fields at Fairfield (elevation 610 ft ) to transport gas to a 400 MW power plant at Beaumont (elevation 350 ft ) 280 mi away, as illustrated in Figure 8.7.

Along the way at Mavis (milepost 50, elevation 1200 ft ), an industrial consumer requires 10 MMSCFD , and a small community at Mayberry (milepost 110, elevation 1800 ft ) requires natural gas for a municipal gas distribution system with a city gate pressure of 600 psig and 20 MMSCFD . During the first 2 years of operation, the gas flow requirements are as follows:

Mavis: 10 MMSCFD at 300 psig
Mayberry: 20 MMSCFD at 600 psig
Beaumont: 100 MMSCFD at 400 psig
Total: 130 MMSCFD


Figure 8.7 Fairfield to Beaumont and Travis pipeline.

At the end of the second year, a 240 MW power plant at Travis (elevation 420 ft ) will come on stream and require a gas delivery of 60 MMSCFD at 350 psig . This requires a total pipeline capacity of 190 MMSCFD out of Fairfield. The gas pressure and temperature at the inlet to the pipeline are 500 psig and \(70^{\circ} \mathrm{F}\). The soil temperatures can be assumed to be as follows:

\author{
Fairfield to Mavis: \(60^{\circ} \mathrm{F}\) \\ Mavis to Mayberry: \(50^{\circ} \mathrm{F}\) \\ Mayberry to Beaumont: \(70^{\circ} \mathrm{F}\)
}

The branch pipe to Travis starts at the Travis junction (milepost 200, elevation 750 ft ) and extends 20 mi to the Travis power plant. It is an NPS 16, 0.250 in . wall thickness pipe. It is anticipated that API 5LX-70 material will be used for the pipe. The cost of pipe material is \(\$ 1200\) per ton for pipe coated, wrapped, and delivered to the field. Construction cost of the pipeline can be estimated at \(\$ 20,000\) per in.-diameter mi. Compressor stations cost is \(\$ 2000\) per installed \(H P\). Mainline valves are to be installed at 20 mi intervals and cost \(\$ 100,000\) per site. Receipt and delivery meters at Fairfield, Mavis, Mayberry, Travis, and Beaumont are expected to cost as follows:

Fairfield meter: \(\$ 500,000\)
Mavis meter: \$200,000
Mayberry meter: \(\$ 250,000\)
Travis meter: \(\$ 300,000\)
Beaumont meter: \(\$ 350,000\)

Fuel consumption can be estimated at 0.2 MCF per day per \(H P\). Fuel gas cost is \(\$ 4\) per MCF.

Assume base pressure \(=14.7 \mathrm{psia}\) and base temperature \(=60^{\circ} \mathrm{F}\). The MOP of the pipeline is 1440 psig. Use the General Flow equation with a Colebrook friction factor and the CNGA equation for the compressibility factor.
(a) Determine the pipe wall thickness required for the specified MOP.
(b) Determine the locations and \(H P\) of the compressor stations necessary for the first 2 years (phase 1) and after that (phase 2).
(c) Estimate the total capital cost of pipeline, compressor stations, and other facilities for phase 2.

\section*{Solution}

During the phase 1 operation, we will calculate the pressures and \(H P\) required, considering 100 MMSCFD delivery to the Beaumont power plant at 400 psig . Since the MOP is 1440 psig , the minimum wall thickness needed for the class 1 location is calculated from Equation 6.8:
\[
1440=\frac{2 t \times 70,000 \times 0.72}{24}
\]

Solving for \(t\), we get
\[
\text { Wall thickness } t=0.343 \mathrm{in} \text {. }
\]
or
Use 0.375 in. standard size pipe.
Inside diameter \(D=24-2 \times 0.375=23.25 \mathrm{in}\).
First, calculate the upstream pressure at milepost 110, assuming a gas flow temperature at \(70^{\circ} \mathrm{F}\) and compressibility factor of 0.85 .

The Reynolds number from Equation 2.34 is
\[
R e=0.0004778\left(\frac{14.7}{520}\right)\left(\frac{0.6 \times 100 \times 10^{6}}{8 \times 10^{-6} \times 23.25}\right)=4,357,109
\]

Assuming an internal roughness \(e=0.0007 \mathrm{in}\). and using Equation 2.45, we get the transmission factor \(F\) as follows:
\[
F=-4 \log _{10}\left(\frac{0.0007}{3.7 \times 23.25}+\frac{1.255 F}{4.3571 \times 10^{6}}\right)
\]

Solving by successive iteration,
\[
F=19.45
\]

The upstream pressure at milepost 110 is calculated from General Flow Equation 2.4, considering the elevation difference.

Elevation adjustment parameter \(s=0.0375 \times 0.6\left(\frac{350-1800}{530 \times 0.85}\right)=-0.0724\)
\[
e^{s}=e^{-0.0724}=0.9301
\]
and
\[
\begin{aligned}
L_{e} & =170 \times \frac{(0.9301-1)}{-0.0724}=164.03 \mathrm{mi} \\
100 \times 10^{6} & =38.77 \times 19.45\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-0.9301 \times 414.7^{2}}{0.6 \times 530 \times 164.03 \times 0.85}\right)^{0.5} 23.25^{2.5}
\end{aligned}
\]

Solving for \(P_{1}\), we get
\[
P_{1}=501.7 \mathrm{psia}=487.0 \mathrm{psig}
\]

The average pressure in the pipe segment is, by Equation 2.14,
\[
P_{\text {avg }}=\frac{2}{3}\left(501.7+414.7-\frac{501.7 \times 414.7}{501.7+414.7}\right)=459.6 \mathrm{psia}=444.9 \mathrm{psig}
\]

We will confirm the value of the compressibility factor \(Z\) we used earlier, using CNGA Equation 1.34:
\[
Z=\frac{1}{1+\frac{444.9 \times 344,400 \times(10)^{1.785 \times 0.6}}{(530)^{3.825}}}
\]
or
\[
Z=0.9359
\]

This value of \(Z\) is way off compared to the 0.85 value we used in our calculations. Recalculating \(P_{1}\) using the recent value of \(Z\), we obtain
\[
100 \times 10^{6}=38.77 \times 19.45\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-0.9301 \times 414.7^{2}}{0.6 \times 530 \times 164.03 \times 0.9359}\right)^{0.5} 23.25^{2.5}
\]

Solving for \(P_{1}\) by proportion, we get
\[
\frac{501.7^{2}-0.9301 \times 414.7^{2}}{0.85}=\frac{P_{1}^{2}-0.9301 \times 414.7^{2}}{0.9359}
\]
or
\[
P_{1}=510.86 \mathrm{psia}
\]

Recalculating the average pressure and the new compressibility factor \(Z\), we find
\[
P_{\text {avg }}=\frac{2}{3}\left(510.86+414.7-\frac{510.86 \times 414.7}{510.86+414.7}\right)=464.45 \mathrm{psia}=449.75 \mathrm{psig}
\]
and
\[
Z=\frac{1}{1+\frac{449.75 \times 344,400 \times(10)^{1.785 \times 0.6}}{(530)^{3.825}}}
\]
or
\[
Z=0.9352
\]

The percentage difference between this value of \(Z\) compared to the previously calculated value is
\[
\frac{0.9352-0.9359}{0.9359}=-0.07 \%
\]

This is good enough, and we won't iterate any further.

Therefore,
Pressure at Mayberry takeoff \((\) milepost 110\()=510.86 \mathrm{psia}=496.16 \mathrm{psig}\)
Next, calculate the pressure at milepost 50 , considering the pipe segment between Mavis and Mayberry at \(50^{\circ} \mathrm{F}\) flowing temperature and a flow rate of 120 MMSCFD .

The Reynolds number, by proportion, is
\[
R=4,357,109 \times \frac{120}{100}=5,228,531
\]

The transmission factor \(F\) is calculated from
\[
F=-4 \log \left(\frac{0.0007}{3.7 \times 23.25}+\frac{1.255 F}{5.2285 \times 10^{6}}\right)
\]

Solving by iteration,
\[
F=19.57
\]

The upstream pressure at milepost 50 is found from General Flow Equation 2.4, considering the elevation difference, as follows:
\[
\text { Elevation adjustment parameter } s=0.0375 \times 0.6\left(\frac{1800-1200}{510 \times 0.85}\right)=0.0311
\]
\[
e^{s}=e^{0.0311}=1.0316
\]
and
\[
\begin{gathered}
L_{e}=60 \times \frac{(1.0316-1)}{0.0311}=60.96 \mathrm{mi} \\
120 \times 10^{6}=38.77 \times 19.57\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-1.0316 \times 510.86^{2}}{0.6 \times 510 \times 60.96 \times 0.85}\right)^{0.5} 23.25^{2.5}
\end{gathered}
\]

Solving for \(P_{1}\), we get
\[
P_{1}=562.03 \mathrm{psia}=547.33 \mathrm{psig}
\]

Calculating the average pressure,
\[
P_{\text {avg }}=\frac{2}{3}\left(562.03+510.86-\frac{562.03 \times 510.86}{562.03+510.86}\right)=536.85 \mathrm{psia}=522.15 \mathrm{psig}
\]

Recalculating the compressibility factor \(Z\) using the new average pressure, we get
\[
Z=\frac{1}{1+\frac{522.15 \times 344,400 \times(10)^{1.785 \times 0.6}}{(510)^{3.825}}}
\]
or
\[
Z=0.9147
\]

Next, we recalculate the pressure at milepost 50 :
\[
\begin{aligned}
s & =0.0311 \times \frac{0.85}{0.9147}=0.0289 \\
e^{s} & =1.0293
\end{aligned}
\]
and
\[
\begin{gathered}
L_{e}=60 \times \frac{0.0293}{0.0289}=60.88 \mathrm{mi} \\
120 \times 10^{6}=38.77 \times 19.57\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-1.0293 \times 510.86^{2}}{0.6 \times 510 \times 60.88 \times 0.9147}\right)^{0.5} 23.25^{2.5}
\end{gathered}
\]

Solving for \(P_{1}\) by proportion, we get
\[
\begin{aligned}
& \frac{562.03^{2}-1.0316 \times 510.86^{2}}{0.85 \times 60.96}=\frac{P_{1}^{2}-1.0293 \times 510.86^{2}}{0.9147 \times 60.88} \\
& P_{1}=564.59 \mathrm{psia}=549.9 \mathrm{psig}
\end{aligned}
\]

The average pressure and \(Z\) are calculated next:
\[
P_{\text {avg }}=\frac{2}{3}\left(564.59+510.86-\frac{564.59 \times 510.86}{564.59+510.86}\right)=538.33 \mathrm{psia}=523.63 \mathrm{psig}
\]
and
\[
Z=\frac{1}{1+\frac{523.63 \times 344,400 \times(10)^{1.785 \times 0.6}}{(510)^{3.825}}}
\]
or
\[
Z=0.9145
\]

This is not too far from the previously calculated \(Z\) value of 0.914 . Therefore, we will not iterate any further.

The pressure at milepost 50 is
\[
P_{1}=564.59 \mathrm{psia}=549.9 \mathrm{psig}
\]

Next, calculate the upstream pressure at Fairfield, considering the 50 mi pipe segment flowing 130 MMSCFD at \(60^{\circ} \mathrm{F}\).

The Reynolds number from Equation 2.34 is
\[
\operatorname{Re}=0.0004778\left(\frac{14.7}{520}\right)\left(\frac{0.6 \times 130 \times 10^{6}}{8 \times 10^{-6} \times 23.25}\right)=5,664,242
\]

Using Equation 2.45, we get the transmission factor \(F\) as follows:
\[
F=-4 \log _{10}\left(\frac{0.0007}{3.7 \times 23.25}+\frac{1.255 F}{5.6642 \times 10^{6}}\right)
\]

Solving for \(F\), we get
\[
F=19.61
\]

The elevation adjustment is
\[
\begin{aligned}
s=0.0375 \times 0.6 \frac{1200-610}{520 \times 0.90} & =0.0284, \text { where } Z=0.9 \text { is assumed. } \\
e^{s} & =1.0288
\end{aligned}
\]
and
\[
L_{e}=50 \times \frac{0.0288}{0.0284}=50.65 \mathrm{mi}
\]

Using the General Flow equation, we calculate the upstream pressure \(P_{1}\) as
\[
130 \times 10^{6}=38.77 \times 19.61\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-1.0288 \times 564.59^{2}}{0.6 \times 520 \times 50.65 \times 0.9}\right)^{0.5} 23.25^{2.5}
\]

Solving for the pressure at Fairfield, we get
\[
P_{1}=613.89 \mathrm{psia}=599.2 \mathrm{psig}
\]

The average pressure and \(Z\) are calculated next:
\[
P_{\text {avg }}=\frac{2}{3}\left(613.89+564.59-\frac{613.89 \times 564.59}{613.89+564.59}\right)=589.58 \mathrm{psia}=574.88 \mathrm{psig}
\]
and
\[
Z=\frac{1}{1+\frac{574.88 \times 344,400 \times(10)^{1.785 \times 0.6}}{(520)^{3.825}}}
\]
or
\[
Z=0.913
\]

Recalculating, the elevation adjustment is
\[
\begin{aligned}
s & =\frac{0.0284 \times 0.9}{0.913}=0.028 \\
e^{s} & =1.0284
\end{aligned}
\]
and
\[
L_{e}=50 \times \frac{1.0284-1}{0.028}=50.70 \mathrm{mi}
\]

Recalculating the pressure at Fairfield, using the General Flow equation, we get
\[
130 \times 10^{6}=38.77 \times 19.61\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-1.0284 \times 564.59^{2}}{0.6 \times 520 \times 50.7 \times 0.913}\right)^{0.5} 23.25^{2.5}
\]

By proportions,
\[
\frac{P_{1}^{2}-1.0284 \times 564.59^{2}}{50.7 \times 0.913}=\frac{613.89^{2}-1.0288 \times 564.59^{2}}{50.65 \times 0.90}
\]
or
\[
P_{1}=614.40 \mathrm{psia}=599.7 \mathrm{psig}
\]

Recalculating, the average pressure and \(Z\) are
\[
P_{\mathrm{avg}}=\frac{2}{3}\left(614.4+564.59-\frac{614.4 \times 564.59}{614.4+564.59}\right)=589.85 \mathrm{psia}=575.15 \mathrm{psig}
\]
and
\[
Z=\frac{1}{1+\frac{575.15 \times 344,400 \times(10)^{1.785 \times 0.6}}{(520)^{3.825}}}
\]
or
\[
Z=0.913, \text { which is the same as before. }
\]

Therefore, the pressure at Fairfield is
\[
P_{1}=614.40 \mathrm{psia}=599.7 \mathrm{psig}
\]

The \(H P\) required is calculated from Equation 4.15:
\[
H P=0.0857\left(\frac{1.4}{0.4}\right) 130 \times 520\left(\frac{1+0.913}{2}\right)\left(\frac{1}{0.8}\right)\left[\left(\frac{614.4}{514.7}\right)^{\frac{0.4}{1.4}}-1\right]
\]
or
\[
H P=1258 \text { for phase } 1
\]

For phase 2, the inlet volume at Fairfield increases to 190 MMSCFD.

The pressure at milepost 200 will be calculated considering the pipe segment from milepost 200 to Beaumont at 100 MMSCFD.

From earlier calculations,
\[
\begin{gathered}
F=19.45 \\
s=0.0375 \times 0.6\left(\frac{350-750}{530 \times 0.9}\right)=-0.0189 \\
e^{s}=0.9813
\end{gathered}
\]
and
\[
L_{e}=80 \times \frac{0.9813-1}{-0.0189}=79.15 \mathrm{mi}
\]

We will assume \(Z=0.9\) throughout for simplicity. The pressure at milepost 200 is found from the General Flow equation
\[
100 \times 10^{6}=38.77 \times 19.45\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-0.9813 \times 414.7^{2}}{0.6 \times 530 \times 79.15 \times 0.9}\right)^{0.5} 23.25^{2.5}
\]

Solving for \(P_{1}\), we get
\[
P_{1}=464.35 \mathrm{psia}
\]

Next, calculate the pressure at milepost 110 , considering 90 mi of pipe at \(70^{\circ} \mathrm{F}\) with a flow of 160 MMSCFD . We will assume \(F=19.45\) and \(Z=0.9\) throughout for simplicity.
\[
\begin{gathered}
s=0.0375 \times 0.6\left(\frac{750-1800}{530 \times 0.9}\right)=-0.0495 \\
e^{s}=0.9517
\end{gathered}
\]
and
\[
L_{e}=90 \times \frac{0.9517-1}{-0.0495}=87.86 \mathrm{mi}
\]

The pressure at milepost 110 is found from the General Flow equation
\[
160 \times 10^{6}=38.77 \times 19.45\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-0.9517 \times 464.35^{2}}{0.6 \times 530 \times 87.86 \times 0.9}\right)^{0.5} 23.25^{2.5}
\]

Solving for \(P_{1}\), we get
\[
P_{1}=581.7 \mathrm{psia}
\]

Next, calculate the pressure at milepost 50 , considering 60 mi of pipe at \(50^{\circ} \mathrm{F}\) with a flow of 180 MMSCFD.
\[
\begin{aligned}
s & =0.0375 \times 0.6\left(\frac{1800-1200}{510 \times 0.9}\right)=0.0294 \\
e^{s} & =1.0298
\end{aligned}
\]
and
\[
L_{e}=60 \times \frac{1.0298-1}{0.0294}=60.92 \mathrm{mi}
\]

Therefore, the pressure at milepost 50 is found from the General Flow equation
\[
180 \times 10^{6}=38.77 \times 19.45\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-1.0298 \times 581.7^{2}}{0.6 \times 510 \times 60.92 \times 0.9}\right)^{0.5} 23.25^{2.5}
\]

Solving for \(P_{1}\), we get
\[
P_{1}=678.04 \mathrm{psia}
\]

Finally, calculate the pressure at Fairfield considering 50 mi of pipe at \(60^{\circ} \mathrm{F}\) with a flow rate of 190 MMSCFD.
\[
\begin{gathered}
s=0.0375 \times 0.6\left(\frac{1200-610}{520 \times 0.9}\right)=0.0284 \\
e^{s}=1.0288
\end{gathered}
\]
and
\[
L_{e}=50 \times \frac{1.0288-1}{0.0284}=50.65 \mathrm{mi}
\]

Therefore, the pressure at Fairfield is found from the General Flow equation
\[
190 \times 10^{6}=38.77 \times 19.45\left(\frac{520}{14.7}\right)\left(\frac{P_{1}^{2}-1.0288 \times 678.04^{2}}{0.6 \times 520 \times 50.65 \times 0.9}\right)^{0.5} 23.25^{2.5}
\]

Solving for \(P_{1}\), we get
\[
P_{1}=761.04 \mathrm{psia}
\]

The \(H P\) required at Fairfield for phase 2 is calculated from Equation 4.15:
\[
H P=0.0857\left(\frac{1.4}{0.4}\right) 190 \times 520\left(\frac{1+0.9}{2}\right)\left(\frac{1}{0.8}\right)\left[\left(\frac{761.04}{514.7}\right)^{\frac{0.4}{1.4}}-1\right]
\]
or
\[
H P=4161 \text { for phase } 2
\]

The capital cost is calculated next.
The weight per foot of NPS 24 pipe is calculated using Equation 6.11:
\[
w=10.68 \times 0.375 \times(24-0.375)=94.62 \mathrm{lb} / \mathrm{ft}
\]

Similarly, for the Travis branch, the weight per foot of NPS 16 pipe is calculated using Equation 6.11:
\[
w=10.68 \times 0.25 \times(16-0.25)=42.05 \mathrm{lb} / \mathrm{ft}
\]

The tonnage for 280 mi of NPS 24 pipe is
\[
\text { Tons }=\frac{94.62 \times 5280 \times 280}{2000}=69,943
\]

The tonnage for 20 mi of NPS 16 pipe is
\[
\text { Tons }=\frac{42.05 \times 5280 \times 20}{2000}=2220
\]
\[
\text { Total pipe cost }=\$ 1200 \times(69,943+2220)=\$ 86.61 \text { million }
\]

The installation cost of the pipe is calculated next:
\[
\text { Installation cost }=\$ 20,000 \times(24 \times 280+16 \times 20)=\$ 140.8 \text { million }
\]

The installation cost of the compressor station for phase 2 is
\[
\text { Compressor cost }=\$ 2000 \times 4161=\$ 8.33 \text { million }
\]

Considering mainline valves at 20 mi intervals, the total number of valves required for both the main line and the Travis branch is
\[
\begin{aligned}
\begin{aligned}
& \text { Number of valves }=\frac{280}{20}+1+\frac{20}{20}+1=17 \\
& \text { Total cost of valves }=\$ 100,000 \times 17=\$ 1.7 \text { million } \\
& \text { Total cost of all meter stations }=(\$ 500+\$ 200+\$ 250+\$ 300+\$ 350) \text { thousand } \\
&=\$ 1.6 \text { million } \\
& \text { Therefore, } \\
& \text { Total capital cost of all facilities }=(\$ 140.8+\$ 8.33+\$ 1.7+\$ 1.6+\$ 86.61) \text { million } \\
&=\$ 239.04 \text { million }
\end{aligned}
\end{aligned}
\]

To account for other items and indirect costs, increase the above by \(30 \%\) :
\[
\text { Total capital cost }=\$ 239.04 \times 1.3=\$ 310.75 \text { million. }
\]

\subsection*{8.3 SUMMARY}

This chapter reviewed some elementary concepts of transient pressures caused by valve closures and compressor station shutdown. Since the calculation methodology of transient pressures and flow rates requires the solution of partial differential equations and manual calculation is quite laborious, we refer the reader to an advanced text that specializes in this area of hydraulics. We also covered several real-world pipeline case studies.

\section*{PROBLEMS}
1. A natural gas pipeline system from an offshore facility is used to compress natural gas through 120 mi of NPS 24, 0.375 inch wall thickness pipe to an onshore location, similar to Figure 8.5. The compressor station located onshore is used to pump the gas through an 80 mi , NPS 20, 0.375 inch wall thickness pipe to a power plant. Determine the maximum flow rate possible under the following conditions. Neglect elevation effects. The compression ratio is 1.5 . Use the Weymouth equation with \(95 \%\) efficiency. Assume a base pressure of 14.7 psia and base temperature of \(60^{\circ} \mathrm{F}\). The gas flowing temperature is \(60^{\circ} \mathrm{F}\), and the compressibility factor is 0.9 . The gas gravity is 0.6 .
a. The gas pressure at the platform equals 1440 psig and free flow occurs without use of any compression offshore or onshore. The delivery pressure at the power plant is 500 psig .
b. Considering the MOP at the platform and onshore equal to 1440 psig , determine the maximum throughput possible with compression offshore and onshore.
2. Natural gas gathered from the Blanco fields is collected at Tapas and transported through a DN 600, 10 mm wall thickness pipeline system, 240 km long, that ties
into another DN 600, 10 mm wall thickness gas transmission pipeline at Rojas for eventual delivery to a power plant at Montecito. Tapas is at an elevation of 1500 m , whereas Rojas and Montecito are at 650 m and 300 m , respectively. The length of the pipeline from Rojas to Montecito is 140 km . The required delivery pressure at Montecito is 40 Bar gauge. The gas gravity and viscosity are 0.60 and 0.012 cP , respectively. The gas inlet temperature at Tapas is \(20^{\circ} \mathrm{C}\) and the pressure is 40 Bar gauge. Assume a constant gas flow temperature of \(20^{\circ} \mathrm{C}\). The pipeline MOP is 100 Bar gauge. The base temperature and base pressure are \(15^{\circ} \mathrm{C}\) and 1 Bar absolute, respectively. Use the Panhandle B equation with a pipeline efficiency of \(95 \%\). Assume a gas compressibility factor of 0.90 throughout.
a. Determine the compressor station power required to deliver \(4 \mathrm{Mm}^{3} /\) day at Montecito. Use \(80 \%\) isentropic efficiency and a specific heat ratio of 1.4.
b. What pipeline capacity can be expected if all compressor stations are shut down and free flow occurs from Tapas to Rojas and Montecito?

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\section*{CHAPTER 9}

\section*{Valves and Flow Measurements}

In this chapter we will discuss the various types of valves and flow measurements used on gas pipelines. The design and construction codes for valves, materials of construction, and application of the different types of valves and their performance characteristics will be explained. The importance of flow measurement in a gas pipeline and the accuracy of available instruments, codes, and standards used will be discussed. Various American National Standards Institute (ANSI), American Petroleum Institute (API), and American Gas Association (AGA) formulas used in connection with orifice meters will be reviewed. Since a small error in measurement in gas flow in a pipeline can translate to several thousand dollars of loss of revenue, it is important that industry strives to improve upon measurement methods. Accordingly, gas transportation companies and related industries have been researching better ways to improve flow measurement accuracy. For a detailed discussion of gas flow measurement, the reader is referred to the publications listed in the Reference section.

\subsection*{9.1 PURPOSE OF VALVES}

Valves are installed on pipelines and piping systems to isolate sections of piping for maintenance, to direct the fluid from one location to another, to shut down flow through pipe sections, and to protect pipe and prevent loss of fluid in the event of a rupture. On long-distance pipelines transporting natural gas and other compressible fluids, design codes and regulatory requirements dictate that sections of pipeline be isolated by installing mainline block valves at certain fixed spacing. For example, DOT 49 CFR, Part 192 requires that in class 1 locations, mainline valves be installed 20 mi apart. Class locations were discussed in Chapter 6.


Figure 9.1 Mainline valve installation. (Reproduced from Katz et al., Handbook of Natural Gas Engineering, McGraw-Hill, New York, 1959. With permission.)

A typical mainline block valve installation on a gas transmission pipeline is illustrated in Figure 9.1.

\subsection*{9.2 TYPES OF VALVES}

The various types of valves used in the gas pipeline industry include the following:
- Gate valve
- Ball valve
- Plug valve
- Butterfly valve
- Globe valve
- Check valve
- Control valve
- Relief valve
- Pressure regulating valve

Each of these valves listed will be discussed in detail in the following sections.
Valves can be of screwed design, welded ends, or flanged ends. In the gas industry, large valves are generally of the welded type, in which the valve is attached to the pipe on either side by a welded joint to prevent gas leakage to the atmosphere.


Figure 9.2 Mainline block valve.

In smaller sizes, screwed valves are used. A typical welded end mainline valve, along with smaller valves on either side, is shown in Figure 9.2.

Valves may be operated manually using a hand wheel or using an electric, pneumatic, or gas operator, as shown in Figure 9.3.


Figure 9.3 Valve with motor operator. (Reproduced from Nayyar, M.L., Piping Handbook, McGraw-Hill, New York, 2000. With permission.)

Table 9.1 ANSI Pressure Ratings for Valves
\begin{tabular}{rc}
\hline Class & Allowable Pressure, psi \\
\hline 150 & 275 \\
300 & 720 \\
400 & 960 \\
600 & 1440 \\
900 & 2160 \\
1500 & 3600 \\
\hline
\end{tabular}

\subsection*{9.3 MATERIAL OF CONSTRUCTION}

Most valves used in gas pipelines are constructed of steel and conform to specifications such as API, ASME, and ANSI standards. For certain gases that are corrosive and require certain special properties, some exotic materials can be used. The next section lists applicable standards and codes used in the design and construction of valves and fittings on gas pipelines.

The valve trim material, which refers to the various working parts of a valve such as the stem, wedge, and disc, are constructed of many different materials depending upon the pressure rating and service. Valve manufacturers designate their products using some form of a proprietary numbering system. However, the purchaser of the valve must specify the type of material and operating conditions required. A typical gate valve specification might be as follows: NPS 12, ANSI 600 gate valve, cast steel flanged ends rising stem 13\% CR, single wedge CS, stellite faced, seat rings SS 304, ABC company \#2308.

Valve operators may consist of a hand wheel or lever that is attached to the stem of the valve. Gear systems are used for larger valves. Electric motor operated valves are quite commonly used in gas pipeline systems, as are gas and pneumatic operators. Many valves can be buried, resulting in a portion of the valve and the operator above ground.

The pressure rating of a valve represents the internal pressure that the valve can be subject to under normal operating conditions. For example, an ANSI 600 rating refers to a valve that can be safely operated at pressures up to 1440 psig. Most gas pipelines are operated around this pressure rating. Table 9.1 shows the ANSI pressure ratings for valves and pipes.

If a valve is designated as an ANSI 600 rated valve, the manufacturer of the valve must hydrostatically test the valve at a higher pressure for a specified period of time, as required by the design code. Generally, the hydrotest pressure is \(150 \%\) of the valve rating. This compares with a hydrotest pressure of \(125 \%\) of MOP for pipelines, as discussed in Chapter 6.

\subsection*{9.4 CODES FOR DESIGN AND CONSTRUCTION}

The following is a list of applicable standards and codes used in the design and construction of valves and fittings on gas pipelines.

\footnotetext{
ASME B31.8: Gas Transmission and Distribution Piping Systems
ASME B16.3: Malleable Iron Threaded Fittings
}
```

ASME B16.5: Pipe Flanges and Flanged Fittings
ASME B16.9: Factory Made Wrought Steel Butt Welding Fittings
ASME B16.10: Face to Face and End to End Dimensions of Valves
ASME B16.11: Forged Steel Fittings, Socket Welding and Threaded Fittings
ASME B16.14: Ferrous Pipe Plugs, Bushing, etc.
ASME B16.20: Metallic Gaskets
ASME B16.21: Nonmetallic Gaskets
ASME B16.25: Butt Welding Ends
ASME B16.28: Wrought Steel, Butt Welding, Short Radius Elbows and Returns
ASME B16.36: Orifice Flanges
ANSI/ASTM A182: Forged or rolled alloy-steel pipe flanges, forged fittings, and
valves and parts for high temperature service
API 593: Ductile iron plug valves
API 594: Wafer type check valves
API 595: Cast iron gate valves
API 597: Steel venturi gate valves
API 599: Steel plug valves
API 600: Steel gate valves
API 602: Compact cast-steel gate valves
API 603: Class 150 corrosion-resistant gate valves
API 604: Ductile iron gate valves
API 606: Compact carbon-steel gate valves (extended bodies)
API 609: Butterfly valves to 150 psig and $150^{\circ} \mathrm{F}$
API 6D: Pipeline valves
MSS DS-13: Corrosion resistant cast flanged valves
MSS SP-25: Standard marking system for valves, fittings, and flanges

```

\subsection*{9.5 GATE VALVE}

A gate valve is generally used to completely shut off fluid flow or, in the fully open position, provide full flow in a pipeline. Thus, it is used either in the fully closed or fully open position. A gate valve consists of a valve body, seat, and disc; a spindle; a gland; and a wheel for operating the valve. The seat and the gate together perform the function of shutting off the flow of fluid. A typical gate valve is shown in Figure 9.4.

Gate valves are generally not suitable for regulating flow or pressure or operating in a partially open condition. For this service, a plug valve or control valve should be used. It must be noted that, due to the type of construction, a gate valve requires many turns of the hand wheel to completely open or close the valve. When fully opened, gate valves offer little resistance to flow, and their equivalent length to diameter ratio (L/D) is approximately 8 . The equivalent \(\mathrm{L} / \mathrm{Ds}\) for commonly used valves and fittings are listed in Table 9.2. This ratio represents the resistance of the valve.

The gate valves used in the main lines carrying oil or gas must be of full bore or through conduit design to enable smooth passage of scrapers or pigs used for cleaning or monitoring pipelines. Such gate valves are referred to as full bore or through conduit gate valves.


Figure 9.4 Typical gate valve.

Table 9.2 Equivalent Lengths of Valves and Fittings
\begin{tabular}{lr}
\hline Description & L/D \\
\hline Gate valve & 8 \\
Globe valve & 340 \\
Angle valve & 55 \\
Ball valve & 3 \\
Plug valve straightway & 18 \\
Plug valve 3-way thru-flo & 30 \\
Plug valve branch flo & 90 \\
Swing check valve & 50 \\
Lift check valve & 600 \\
Standard elbow- \(90^{\circ}\) & 30 \\
Standard elbow-45 & 16 \\
Standard elbow long radius \(90^{\circ}\) & 16 \\
Standard tee thru-flo & 20 \\
Standard tee thru-branch & 60 \\
Mitre bends- \(\alpha=0\) & 2 \\
Mitre bends- \(\alpha=30\) & 8 \\
Mitre bends- \(\alpha=60\) & 25 \\
Mitre bends- \(\alpha=90\) & 60 \\
\hline
\end{tabular}


Figure 9.5 Typical ball valve. (Reproduced from Nayyar, M.L., Piping Handbook, McGrawHill, New York, 2000. With permission.)

\subsection*{9.6 BALL VALVE}

A ball valve consists of a valve body in which a large sphere with a central hole equal to the inside diameter of the pipe is mounted. As the ball is rotated, in the fully open position the valve provides the through conduit or full bore required for unrestricted flow of the fluid and scrapers or pigs. Compared to a gate valve, a ball valve has very little resistance to flow in the fully open position. When fully open, the L/D ratio for a ball valve is approximately 3.0. The ball valve, like the gate valve, is generally used in the fully open or fully closed position. A typical ball valve is shown in Figure 9.5.

Unlike a gate valve, a ball valve requires one-quarter turn of the hand wheel to go from the fully open to the fully closed position. Such quick opening and closing of a ball valve can be important in some installations where isolating pipe sections quickly is needed in the event of emergency.

\subsection*{9.7 PLUG VALVE}

The plug valve traces its origin to the beginnings of the valve industry. It is a simple device for shutting off or allowing the flow of a fluid in a pipe by a simple quarter turn of the handle. In this sense, it is similar to the ball valve. Plug valves are generally used in screwed piping and in small pipe sizes. Plug valves can be hand wheel operated or operated using a wrench or gearing mechanism. The L/D ratio for this type of valve ranges from 18 to 90 , depending upon the design. A typical plug valve is shown in Figure 9.6.

\subsection*{9.8 BUTTERFLY VALVE}

The butterfly valve was originally used where a tight closure was not absolutely necessary. However, over the years, this valve has been manufactured with fairly tight seals made of rubber or elastomeric materials that provide good shutoff similar


Figure 9.6 Typical plug valve.
to other types of valves. Butterfly valves are used where space is limited. Unlike gate valves, butterfly valves can be used for throttling or regulating flow, as well as in the full open and fully closed positions. The pressure loss through a butterfly valve is higher in comparison with the gate valve. The L/D ratio for this type of valve is approximately 45 . Butterfly valves are used in large and small sizes. They can be hand wheel operated or operated using a wrench or gearing mechanism. A typical butterfly valve is shown in Figure 9.7.

\subsection*{9.9 GLOBE VALVE}

Globe valves, so called because of their outside shape, are widely used in plant piping. They are suitable for manual and automatic operation. Unlike the gate valve, globe valves can be used for regulating flow or pressures as well as complete


Figure 9.7 Typical butterfly valve.


Figure 9.8 Typical globe valve.
shutoff of flow. They can also be used as pressure relief valves or as check valves. Compared to a gate valve or ball valve, the globe valve has considerably higher pressure loss in the fully open position. This is due to the fact that the flow of fluid changes direction as it goes through the valve. The L/D ratio for this type of valve is approximately 340 . Globe valves are manufactured in sizes up to NPS 16. They are generally hand wheel operated. A typical globe valve is shown in Figure 9.8.

\subsection*{9.10 CHECK VALVE}

Check valves are normally in the closed position and are open when the fluid flows through them. They also have the capability of shutting off the flow in the event the pressure downstream exceeds the upstream pressure. In this respect, they are used for flow in one direction only. Thus, they prevent back flow through the valve. Since flow of the fluid through these valves is allowed to be in one direction only, check valves must be installed properly by noting the normal direction of flow. An arrow stamped on the outside of the valve body indicates the direction of flow. Check valves can be classified as swing check valves and lift check valves. The L/D ratios for check valves range from 50 for the swing check valve to as high as 600 for lift check valves. Examples of typical check valves are shown in Figure 9.9.


Figure 9.9 Typical check valves. (Reproduced from Tullis, J.P., Hydraulics of Pipelines, John Wiley \& Sons, New York, 1989. With permission.)

\subsection*{9.11 PRESSURE CONTROL VALVE}

A pressure control valve is used to automatically control the pressure at a certain point in a pipeline. In this respect it is similar to a pressure regulator discussed next. Whereas the pressure regulator is generally used to maintain a constant downstream pressure, a pressure control valve is used to control the upstream pressure. The upstream and downstream are relative to the location of the valve on the pipeline. Generally, a bypass piping system around the control valve is installed to isolate the control valve in the event of an emergency or for maintenance work on the control valve. This is illustrated in Figure 9.10.


Figure 9.10 Pressure control valve.


Figure 9.11 Pressure regulator. (Reproduced from Katz et al., Handbook of Natural Gas Engineering, McGraw-Hill, New York, 1959. With permission.)

\subsection*{9.12 PRESSURE REGULATOR}

A pressure regulator is a valve that is similar to a pressure control valve. Its function is to control or regulate the pressure in a certain section of a pipeline system. For example, on a lateral piping that comes off a main pipeline, used for delivering gas to a customer, a lower pressure might be required on the customer side. If the main pipeline pressure at the point of connection to the lateral pipeline is 800 psig , but the customer's piping is limited to 600 psig , a pressure regulator is used to reduce the pressure by 200 psig, as shown in Figure 9.11.

\subsection*{9.13 PRESSURE RELIEF VALVE}

The pressure relief valve is used to protect a section of piping by relieving the pipeline pressure when it reaches a certain value. For example, if the MOP of a pipeline system is 1400 psig , a pressure relief valve may be set at 1450 psig . Any upset conditions
that cause the pipeline pressure to exceed the normal 1400 psig will cause the relief valve to open at the set point of 1450 psig and expel the gas to the atmosphere or to a relief vessel, thereby protecting the pipeline from overpressure and, eventually, rupture. The difference between the relief valve set point ( 1450 psig ) and the pipeline MOP ( 1400 psig ) will depend on the actual application, the valve type, and expected fluctuations in pressure. Generally, the difference will range between 20 and 50 psig . Too close a difference will result in frequent operation of the relief valve, which will be a nuisance and, in many cases, a waste of valuable gas. A large difference between a relief valve set point and the pipeline MOP may render the valve ineffective.

\subsection*{9.14 FLOW MEASUREMENT}

Gas flow measurement in a pipeline is necessary for properly accounting for the amount of gas transported from one point to another along a gas pipeline. The owner of the gas and the customer who purchases the gas both require that the correct amount of gas be delivered for the agreed-upon price. Even a very small error in flow measurement on large-capacity pipelines can result in huge losses to either the owner or customer of the gas. For example, consider a gas pipeline transporting 300 MMSCFD at a tariff of 50 cents per MCF. An error of \(1 \%\) in the gas flow measurement can translate to a loss of more than \(\$ 500,000\) per year to either the seller or the buyer. Hence, it is easy to appreciate the importance of good, accurate flow measurement in gas pipelines. Over the years, gas flow measurement technology have improved considerably. Many organizations have jointly developed standards and procedures for measurement of natural gas through orifice meters installed in pipelines. AGA, API, ANSI, and ASME have together endorsed standards for orifice metering of natural gas. The AGA Measurement Committee Report No. 3 is considered to be the leading publication in this regard. This standard is also endorsed by ANSI and API and is referred to as the ANSI/API 2530 standard. We will refer to sections of this standard when discussing orifice meters.

\subsection*{9.15 FLOW METERS}

Since the orifice meter is the main flow measurement instrument used in the gas industry, we will discuss this first.

\subsection*{9.15.1 Orifice Meter}

The orifice meter is a flat steel plate that has a concentric machined hole with a sharp edge and is positioned inside the pipe, as shown in Figure 9.12.

As the gas flows through the pipeline and then through the orifice plate, due to the reduction in cross-sectional area as the gas approaches the orifice, the velocity of flow increases and, correspondingly, the pressure drops. After the orifice, the crosssectional area increases again back to the full pipe diameter, which results in expansion of gas and decrease in flow velocity. This process of accelerating flow through


Figure 9.12 Orifice meter.
the orifice and subsequent expansion forms a vena contracta, or a throat, immediately past the orifice, as shown in the figure. Three different types of orifice meters are illustrated in Figure 9.13.

The different types of orifice meters shown have different crest shapes, which affect the extent of contraction of the jet of gas as it flows through the orifice. The contraction coefficient \(C_{c}\) is defined in terms of the area of cross section of the vena contracta compared to the cross-sectional area of the orifice, as defined below:
\[
\begin{equation*}
C_{c}=\frac{A_{c}}{A_{o}} \tag{9.1}
\end{equation*}
\]
where
\(C_{c}=\) contraction coefficient, dimensionless
\(A_{c}=\) cross-sectional area of the vena contracta, \(\mathrm{in}^{2}\)
\(A_{o}=\) cross-sectional area of the orifice, \(\mathrm{in}^{2}\)


Figure 9.13 Different types of orifice meters. (Reproduced from Liu, H., Pipeline Engineering, CRC Press, Boca Raton, FL, 2003.)

The discharge through the orifice meter is represented by the following basic equation:
\[
\begin{equation*}
Q=C_{c} C_{v} A_{o} \sqrt{\frac{2\left[\left(p_{1}-p_{2}\right) / \rho+g\left(z_{1}-z_{2}\right)\right]}{1-C_{c}^{2}\left(A_{o} / A\right)^{2}}} \tag{9.2}
\end{equation*}
\]
where
\(Q=\) flow rate, \(\mathrm{ft}^{3} / \mathrm{s}\)
\(C_{c}=\) contraction coefficient, dimensionless
\(C_{v}=\) discharge coefficient, dimensionless
\(A_{o}=\) cross-sectional area of the orifice, \(\mathrm{in}^{2}\)
\(A=\) cross-sectional area of pipe containing the orifice, \(\mathrm{in}^{2}\)
\(p_{1}=\) upstream pressure, psig
\(p_{2}=\) downstream pressure, psig
\(\rho=\) density of gas, \(\mathrm{lb} / \mathrm{ft}^{3}\)
\(z_{1}=\) upstream elevation, ft
\(z_{2}=\) downstream elevation, ft
\(g=\) acceleration due to gravity
When the elevation difference between the upstream and downstream pressure taps is negligible, the discharge equation for the orifice meter can be simplified to
\[
\begin{equation*}
Q=C_{c} C_{v} A_{o} \sqrt{\frac{2\left(p_{1}-p_{2}\right) / \rho}{1-C_{c}^{2}\left(\frac{A_{o}}{A}\right)^{2}}} \tag{9.3}
\end{equation*}
\]
where all symbols are as defined earlier.
For round-crested and nozzle-crested orifice meters, shown in Figure 9.13, the value of \(C_{c}\) can be taken as 1.0 . This indicates an absence of vena contracta for these types of orifices. For the sharp-crested orifice at high Reynolds numbers or for turbulent flow, \(C_{c}\) is calculated from the equation
\[
\begin{equation*}
C_{c}=0.595+0.29\left(\frac{A_{o}}{A}\right)^{\frac{5}{2}} \tag{9.4}
\end{equation*}
\]
where all symbols are as defined earlier.
There are basically two types of pressure measurements in orifice meters. These are called flange taps and pipe taps. They relate to the locations where the pressure measurements are taken. A flange tap requires that the upstream tap be located at a distance of 1 in . upstream of the nearest plate face and that the downstream tap be located 1 in . downstream of the nearest plate face. Pipe taps are such that the upstream tap be located at a distance of 2.5 times the inside diameter of the pipe, upstream of the nearest plate face and that the downstream tap be located at a distance of 8 times the inside diameter of the pipe, downstream of the nearest plate face. Figure 9.14 illustrates the location of flange taps and pipe taps.


Figure 9.14 Flange taps and pipe taps.

Several terms used in the calculation of the orifice flow must be explained first. The differential pressure for an orifice is the pressure difference between the upstream and downstream taps. The orifice diameter is defined as the arithmetic average of four or more inside diameter measurements evenly spaced. Strict tolerances for the orifice diameters are specified in the AGA3/ANSI 2530 standard. Table 9.3 shows these tolerances taken from the standard.

\subsection*{9.15.1.1 Meter Tube}

The meter tube is the piece of pipe in which the orifice plate is installed, along with straightening vanes as needed. A typical meter tube consisting of the orifice plate and straightening vanes is illustrated in Figure 9.15.

The dimensions of the meter tube, such as \(\mathrm{A}, \mathrm{B}, \mathrm{C}\), and \(\mathrm{C}^{\prime}\), depend upon the orifice to pipe diameter ratio, also known as beta ratio \(\beta\), and are specified in AGA Report No. 3. For example, for beta \(=0.5\),
\[
A=25 \quad A^{\prime}=10 \quad B=4 \quad C=5 \quad C^{\prime}=5.5
\]

Table 9.3 Orifice Plate Diameter Tolerances
\begin{tabular}{cc}
\hline \begin{tabular}{c} 
Orifice \\
Diameter, in.
\end{tabular} & \begin{tabular}{c} 
Tolerance \\
\(+/-\), in.
\end{tabular} \\
\hline 0.250 & 0.0003 \\
0.375 & 0.0004 \\
0.500 & 0.0005 \\
0.625 & 0.0005 \\
0.750 & 0.0005 \\
0.875 & 0.0005 \\
1.000 & 0.0005 \\
above 1.000 & 0.0005 (per inch dia) \\
\hline
\end{tabular}


Figure 9.15 Meter tube installation.

These numbers are actually multiples of the pipe or meter tube diameter. The requirements of straightening vanes before the orifice plate depend on the specific installation. The main reason for straightening vanes is to reduce flow disturbance at the orifice plate from upstream fittings. Refer to AGA Report No. 3 for various meter tube configurations.

The orifice flow rate is the mass flow rate or volume flow rate of gas per unit of time. The density is the mass per unit volume of gas at a specific temperature and pressure.

\subsection*{9.15.1.2 Expansion Factor}

The expansion factor is a dimensionless factor used to correct the calculated flow rate to take into account the reduction in gas density as it flows through an orifice, which is caused by the increased velocity and corresponding reduced static pressure. Methods of calculating the expansion factor \(Y\) will be discussed in subsequent sections.

The beta ratio is defined as the ratio of the orifice diameter to the meter tube diameter, as follows:
\[
\begin{equation*}
\beta=\frac{d}{D} \tag{9.5}
\end{equation*}
\]

For orifice meters with flange taps, the beta ratio ranges between 0.15 and 0.70 . For orifice meters with pipe taps, the beta ratio ranges between 0.20 and 0.67 , where
\(\beta=\) beta ratio, dimensionless
\(d=\) orifice diameter, in.
\(D=\) meter tube diameter, in.

The fundamental orifice meter flow equation described in ANSI 2530/AGA Report No. 3 is as follows:
\[
\begin{equation*}
q_{m}=\frac{C}{\left(1-\beta^{4}\right)^{0.5}} Y \frac{\pi}{4} d^{2}\left(2 g \rho_{f} \Delta P\right)^{0.5} \tag{9.6}
\end{equation*}
\]
or
\[
\begin{align*}
q_{m} & =K Y \frac{\pi}{4} d^{2}\left(2 g \rho_{f} \Delta P\right)^{0.5}  \tag{9.7}\\
\beta & =\frac{d}{D}  \tag{9.8}\\
K & =\frac{C}{\left(1-\beta^{4}\right)^{0.5}}=\frac{C D^{2}}{\left(D^{4}-d^{4}\right)^{0.5}} \tag{9.9}
\end{align*}
\]
where
\(q_{m}=\) mass flow rate of gas, \(\mathrm{lb} / \mathrm{s}\)
\(\rho_{f}=\) density of gas, \(\mathrm{lb} / \mathrm{ft}^{3}\)
\(C=\) discharge coefficient
\(\beta=\) beta ratio, dimensionless
\(d=\) orifice diameter, in.
\(D=\) meter tube diameter, in.
\(Y=\) expansion factor, dimensionless
\(g=\) acceleration due to gravity, \(\mathrm{ft} / \mathrm{s}^{2}\)
\(\Delta P=\) pressure drop across the orifice, psi
\(K=\) flow coefficient, dimensionless
These equations were arrived at using the conservation of energy and mass equations with thermodynamics and the equation of state for the gas in question. It can be seen that, essentially, these formulas give the mass flow rate of gas. We need to convert these to the volume flow rate using the density. The coefficient of discharge \(C\) in the preceding equation is approximately 0.6 , and the flow coefficient \(K\) is a value that is between 0.6 and 0.7 . Both the flow coefficient \(K\) and the expansion factor \(Y\) are determined using test data. The volume flow rate at standard (base) conditions is calculated from the mass flow rate as follows:
\[
\begin{equation*}
q_{v}=\frac{q_{m}}{\rho_{b}} \tag{9.10}
\end{equation*}
\]
where
\(q_{v}=\) volume flow rate, \(\mathrm{ft}^{3} / \mathrm{s}\)
\(q_{m}=\) mass flow rate, \(\mathrm{lb} / \mathrm{s}\)
\(\rho_{b}=\) gas density at base temperature, \(\mathrm{lb} / \mathrm{ft}^{3}\)
The expansion factor \(Y\) for low-compressibility fluids, such as water at \(60^{\circ} \mathrm{F}\) and 1 atmosphere pressure, is taken as 1.0 . For gases, \(Y\) can be calculated as explained in the next section. The flow coefficient \(K\) is found to vary with the diameter of the meter tube D , orifice diameter \(d\), mass flow rate \(q_{m}\), and fluid density and viscosity at the flowing temperature. For gases, \(K\) also varies with the ratio of
the differential pressure to the static pressure and \(k\), the ratio of specific heat of the gas. In many cases, the flow coefficient \(K\) is considered to be a function of the Reynolds number, acoustic ratio, meter tube diameter, and beta ratio. Rearranging Equation 9.7, we get
\[
\begin{equation*}
K Y=\frac{4 q_{m}}{\pi d^{2}\left[2 g_{c} \rho_{f} \Delta P\right]^{0.5}} \tag{9.11}
\end{equation*}
\]

Several empirical equations are available to calculate the flow coefficient \(K\). The following equation by Buckingham and Bean is endorsed by the National Bureau of Standards (NBS) and is listed in AGA Report No.3.

For flange taps,
\[
\begin{align*}
K_{e}= & 0.5993+\frac{0.007}{D}+\left(0.364+\frac{0.076}{D^{0.5}}\right) \beta^{4}+0.4\left(1.6-\frac{1}{D}\right)^{5}\left[\left(0.07+\frac{0.5}{D}\right)-\beta\right]^{2.5} \\
& -\left(0.009+\frac{0.034}{D}\right)(0.5-\beta)^{1.5}+\left(\frac{65}{D^{2}}+3\right)(\beta-0.7)^{2.5} \tag{9.12}
\end{align*}
\]
where
\(K_{e}=\) flow coefficient for Reynolds number \(R_{d}=d\left(10^{6} / 15\right)\), dimensionless
\(D=\) meter tube diameter, in.
\(d=\) orifice diameter, in.
\(\beta=\) beta ratio, dimensionless
For pipe taps,
\[
\begin{align*}
K_{e}= & 0.5925+\frac{0.0182}{D}+\left(0.440-\frac{0.06}{D}\right) \beta^{2}+\left(0.935+\frac{0.225}{D}\right) \beta^{5} \\
& +1.35 \beta^{14}+\frac{1.43}{D^{0.5}}(0.25-\beta)^{2.5} \tag{9.13}
\end{align*}
\]
where all symbols are as defined before.
For flange taps and pipe taps, the value of \(K_{o}\) is calculated from
\[
\begin{equation*}
K_{o}=\frac{K_{e}}{1+\frac{15 \times 10^{-6} E}{d}} \tag{9.14}
\end{equation*}
\]
where the parameter \(E\) in Equation 9.14 is found from
\[
\begin{equation*}
E=d\left(830-5000 \beta+9000 \beta^{2}-4200 \beta^{3}+B\right) \tag{9.15}
\end{equation*}
\]

The value of parameter \(B\) in Equation 9.15 is defined as follows:
\[
\begin{align*}
\text { For flange taps, } B & =\frac{530}{D^{0.5}}  \tag{9.16}\\
\text { For pipe taps, } B & =\frac{875}{D}+75 \tag{9.17}
\end{align*}
\]

Finally, the flow coefficient \(K\) is calculated from
\[
\begin{equation*}
K=K_{o}\left(1+\frac{E}{R_{d}}\right) \tag{9.18}
\end{equation*}
\]
where
\(K_{o}=\) flow coefficient for infinitely large orifice Reynolds numbers, dimensionless
\(R_{d}=\) Reynolds number at the inlet of orifice, dimensionless
The Reynolds number used in the preceding equations is calculated from
\[
\begin{equation*}
R_{d}=\frac{V_{f} d \rho_{f}}{\mu} \tag{9.19}
\end{equation*}
\]
where
\(R_{d}=\) Reynolds number at the inlet of orifice, dimensionless
\(V_{f}=\) velocity of fluid at inlet of orifice, \(\mathrm{ft} / \mathrm{s}\)
\(d=\) orifice diameter, ft
\(\rho_{f}=\) fluid density at flowing conditions, \(\mathrm{lb} / \mathrm{ft}^{3}\)
\(\mu=\) dynamic viscosity of fluid, lb/ft.s

The values of flow coefficient \(K\) calculated using the preceding equations apply to orifice meters manufactured and installed in accordance with AGA Report No. 3, as long as the meter tube is greater than 1.6 in . inside diameter and the beta ratio is between 0.10 and 0.75 .

The uncertainties in flow coefficient \(K\), in accordance with AGA Report No. 3, follow:

For flange taps, the uncertainty is \(+/-0.5 \%\) for \(0.15<\beta<0.70\)
For flange taps, the uncertainty is greater than \(+/-1.0 \%\) for \(0.15>\beta>0.70\)
For pipe taps, the uncertainty is \(+/-0.75 \%\) for \(0.20<\beta<0.67\)
For pipe taps, the uncertainty is greater than \(+/-1.5 \%\) for \(0.20>\beta>0.67\)
The expansion factor \(Y\) is calculated in two ways. In the first method, it is calculated using the upstream pressure, and in the second method, it is calculated using the downstream pressure. The following equation is used for the expansion factor \(Y_{1}\) with reference to upstream pressure.

For flange taps,
\[
\begin{equation*}
Y_{1}=1-\left(0.41+0.35 \beta^{4}\right) \frac{x_{1}}{k} \tag{9.20}
\end{equation*}
\]

For pipe taps,
\[
\begin{equation*}
Y_{1}=1-\left[0.333+1.145\left(\beta^{2}+0.7 \beta^{5}+12 \beta^{13}\right)\right] \frac{x_{1}}{k} \tag{9.21}
\end{equation*}
\]
and the pressure ratio \(x_{1}\) is
\[
\begin{equation*}
x_{1}=\frac{P_{f 1}-P_{f 2}}{P_{f 1}}=\frac{h_{w}}{27.707 P_{f 1}} \tag{9.22}
\end{equation*}
\]
where
\(Y_{1}=\) expansion factor based on upstream pressure
\(x_{1}=\) ratio of differential pressure to absolute upstream static pressure
\(h_{w}=\) differential pressure between upstream and downstream taps in in. of water at \(60^{\circ} \mathrm{F}\)
\(P_{f 1}=\) static pressure at upstream tap, psia
\(P_{f 2}=\) static pressure at downstream tap, psia
\(x_{1} / k=\) acoustic ratio, dimensionless
\(k=\) ratio of specific heats of gas, dimensionless

The value of \(Y_{1}\) calculated using these equations is subject to a tolerance from 0 to \(+/-0.5 \%\) for the range of \(x=0\) to 0.20 . For larger values of \(x\), the uncertainty is larger. For flange taps, the values of \(Y_{1}\) are valid for a beta ratio range of 0.10 to 0.80 . For pipe taps, the beta ratio range is 0.10 to 0.70 .

With reference to the downstream pressure, the expansion factor \(Y_{2}\) is calculated using the following equations.

For flange taps,
\[
\begin{align*}
& Y_{2}=Y_{1}\left(\frac{1}{1-x_{1}}\right)^{0.5}  \tag{9.23}\\
& Y_{2}=\left(1+x_{2}\right)^{0.5}-\left(0.41+0.35 \beta^{4}\right) \frac{x_{2}}{k\left(1+x_{2}\right)^{0.5}} \tag{9.24}
\end{align*}
\]

For pipe taps,
\[
\begin{equation*}
Y_{2}=\left(1+x_{2}\right)^{0.5}-\left[0.333+1.145\left(\beta^{2}+0.7 \beta^{5}+12 \beta^{13}\right)\right] \frac{x_{2}}{k\left(1+x_{2}\right)^{0.5}} \tag{9.25}
\end{equation*}
\]
and the pressure ratio \(x_{2}\) is
\[
\begin{equation*}
x_{2}=\frac{P_{f 1}-P_{f 2}}{P_{f 2}}=\frac{h_{w}}{27.707 P_{f 2}} \tag{9.26}
\end{equation*}
\]
where all symbols are as defined before.
The density of the flowing gas used in Equation 9.6 must be obtained from the equation of state or from tables. It is important to use the correct density in the flow equations. Otherwise, the uncertainty in flow measurement could be as great as \(10 \%\). Generally, the density of the gas can be calculated from the perfect gas law discussed in Chapter 1, with the modification using the compressibility factor. The following equation is obtained by rearranging the real gas equation and using the gravity of gas (see Chapter 1 for details):
\[
\begin{align*}
& \rho_{f}=\frac{m}{V}=\frac{G_{i} M P_{f}}{Z_{f} R T_{f}}  \tag{9.27}\\
& \rho_{f 1}=\frac{G_{i} M P_{f 1}}{Z_{f 1} R T_{f}}  \tag{9.28}\\
& \rho_{b}=\frac{M G_{i} P_{b}}{R Z_{b} T_{b}} \tag{9.29}
\end{align*}
\]
where
\(m=\) mass of gas
\(V=\) volume of gas
\(G_{i}=\) gravity of gas (air =1.00)
\(M=\) molecular weight of gas
\(P_{f}=\) absolute gas pressure
\(Z_{f}=\) compressibility factor at flowing temperature
\(R=\) gas constant
\(T_{f}=\) absolute flowing temperature
subscript \(f 1\) refers to upstream tap flowing conditions
subscript \(f 2\) refers to downstream tap flowing conditions
subscript \(b\) refers to base conditions
Two other equations, based on real gas specific gravity and taking the base conditions of 14.73 psia and \(60^{\circ} \mathrm{F}\), result in the gas densities at the upstream tap and at the base conditions as follows:
\[
\begin{align*}
\rho_{f 1} & =\frac{M Z_{b} G P_{f 1}}{0.99949 R Z_{f 1} T_{f}}  \tag{9.30}\\
\rho_{b} & =\frac{M G P_{b}}{0.99949 R T_{b}} \tag{9.31}
\end{align*}
\]

Knowing the densities at the upstream tap and at the base condition, the following equation is used for the volume flow rate. This equation is derived from the equations listed in the preceding sections.
\[
\begin{equation*}
q_{v}=\frac{\pi}{4} \frac{\sqrt{2 g}\left(K Y_{1} d^{2}\right) \sqrt{\left(\rho_{f 1} \Delta P\right)}}{\rho_{b}} \tag{9.32}
\end{equation*}
\]

Combining all equations we have reviewed so far, AGA Report No. 3 shows a compact equation for the flow of gas through an orifice meter as follows:
\[
\begin{equation*}
Q_{v}=C \sqrt{h_{w} P_{f}} \tag{9.33}
\end{equation*}
\]
where
```

$Q_{v}=$ gas flow rate at base conditions, $\mathrm{ft}^{3} / \mathrm{h}$
$h_{w}=$ differential pressure between upstream and downstream taps in in. of water
at $60^{\circ} \mathrm{F}$
$P_{f}=$ absolute static pressure, psia
$C=$ orifice flow constant

```

For \(P_{f}\), subscript 1 is used for upstream and subscript 2 for downstream pressure.
The orifice flow constant \(C\) consists of the product of several factors that depend on the Reynolds number, expansion factor, base pressure, base temperature, flowing temperature, gas gravity, and supercompressibility factor of gas. It is given by the following equation:
\[
\begin{equation*}
C=F_{b} F_{r} F_{p b} F_{t b} F_{t f} F_{g r} F_{p v} Y \tag{9.34}
\end{equation*}
\]
where the dimensionless factors are
\(F_{b}=\) basic orifice factor
\(F_{r}=\) Reynolds number factor
\(F_{p b}=\) pressure base factor
\(F_{t b}=\) temperature base factor
\(F_{t f}=\) flowing temperature factor
\(F_{g r}=\) gas relative density factor
\(F_{p v}=\) supercompressibility factor
\(Y=\) expansion factor
These values of the factors that constitute the orifice flow constant \(C\) are defined in AGA Report No. 3 and are listed in Appendix B of that publication. However, each of these factors can be calculated as follows.

The basic orifice factor is
\[
\begin{equation*}
F_{b}=338.178 d^{2} K_{o} \tag{9.35}
\end{equation*}
\]
where \(K_{o}\) is calculated using Equation 9.14.

The Reynolds number factor is
\[
\begin{align*}
F_{r} & =1+\frac{E}{R_{d}}  \tag{9.36}\\
K & =K_{o} F_{r} \tag{9.37}
\end{align*}
\]

The pressure base factor is
\[
\begin{equation*}
F_{p b}=\frac{14.73}{P_{b}} \tag{9.38}
\end{equation*}
\]

The temperature base factor is
\[
\begin{equation*}
F_{t b}=\frac{T_{b}}{519.67} \tag{9.39}
\end{equation*}
\]

The flowing temperature factor is
\[
\begin{equation*}
F_{t f}=\left(\frac{519.67}{T_{f}}\right)^{0.5} \tag{9.40}
\end{equation*}
\]

The gas relative density factor is
\[
\begin{equation*}
F_{g r}=\left(\frac{1}{G_{r}}\right)^{0.5} \tag{9.41}
\end{equation*}
\]
where all symbols in the preceding equations are as defined before.

\subsection*{9.16 VENTURI METER}

The venturi meter, shown in Figure 9.16, is based upon Bernoulli's equation. It consists of a smooth gradual contraction from the main pipe size to a reduced section known as the throat, finally expanding back gradually to the original pipe diameter.

This type of venturi meter is called the Herschel type. The angle of contraction from the main pipe to the throat section is in the range of \(21^{\circ}+/-2^{\circ}\). The gradual expansion from the throat to the main pipe section is in the range of 5 to \(15^{\circ}\). This design causes the least energy loss such that the discharge coefficient can be assumed at 1.0. Venturi meters range in size from 4.0 in . to 48.0 in . The beta ratio, equal to \(d / D\), generally ranges between 0.30 and 0.75 .

The gas pressure in the main pipe section is represented by \(P_{1}\) and that at the throat is represented by \(P_{2}\). As gas flows through a venturi meter, it increases in flow velocity in the narrow throat section. Correspondingly, the pressure reduces in the throat section according to Bernoulli's equation. After gas leaves the throat section,


Figure 9.16 Venturi meter.
it reduces in flow velocity due to the increase in pipe cross-sectional area, and it reaches the original flow velocity.

The flow velocity in the main pipe section before the throat is calculated from the known pressures \(P_{1}\) and \(P_{2}\) :
\[
\begin{equation*}
V_{1}=\frac{\sqrt{\left[\frac{2 g\left(P_{1}-P_{2}\right)}{\rho}+\left(Z_{1}-Z_{2}\right)-h_{L}\right]}}{\left(\frac{A_{1}}{A_{2}}\right)^{2}-1} \tag{9.42}
\end{equation*}
\]

Neglecting the elevation difference \(Z_{1}-Z_{2}\) and the friction loss \(h_{L}\), this equation reduces to the following:
\[
\begin{equation*}
V_{1}=C \frac{\sqrt{\left[\frac{2 g\left(P_{1}-P_{2}\right)}{\rho}\right]}}{\left(\frac{A_{1}}{A_{2}}\right)^{2}-1} \tag{9.43}
\end{equation*}
\]
where
\(V_{1}=\) velocity of gas in the main pipe section before the throat
\(\rho=\) the average gas density
\(A_{1}=\) cross-sectional area of the pipe
\(A_{2}=\) cross-sectional area of the throat
\(C=\) discharge coefficient, dimensionless
The volume flow rate is then calculated by multiplying the velocity by the crosssectional area, resulting in the following equation:
\[
\begin{equation*}
Q=C A_{1} \frac{\sqrt{\left[\frac{2 g\left(P_{1}-P_{2}\right)}{\rho}\right]}}{\left(\frac{A_{1}}{A_{2}}\right)^{2}-1} \tag{9.44}
\end{equation*}
\]

Using the beta ratio, we simplify the above equation as follows:
\[
\begin{equation*}
Q=C A_{1} \frac{\sqrt{\left[\frac{2 g\left(P_{1}-P_{2}\right)}{\rho}\right]}}{\left(\frac{1}{\beta}\right)^{4}-1} \tag{9.45}
\end{equation*}
\]

The discharge coefficient \(C\) is a number less than 1.0 , and it depends on the Reynolds number in the main pipe section. For a Reynolds number greater than \(2 \times 10^{5}\), the value of \(C\) remains constant at 0.984 .

In smaller pipe sizes, such as 2 to 10 in., venturi meters are machined and, therefore, have a better surface finish than the larger rough cast meters. Smaller venturi meters have a \(C\) value of 0.995 for Reynolds numbers larger than \(2 \times 10^{5}\).

\subsection*{9.17 FLOW NOZZLE}

The flow nozzle shown in Figure 9.17 is another device for measuring flow rate. It consists of a main pipe section, followed by a gradual reduction in cross-section area and a short cylindrical section, ending in a gradual expansion to the original pipe size.

The discharge coefficient \(C\) for a flow nozzle is approximately 0.99 for Reynolds numbers greater than \(10^{6}\). At lower Reynolds numbers, due to greater energy loss subsequent to the nozzle throat, \(C\) values are lower.

The discharge coefficient \(C\) depends on the beta ratio and Reynolds number. It is calculated using the following equation:
\[
\begin{equation*}
C=0.9975-6.53 \sqrt{\frac{\beta}{R}} \tag{9.46}
\end{equation*}
\]


> To manometer

Figure 9.17 Flow nozzle.
where
\(\beta=d / D\)
\(R=\) Reynolds number based on the pipe diameter \(D\)

\section*{Example 1}

An orifice meter with 4 in . diameter is installed in a pipe with an inside diameter of 12.09 in . The differential pressure is measured at 30 in . of water, and the static pressure upstream is 600 psig . The gas gravity \(=0.6\) and the gas flowing temperature \(=70^{\circ} \mathrm{F}\). The base temperature and the base pressure are \(60^{\circ} \mathrm{F}\) and 14.7 psia , respectively. Assuming flange taps, calculate the flow rate in standard \(\mathrm{ft}^{3} / \mathrm{h}\).

The barometric pressure is 14.5 psia.

\section*{Solution}

The basic orifice factor \(F_{b}\) is calculated from the AGA 3 appendix as follows:
\[
\begin{aligned}
F_{b} & =3258.5 \\
(h P)^{0.5} & =[30 \times(600+14.5)]^{0.5}=135.78 \\
F_{r} & =1+\frac{0.0207}{135.78}=1.0002 \\
F_{p b} & =\frac{14.73}{14.7}=1.002 \\
F_{t b} & =\frac{60+460}{519.67}=1.006 \\
F_{f f} & =\left(\frac{519.67}{70+460}\right)^{0.5}=0.9902 \\
F_{g r} & =\left(\frac{1}{0.6}\right)^{0.5}=1.291 \\
F_{p v} & =1.0463 \\
\frac{h}{P} & =\frac{30}{614.5}=0.0488 \\
\beta & =\frac{4}{12.09}=0.3309 \\
Y & =0.9995
\end{aligned}
\]
\[
C=3258.5 \times 1.0002 \times 1.002 \times 1.006 \times 0.9902 \times 1.291 \times 1.0463 \times 0.9995=4391.96
\]

Using Equation 9.33, the flow rate is
\[
Q_{v}=4391.96 \times 135.78=596,340 \mathrm{ft}^{3} / \mathrm{h}
\]

\subsection*{9.18 SUMMARY}

In this chapter we covered the topics of valves and flow measurement as they relate to gas pipeline transportation. The various types of valves used and their functions were reviewed. The importance of flow measurement in natural gas pipeline transaction was explained. The predominantly used measuring device known as an orifice meter was discussed in detail. The calculation methodology based on AGA Report No. 3 was reviewed. The venturi meter and the flow nozzle were also discussed.

\section*{PROBLEMS}
1. An orifice meter 2 in . in diameter is installed in a pipe with an inside diameter of 12.09 in . The differential pressure is measured at 20 in . of water, and the static pressure upstream is 500 psig. The gas gravity \(=0.65\) and gas flowing temperature \(=75^{\circ} \mathrm{F}\). The base temperature and the base pressure are \(60^{\circ} \mathrm{F}\) and 14.7 psia, respectively. Assuming pipe taps, calculate the flow rate in standard \(\mathrm{ft}^{3} / \mathrm{h}\). The barometric pressure is 14.6 psia .
2. An orifice meter has a bore size of 1 in . diameter and is installed in a pipe with an inside diameter of 6.125 in . The differential pressure is measured at 10 in . of water, and the static pressure upstream is 300 psig. The gas gravity \(=0.6\) and gas flowing temperature \(=70^{\circ} \mathrm{F}\). The base temperature and the base pressure are \(60^{\circ} \mathrm{F}\) and 14.7 psia , respectively. Assuming flange taps, calculate the flow rate in standard \(\mathrm{ft}^{3} / \mathrm{h}\). The barometric pressure is 14.6 psia .

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\section*{Pipeline Economics}

In the previous chapters we explored different scenarios of pipe sizes and pressures to transport natural gas through pipelines from one location to another. Various pressure drop formulas, compression requirements, and HP required were calculated without delving too much into costs of facilities. In this chapter the economic aspects of pipelines will be reviewed. The economic pipe size required for a particular throughput will be arrived at considering the various costs that make up a pipeline system. The initial capital cost of pipeline and ancillary facilities will be discussed, along with the annual operating and maintenance costs. Since pipelines are generally designed to transport gas belonging to one company by another company, a methodology for determining transportation cost or tariff will be analyzed.

A pipeline can be constructed to transport natural gas for the owner of the pipeline, to sell gas to another company, or to transport some other company's gas. These three scenarios represent three major uses of pipeline transportation of natural gas. The economics involved in the selection of pipe diameter, compressor station, and related facilities will vary slightly for each scenario. As an owner company transporting its own gas, minimal facilities will probably be built. However, Department of Transportation (DOT) codes and other regulatory requirements will still have to be met to ensure a safe pipeline operation that will not endanger humans or the environment. In the second scenario, in which a company builds a pipeline to transport its gas and sells the gas at the end of the pipeline to a customer, minimal facilities will be constructed without too much regulatory control. In the third scenario, a pipeline company constructs and operates a pipeline for the purpose of transporting gas belonging to other companies. This will be under the jurisdiction of the Federal Energy Regulatory Commission (FERC) or a state agency such as the Public Utilities Commission (PUC) in California or the Texas Railroad Commission in Texas. An interstate pipeline in which the pipeline crosses one or more state boundaries will be regulated by the FERC. A pipeline that is intrastate, such as wholly within California, will be subject to PUC rules and not FERC. Such regulatory requirements impose strict guidelines on the type and number of facilities and costs that may be passed on to the customer requesting gas transportation. These regulatory requirements will dictate that excessive capital facilities not be built and the amortized cost passed on
to the customers. Whereas a private pipeline company transporting its own gas may build in extra compressor units as spares to ensure uninterrupted operation in the event of equipment failure, FERC-regulated pipelines may not be able to do so. Thus, pipeline economics will differ slightly from case to case.

In this chapter we will not discuss other modes of transportation of gas, such as truck transport of pressurized gas containers. The general economic principles discussed here are applicable to private unregulated pipelines as well as FERC-regulated pipelines used for interstate transportation of natural gas.

\subsection*{10.1 COMPONENTS OF COST}

In a gas pipeline system the major components that contribute to the initial capital cost are the pipeline, compressor stations, mainline valve stations and metering facilities, telecommunications, and supervisory control and data acquisition (SCADA). Other costs include environmental and permitting costs, right of way (ROW), acquisition cost, engineering and construction management, legal and regulatory costs, contingency, and allowance for funds used during construction (AFUDC).

The recurring annual costs will include operating and maintenance (O\&M) costs, fuel, energy and utility costs, rental, permitting, and annual right of way costs. The O\&M costs will include payroll and general and administrative (G\&A) costs.

In any pipeline system constructed to provide transportation of gas, there will be capital costs and annual operating costs. If we decide on a useful life of the pipeline (say, 30 or 40 years) we can annualize all costs and also determine the revenue stream necessary to amortize the total investment in the pipeline project. The revenue earned after expenses and taxes plus a percentage for profit divided by the volume transported will give the transportation tariff necessary. The calculation of capital cost, operating cost, and transportation tariff will be illustrated using an example.

Throughout this chapter we will need to convert annual cash flows or expenses into present value and vice versa. A useful equation relating the present value of a series of annual payments over a number of years at a specified interest rate is as follows:
\[
\begin{equation*}
P V=\frac{R}{i}\left(1-\frac{1}{(1+i)^{n}}\right) \tag{10.1}
\end{equation*}
\]
where
\(P V=\) present value, \(\$\)
\(R=\) series of cash flows, \(\$\)
\(i=\) interest rate, decimal value
\(n\) = number of periods, years
For example, \(\$ 10,000\) in annual payments for 20 years at an annual interest rate of \(10 \%\) results in a present value of
\[
P V=\frac{10,000}{0.10}\left(1-\frac{1}{(1+0.10)^{20}}\right)=\$ 85,136
\]

Similarly, we can convert a present value of \(\$ 10\) million into an annualized cost based on \(8 \%\) interest for 30 years as follows. From Equation 10.1,
\[
10,000,000=\frac{R}{0.08}\left(1-\frac{1}{(1.08)^{30}}\right)
\]

Solving for the annual cost \(R\), we get
\[
R=\$ 888,274
\]

Next, we will calculate the cost of service and transportation tariff using a simple example.

\section*{Example 1}

A natural gas pipeline transports 100 MMSCFD at a load factor of \(95 \%\). The capital cost is estimated at \(\$ 60\) million and the annual operating cost is \(\$ 5\) million. Amortizing the capital at \(10 \%\) for a project life of 25 years, calculate the cost of service and transportation tariff for this pipeline.

Solution

All costs will be converted to annualized values for a 25 -year project life and \(10 \%\) interest rate. This will be the cost of service on an annual basis. When this cost is divided by the annual pipeline throughput, we obtain the transportation tariff.

The capital cost of \(\$ 60\) million is first converted to annual cash flow at a \(10 \%\) interest rate for a period of 25 years. Using Equation 10.1,
\[
\text { Annualized capital cost }=\frac{60 \times 0.10}{\left[1-\frac{1}{(1.10)^{25}}\right]}=\$ 6.61 \text { million }
\]

This assumes zero salvage value at the end of the 25 -year useful life of the pipeline.

Therefore, for a project life of 25 years and a discount rate of \(10 \%\), the capital cost of \(\$ 60\) million is equivalent to annual cost of \(\$ 6.61\) million. Adding the annual operating cost of \(\$ 5\) million, the total annual cost is \(\$ 6.61+\$ 5=\$ 11.61\) million per year. This annual cost is defined as the cost of service incurred each year. Actually, to be accurate, we should take into account several other factors such as the tax rate, depreciation of assets, and profit margin to arrive at a true cost of service.

The transportation tariff is defined as the cost of service divided by the annual volume transported. At a \(95 \%\) load factor and flow rate of 100 MMSCFD, the transportation tariff is
\[
\text { Tariff }=\frac{\$ 11.61 \times 10^{6} \times 10^{3}}{100 \times 10^{6} \times 365 \times 0.95}=\$ 0.3348 \text { per MCF }
\]

In other words, for this pipeline, every MCF of gas transported requires a payment of approximately 33.5 cents to the pipeline owner that provides the transportation. This is a very rough and simplistic calculation of an example of tariff. In reality, we must take into account many other factors to arrive at an accurate cost of service. For example, the annual operating cost will vary from year to year over the life of the pipeline, due to inflation and other reasons. Taxes, depreciation of assets, and salvage value at the end of the life of the pipeline must also be considered. Nevertheless, the preceding analysis gives a quick overview of the approach used to calculate a rough value of the transportation cost.

\subsection*{10.2 CAPITAL COSTS}

The capital cost of a pipeline project consists of the following major components:
- Pipeline
- Compressor stations
- Mainline valve stations
- Meter stations
- Pressure regulator stations
- SCADA and telecommunication
- Environmental and permitting
- Right of way acquisitions
- Engineering and construction management

In addition, there are other costs such as allowance for funds used during construction (AFUDC) and contingency. Each of the preceding major categories of capital cost will be discussed next.

\subsection*{10.2.1 Pipeline}

The pipeline cost consists of those costs associated with the pipe material, coating, pipe fittings, and the actual installation or labor cost. In Chapter 6, we introduced a simple formula to calculate the weight of pipe per unit length. From this and the pipe length, the total tonnage of pipe can be calculated. Given the cost per ton of pipe material, the total pipe material cost can be calculated. Knowing the construction cost per unit length of pipe, we can also calculate the labor cost for installing the pipeline. The sum of these two costs is the pipeline capital cost.

Using Equation 6.11 for pipe weight, the cost of pipe required for a given pipeline length is found from
\[
\begin{equation*}
P M C=\frac{10.68(D-T) T L C \times 5280}{2000} \tag{10.2}
\end{equation*}
\]
where
\(P M C=\) pipe material cost, \(\$\)
\(L \quad=\) length of pipe, mi
\(D \quad=\) pipe outside diameter, in.
\(T\) = pipe wall thickness, in.
\(C=\) pipe material cost, \$/ton
In SI units,
\[
\begin{equation*}
P M C=0.0246(D-T) T L C \tag{10.3}
\end{equation*}
\]
where
\(P M C=\) pipe material cost, \(\$\)
\(L \quad=\) length of pipe, km
\(D \quad=\) pipe outside diameter, mm
\(T\) = pipe wall thickness, mm
\(C=\) pipe material cost, \$/metric ton

Generally, pipe will be supplied externally coated and wrapped. Therefore, we must add this cost or a percentage to the bare pipe cost to account for the extra cost and the delivery cost to the construction site. In the absence of actual cost, we may increase the bare pipe cost by a small percentage, such as \(5 \%\). For example, using Equation 10.2 for a 100 mi pipeline, NPS 20 with 0.500 in . wall thickness, the total pipe cost, based on \(\$ 800\) per ton, is
\[
P M C=\frac{10.68(20-0.5) 0.5 \times 100 \times 800 \times 5280}{2000}=\$ 21.99 \text { million }
\]

If the pipe is externally coated and wrapped and delivered to the field at an extra cost of \(\$ 5\) per ft , this cost can be added to the bare pipe cost as follows:

Pipe coating and wrapping cost \(=\$ 5 \times 5280 \times 100=\$ 2.64\) million
Therefore, the total pipe cost becomes
\[
\$ 21.99+\$ 2.64=\$ 24.63 \text { million }
\]

The labor cost to install the pipeline can be represented in dollars per unit length of pipe. For example, the labor cost might be \(\$ 60\) per ft or \(\$ 316,800\) per mi of pipe for a particular size pipe in a certain construction environment. This number will depend on whether the pipeline is installed in open country, fields, or city streets. Such numbers are generally obtained from contractors who will take into consideration the difficulty of trenching, installing pipe, and backfilling in the area of construction. For estimation purposes, there is a wealth of historical data available for construction cost for various pipe sizes. Sometimes the pipe installation cost is expressed in terms of dollars per in. diameter per mi of pipe. For example, an NPS 16 pipe might have an installation cost of \(\$ 15,000\) per in.-diameter-mile. Thus, if 20 mi of NPS 16 pipe are to be installed, we estimate the labor cost as follows:

Pipe installation cost \(=\$ 15,000 \times 16 \times 20=\$ 4.8\) million

Table 10.1 Typical Pipeline Installation Costs
\begin{tabular}{cc}
\hline \begin{tabular}{c} 
Pipe Diameter, \\
in.
\end{tabular} & \begin{tabular}{c} 
Average Cost, \\
S/in.-dia/mi
\end{tabular} \\
\hline 8 & 18,000 \\
10 & 20,000 \\
12 & 22,000 \\
16 & 14,900 \\
20 & 20,100 \\
24 & 33,950 \\
30 & 34,600 \\
36 & 40,750 \\
\hline
\end{tabular}

If we convert this cost on a unit length basis, we get
\[
\text { Pipe installation cost }=\frac{4.8 \times 10^{6}}{20 \times 5280}=\$ 45.45 \text { per ft }
\]

Table 10.1 shows typical installation costs for pipelines. These numbers must be verified by discussions with construction contractors who are familiar with the construction location.

Several other construction costs must be added to the installation costs for straight pipe. These expeditures include costs for road, highway, and railroad crossings and stream and river crossings. These costs can be provided as lump sum numbers, which can be added to the pipeline installation costs to come up with a total pipeline construction cost. For example, a pipeline might include two road and highway crossings that total \(\$ 300,000\) in addition to a couple of river crossings costing \(\$ 1\) million. Compared to the installation cost of a long-distance pipeline, the road and river crossings total might be a small percentage.

\subsection*{10.2.2 Compressor Stations}

In order to provide transportation of gas through a pipeline, we have to install one or more compressor stations to provide the necessary gas pressure. Once we decide on the details of the compressor station equipment and piping, a detailed bill of materials can be developed from the engineering drawings. Based upon quotations from equipment vendors, a detailed cost estimate of the compressor stations can be developed. In the absence of vendor data and in situations where a rough order of magnitude costs for compressor stations is desired, we can use an all-inclusive price of dollars per installed HP. For example, using an installed cost of \(\$ 2000\) per HP, for a 5000 HP compressor station, the capital cost will be estimated as follows:
\[
\text { Compressor station cost }=2000 \times 5000=\$ 10 \text { million }
\]

In the above calculations, the all-inclusive number of \(\$ 2000\) per installed HP is expected to include material and equipment cost and the labor cost for installing the compressor equipment, piping, valves, instrumentation, and controls within the
compressor stations. Generally, the \(\$ / \mathrm{HP}\) number decreases as the size of the compressor HP increases. Thus, a 5000 HP compressor station might be estimated on the basis of \(\$ 2000\) per HP, whereas a 20,000 HP compressor station will be estimated at an installed cost of \(\$ 1500\) per HP. These numbers are mentioned for illustration purposes only. Actual \$/HP values must be obtained from historical pipeline cost data and in consultation with compressor station construction contractors and compressor station equipment vendors. Generally, the pipeline and compressor station costs constitute the bulk of the total pipeline project cost.

\subsection*{10.2.3 Mainline Valve Stations}

Mainline block valves are installed to isolate sections of a pipeline for safety reasons and maintenance and repair. In the event of a pipeline rupture, the damaged pipeline section can be isolated by closing off the mainline valves on either side of the rupture location. For mainline valve stations installed at specified intervals along the pipeline, the cost of facilities can be specified as a lump sum figure that includes the mainline valve and operator, blowdown valves and piping, and other pipe and fittings that constitute the entire block valve installation. Generally, a lump sum figure can be obtained for a typical mainline block valve installation from a construction contractor. For example, an NPS 16 mainline valve installation might be estimated at \(\$ 100,000\) per site. In a 100 mi , NPS 16 pipeline, DOT code requirements might dictate that a mainline valve be installed every 20 mi . Therefore, in this case there would be six mainline valves for a 100 mi pipeline. At \(\$ 100,000\) per site, the total installed cost of all mainline valve stations will be \(\$ 600,000\). This will be added to the capital cost of the pipeline facilities.

\subsection*{10.2.4 Meter Stations and Regulators}

Meter stations are installed for measuring the gas flow rate through the pipeline. These meter stations will consist of meters, valves, fittings, instrumentation, and controls. Meter stations can also be estimated as a fixed price, including material and labor for a particular site. For example, a 10 in . meter station might cost \(\$ 300,000\) lump sum. If there are four such meter stations on a 100 mi gas pipeline, the total meter station cost will be \(\$ 1.2\) million. The meter station costs, like the mainline valve station costs, will be added to the pipeline cost.

Pressure regulating stations are installed at some locations on a gas pipeline to reduce the pressure for delivery to a customer or to protect a section of a pipeline with a lower MOP. Such pressure regulating stations can also be estimated as a lump sum per site and added to the capital cost of the pipeline.

\subsection*{10.2.5 SCADA and Telecommunication System}

Typically on a gas pipeline, the pressures, flow rates, and temperatures are monitored along the pipeline by means of electronic signals sent from remote terminal units (RTU) on various valves and meters to a central control center via telephone lines or
microwave or satellite communication systems. The term supervisory control and data acquisition (SCADA) is used to refer to these facilities. SCADA is used to remotely monitor, operate, and control a gas pipeline system from a central control center. In addition to monitoring valve status, flows, temperatures, and pressures along a pipeline, SCADA also monitors the compressor stations. In many cases, starting and stopping of compressor units are performed remotely using SCADA. The cost of SCADA facilities range from \(\$ 2\) million to \(\$ 5\) million or more, depending on the pipeline length, number of compressor stations, and the number of mainline valves and meter stations. Sometimes this category is estimated as a percentage of the total project cost, such as 2 to \(5 \%\).

\subsection*{10.2.6 Environmental and Permitting}

The environmental and permitting costs are those costs that are associated with the modifications to pipeline, compressor stations, and valve and meter stations to ensure that these facilities do not pollute the atmosphere, streams, and rivers or damage ecosystems including the flora and fauna, fish and game, and endangered species. Many sensitive areas, such as Native American religious and burial sites, must be considered and allowances must be made for mitigation of habitat in certain areas. Permitting costs can include those costs associated with changes needed to compression equipment, pipeline alignment such that toxic emissions from pipeline facilities do not endanger the environment, humans, and plant and animal life. In many cases, these costs include acquisition of land to compensate for the areas that were disturbed due to pipeline construction. Such lands acquired will be allocated for public use, such as parks and wildlife preserves. Permitting costs will also include an environmental study, the preparation of an environmental impact report, and permits for road crossings, railroad crossings, and stream and river crossings. These environmental and permitting costs on a gas pipeline project may range between 10 and \(15 \%\) of total project costs.

\subsection*{10.2.7 Right of Way Acquisitions}

The right of way (ROW) for a pipeline is acquired from private parties and state and local government and federal agencies for a fee. This fee might be a lump sum payment at the time of acquisition with additional annual fees to be paid for a certain duration. For example, the right of way can be acquired from private farms, cooperatives, the Bureau of Land Management (BLM), and railroads. The initial cost for acquiring the ROW will be included in the capital cost of the pipeline. The annual rent or lease payment for land will be considered an expense. The latter will be included in the annual costs, such as operating costs. As an example, the ROW acquisition costs for a gas pipeline might be \(\$ 30\) million. This cost would be added to the total capital costs of the gas pipeline. Also, there might be annual ROW lease payments of \(\$ 300,000\) a year, which would be added to other annual costs such as operating and maintenance costs and administrative costs. For most gas pipelines, the initial ROW costs will be in the range of 6 to \(10 \%\) of the total project costs.

\subsection*{10.2.8 Engineering and Construction Management}

Engineering costs are those costs that pertain to the design and preparation of drawings for the pipeline, compressor stations, and other facilities. This will include both preliminary and detailed engineering design costs, including development of specifications, manuals, purchase documents, equipment inspection, and other costs associated with materials and equipment acquisition for the project. The construction management costs include field personnel cost, rental facilities, office equipment, transportation, and other costs associated with overseeing and managing the construction effort for the pipeline and facilities. On a typical pipeline project, engineering and construction management costs range from 15 to \(20 \%\) of the total pipeline project cost.

\subsection*{10.2.9 Other Project Costs}

In addition to the major cost categories discussed in the preceding sections, there are other costs that should be included in the total pipeline project cost. These include legal and regulatory costs necessary for filing an application with the FERC and state agencies that have jurisdiction over interstate and intrastate transportation of natural gas, as well as a contingency costs intended to cover categories not considered or not envisioned when the project was conceptualized. As the project is engineered, new issues and problems might surface that require additional funds. These are generally included in the category of contingency cost. The final category of cost, referred to as allowance for funds used during construction (AFUDC), is intended to cover the cost associated with financing the project during various stages of construction. Contingency and AFUDC costs can range between 15 and \(20 \%\) of the total project cost. Table 10.2 shows a cost breakdown for a typical natural gas pipeline project.

Table 10.2 Cost Breakdown for a Typical Natural Gas Pipeline Project
\begin{tabular}{rlrr}
\hline Description & & Million \$ \\
\hline 1 & Pipeline & & 160.00 \\
2 & Compressor stations & & 20.00 \\
3 & Mainline valve stations & & 1.20 \\
4 & Meter stations & 1.20 \\
5 & Pressure regulator stations & 0.10 \\
6 & SCADA and telecommunications & 10 to \(15 \%\) & 5.48 \\
7 & Environmental and permitting & 6 to \(10 \%\) & 14.90 \\
8 & Right of way acquisiton & 15 to \(20 \%\) & 36.50 \\
9 & Engineering and construction management & \(10 \%\) & 26.10 \\
10 & Contingency & & 287.08 \\
& Sub-Total & & 5.00 \\
11 & Working capital & \(5 \%\) & 14.35 \\
& AFUDC & & 306.43 \\
& Total & & \\
\hline
\end{tabular}

\subsection*{10.3 OPERATING COSTS}

Once the pipeline, compressor stations, and ancillary facilities are constructed and the pipeline is put into operation, there will be annual operating costs over the useful life of the pipeline, which might be 30 to 40 years or more. These annual costs consist of the following major categories:
- Compressor station fuel or electrical energy cost
- Compressor station equipment maintenance and repair costs
- Pipeline maintenance costs, such as pipe repair, relocation, aerial patrol, and monitoring
- SCADA and telecommunication
- Valve, regulator, and meter station maintenance
- Utility costs, such as water and natural gas
- Annual or periodic environmental and permitting costs
- Lease, rental, and other recurring right of way costs
- Administrative and payroll costs

Compressor station costs include periodic equipment maintenance and overhaul costs. For example, a gas turbine-driven compressor unit may have to be overhauled every 18 to 24 months. Table 10.3 shows the annual operating cost of a typical gas pipeline.

\section*{Example 2}

A new pipeline is being constructed to transport natural gas from a gas processing plant to a power plant 100 mi away Two project phases are envisioned. During the first phase lasting 10 years, the amount of gas shipped is expected to be a constant

Table 10.3 Annual Operating Cost of a Typical Gas Pipeline
\begin{tabular}{rlr}
\hline Description & \$ per year \\
\hline 1 & Salaries & 860,000 \\
2 & Payroll overhead (20\%) & 172,000 \\
3 & Admin and general (50\%) & 516,000 \\
4 & Vehicle expense & 72,800 \\
5 & Office expenses (6\%) & 92,880 \\
6 & Misc materials and tools & 100,000 \\
7 & Compressor station maintenance & \\
8 & Consumable materials & 50,000 \\
9 & Periodic maintenance & 150,000 \\
10 & ROW payments & 350,000 \\
11 & Utilities & 150,000 \\
12 & Gas control & 100,000 \\
13 & SCADA contract install and maintenance & 200,000 \\
14 & Internal corrosion inspection (\$750,000/3 years) & 250,000 \\
15 & Cathodic protection survey & 100,000 \\
\hline & Total O\&M & \(3,163,680\) \\
\hline
\end{tabular}
volume of 120 MMSCFD at a \(95 \%\) load factor. A pipe size of NPS \(16,0.250\) in. wall thickness is required to handle the volumes with two compressor stations with a total of 5000 HP . The total pipeline cost can be estimated at \(\$ 800,000\) per mi and compressor station cost at \(\$ 2000\) per HP installed. The annual operating costs are estimated at \(\$ 8\) million. The pipeline construction project will be financed by borrowing \(80 \%\) of the required capital at an interest rate of \(6 \%\). The regulatory rate of return allowed on equity capital is \(14 \%\). Consider a project life of 20 years and an overall tax rate of \(40 \%\).
(a) Calculate the annual cost of service for this pipeline and the transportation tariff in \(\$ / \mathrm{MCF}\).
(b) The second phase, lasting the next 10 years, is projected to increase throughput to 150 MMSCFD. Calculate the transportation tariff for the second phase, considering the capital cost to increase by \(\$ 20\) million and the annual cost to increase to \(\$ 10\) million, with the same load factor as phase 1 .

Solution
(a) First, calculate the total capital cost of facilities of phase 1.
\[
\text { Pipeline cost }=\$ 800,000 \times 100=\$ 80 \text { million }
\]

Compressor station cost \(=\$ 2000 \times 5000=\$ 10\) million
\[
\text { Total capital cost }=\$ 80+\$ 10=\$ 90 \text { million }
\]
\(80 \%\) of this capital of \(\$ 90\) million will be borrowed at \(6 \%\) interest for 20 years.

From Equation 10.1, the annual cost to amortize the loan is
\[
\text { Loan amortization cost }=\frac{90 \times 0.8 \times 0.06}{1-\left(\frac{1}{1.06}\right)^{20}}=\$ 6.28 \text { million }
\]

Therefore, we need to build into the cost of service the annual payment of \(\$ 6.28\) million to retire the debt of \(\$ 72\) million ( \(80 \%\) of \(\$ 90\) million) over the project life of 20 years. On the remaining capital (equity) of ( \(\$ 90-\$ 72\) ) million or \(\$ 18\) million, a \(14 \%\) rate of return per year is allowed. Therefore, \(14 \%\) of \(\$ 18\) million can be included in the cost of service to account for the equity capital.

Annual revenue on equity capital \(=0.14 \times \$ 18\) million \(=\$ 2.52\) million Since the tax rate is \(40 \%\), the adjusted annual revenue on equity capital is
\[
\frac{\$ 2.52 \text { million }}{1-0.4}=\$ 4.2 \text { million }
\]

Next, add the operating cost of \(\$ 8\) million per year to the annual costs for debt and equity just calculated to arrive at the annual cost of service as follows:

Annual payment to retire debt \(=\$ 6.28\) million
Annual revenue on equity capital \(=\$ 4.2\) million
Annual operating cost \(=\$ 8\) million

Therefore,
\[
\text { Annual cost of service }=6.28+4.2+8=\$ 18.48 \text { million }
\]

The transportation tariff at 120 MMSCFD and \(95 \%\) load factor is
\[
\text { Tariff }=\frac{18.48 \times 10^{6} \times 10^{3}}{120 \times 10^{6} \times 365 \times 0.95}=\$ 0.4441 \text { per MCF. }
\]
(b) In the second phase, which lasts 10 years, the capital cost increases by \(\$ 20\) million. The extra \(\$ 20\) million will be assumed to be financed by \(80 \%\) debt and \(20 \%\) equity as before. The annual cost to amortize the debt is
\[
\text { Loan amortization cost }=\frac{20 \times 0.8 \times 0.06}{1-\left(\frac{1}{1.06}\right)^{10}}=\$ 2.17 \text { million }
\]

The remaining capital of ( \(\$ 20-\$ 16\) ) or \(\$ 4\) million is equity that, according to regulatory guidelines, can earn \(14 \%\) interest. It must be noted that the interest rate and ROR used in this example are approximate and only for the purpose of illustration. The actual ROR allowed on a particular pipeline will depend on various factors such as the state of the economy, current FERC regulations, or state laws, and can range from as low as \(8 \%\) to as high as \(16 \%\) or more. Similarly, the interest rate of \(6 \%\) used for debt amortization is also an illustrative number. The actual interest rate on debt will depend on various factors such as the state of the economy, money supply, and the federal interest rate charged by Federal Reserve (prime rate). This rate will vary with the country where the pipeline is built and the multinational bank that might finance the pipeline project.

For phase 2, the annual revenue on equity capital is
\[
4 \times 0.14=\$ 0.56 \text { million }
\]

Accounting for a \(40 \%\) tax rate, the adjusted annual revenue on equity capital is
\[
\frac{\$ 0.56}{1-0.4}=\$ 0.93 \text { million }
\]

Therefore, for phase 2, the increase in capital of \(\$ 20\) million and operating cost of \(\$ 2\) million will result in an increase in cost of service of
\[
\text { Annual cost of service }=\$ 2.17+0.93+2=\$ 5.1 \text { million }
\]

In summary, for phase 2, the total cost of services is
\[
\$ 18.48+\$ 5.1=\$ 23.58 \text { million }
\]

At a flow rate of 150 MMSCFD and \(95 \%\) load factor, the tariff for phase 2 is
\[
\frac{23.58 \times 10^{6} \times 10^{3}}{150 \times 10^{6} \times 365 \times 0.95}=\$ 0.4534 \text { per MCF }
\]

\subsection*{10.4 DETERMINING ECONOMIC PIPE SIZE}

For a particular pipeline transportation application, there is an economic or optimum pipe diameter that will result in the lowest cost of facilities. For example, a pipeline that requires 100 MMSCFD gas to be transported from a source location to a destination location may be constructed of a wide range of pipe materials and diameters. We may choose to use NPS 14 , NPS 16, or NPS 18 pipe or any other pipe size for this application. Using the smallest-diameter pipe will cause the greatest pressure drop and the highest HP requirement for a given volume flow rate. The largest pipe size will result in the lowest pressure drop and, hence, require the least HP. Therefore, the NPS 14 system will be the lowest in pipe material cost and highest in HP required. On the other hand, the NPS 18 system will require the least HP but considerably more pipe material cost due to the difference in pipe weight per unit length. Determining the optimum pipe size for an application will be illustrated in the next example.

\section*{Example 3}

A gas pipeline is to be constructed to transport 150 MMSCFD of natural gas from Dixie to Florence, 120 mi away. Consider three pipe sizes-NPS 14, NPS 16, and NPS 18-all having 0.250 in . wall thickness. Determine the most economical pipe diameter, taking into account the pipe material cost, cost of compressor stations, and fuel costs. The selection of pipe size may be based on a 20 -year project life and a present value ( \(P V\) ) of discounted cash flow at \(8 \%\) per year. Use \(\$ 800\) per ton for pipe material and \(\$ 2000\) per installed HP for compressor station cost. Fuel gas can be estimated at \(\$ 3\) per MCF.

The following information from hydraulic analysis is available:
NPS 14 pipeline: Two compressor stations, 8196 HP total. Fuel consumption is 1.64 MMSCFD.
NPS 16 pipeline: One compressor station, 3875 HP. Fuel consumption is 0.78 MMSCFD.
NPS 18 pipeline: One compressor station, 2060 HP. Fuel consumption is 0.41 MMSCFD.

Solution
First, calculate the capital cost of 120 mi of pipe for each case.

From Equation 10.2, the cost of NPS 14 pipe is
\[
P M C=\frac{10.68(14-0.250) \times 0.250 \times 120 \times 800 \times 5280}{2000}=\$ 9.3 \text { million }
\]

Similarly, the cost of NPS 16 pipe is
\[
P M C=\frac{10.68(16-0.250) \times 0.250 \times 120 \times 800 \times 5280}{2000}=\$ 10.66 \text { million }
\]
and the cost of NPS 18 pipe is
\[
P M C=\frac{10.68(18-0.250) \times 0.250 \times 120 \times 800 \times 5280}{2000}=\$ 12.01 \text { million }
\]

Next, calculate the installed cost of compressor stations for each pipe size.
For NPS 14 pipe, the compressor station cost is
\[
8196 \times 2000=\$ 16.39 \text { million }
\]

For NPS 16 pipe, the compressor station cost is
\[
3875 \times 2000=\$ 7.75 \text { million }
\]

For NPS 18 pipe, the compressor station cost is
\[
2060 \times 2000=\$ 4.12 \text { million }
\]

The operating fuel cost for each case will be calculated next, considering fuel gas at \(\$ 3\) per MCF and 24 -hour-a-day operation for 350 days a year. A shutdown for 15 days per year is allowed for maintenance and any operational upset conditions.

For NPS 14 pipe, the fuel cost is
\[
1.64 \times 10^{3} \times 350 \times 3=\$ 1.72 \text { million per year }
\]

For NPS 16 pipe, the fuel cost is
\[
0.78 \times 10^{3} \times 350 \times 3=\$ 0.82 \text { million per year }
\]

For NPS 18 pipe, the fuel cost is
\[
0.41 \times 10^{3} \times 350 \times 3=\$ 0.43 \text { million per year }
\]

The actual operating cost includes many other items besides the fuel cost. For simplicity, in this example we will only consider the fuel cost. The annual fuel cost for the project life of 20 years will be discounted at \(8 \%\) in each case. This will then be added to the sum of the pipeline and compressor station capital cost to arrive at a present value \((P V)\).

The present value of a series of cash flows, each equal to \(R\) for a period of \(n\) years at an interest rate of \(i \%\), is given by Equation 10.1.

The \(P V\) of NPS 14 fuel cost is, from Equation 10.1,
\[
P V=\frac{1.72}{0.08}\left(1-\frac{1}{(1+0.08)^{20}}\right)=1.72 \times 9.8181=\$ 16.89 \text { million }
\]

The \(P V\) of NPS 16 fuel cost is
\[
P V=0.82 \times 9.8181=\$ 8.05 \text { million }
\]

The \(P V\) of NPS 18 fuel cost is
\[
P V=0.43 \times 9.8181=\$ 4.22 \text { million }
\]

Therefore, adding up all costs, the \(P V\) for NPS 14 is
\[
P V_{14}=9.3+16.39+16.89=\$ 42.58 \text { million }
\]

Adding up all costs, the \(P V\) for NPS 16 is
\[
P V_{16}=10.66+7.75+8.05=\$ 26.46 \text { million }
\]
and adding up all costs, the \(P V\) for NPS 18 is
\[
P V_{18}=12.01+4.12+4.22=\$ 20.35 \text { million }
\]

Therefore, we see that the lowest cost option is NPS 18 pipeline with a \(P V\) of \(\$ 20.35\) million.

In the preceding example, if the flow rate had been lower or higher, the result may be different. For each pipe size, if we were to calculate the HP required at various flow rates and the corresponding fuel consumption, we could generate a graph showing the variation of total cost with flow rate. Obviously, as flow rate is increased, the HP required and fuel consumption also increase. Performing these calculations for different pipe sizes will yield a graph similar to that shown in Figure 10.1. In the next example, we will consider three pipe sizes (NPS 16, NPS 18, and NPS 20) and calculate the capital cost and \(O \& M\) cost for a range of flow rates to develop curves similar to those shown in Figure 10.1 .


Figure 10.1 Pipeline cost vs. flow rate for various pipe sizes.

\section*{Example 4}

For a natural gas pipeline 120 mi long, three pipe sizes were analyzed for flow rate ranges of 50 to 500 MMSCFD using a hydraulic simulation software application, GASMOD (www.systek.us). The following are the pipe sizes and flow rates studied:
\[
\begin{aligned}
& \text { NPS } 16 \text { pipe: flow rates- } 50 \text { to } 200 \text { MMSCFD } \\
& \text { NPS } 18 \text { pipe: flow rates- } 50 \text { to } 300 \text { MMSCFD } \\
& \text { NPS } 20 \text { pipe: flow rates- } 100 \text { to } 500 \text { MMSCFD }
\end{aligned}
\]

The wall thickness was 0.250 in. for NPS 16 and NPS 18 and 0.500 in. for NPS 20.

From the hydraulic simulation, the number of compressor stations required, HP, and fuel consumption were obtained as shown in Table 10.4.

Table 10.4 Hydraulic Simulation Results for Three Pipe Sizes
\begin{tabular}{cccc}
\hline \multicolumn{4}{c}{ NPS 16 } \\
\begin{tabular}{c} 
Flow Rate, \\
MMSCFD
\end{tabular} & \begin{tabular}{c} 
Compressor \\
Stations
\end{tabular} & \begin{tabular}{c} 
Total \\
HP
\end{tabular} & \begin{tabular}{c} 
Fuel, \\
MMSCFD
\end{tabular} \\
\hline 50 & 1 & 49 & 0.01 \\
100 & 1 & 1072 & 0.21 \\
150 & 1 & 3875 & 0.78 \\
175 & 2 & 5705 & 1.14 \\
200 & 2 & 9203 & 1.84
\end{tabular}

NPS 18
\begin{tabular}{ccrc}
\hline \begin{tabular}{c} 
Flow Rate, \\
MMSCFD
\end{tabular} & \begin{tabular}{c} 
Compressor \\
Stations
\end{tabular} & \multicolumn{1}{c}{\begin{tabular}{c} 
Total \\
HP
\end{tabular}} & \begin{tabular}{c} 
Fuel, \\
MMSCFD
\end{tabular} \\
\hline 50 & 1 & 49 & 0.01 \\
100 & 1 & 209 & 0.04 \\
150 & 1 & 2060 & 0.41 \\
175 & 1 & 3394 & 0.68 \\
200 & 1 & 4954 & 1 \\
250 & 2 & 9348 & 1.87 \\
300 & 2 & 17902 & 3.58
\end{tabular}

NPS 20
\begin{tabular}{cccc}
\hline \begin{tabular}{c} 
Flow Rate, \\
MMSCFD
\end{tabular} & \begin{tabular}{c} 
Compressor \\
Stations
\end{tabular} & \begin{tabular}{c} 
Total \\
HP
\end{tabular} & \begin{tabular}{c} 
Fuel, \\
MMSCFD
\end{tabular} \\
\hline 100 & 1 & 98 & 0.02 \\
150 & 1 & 1053 & 0.21 \\
175 & 1 & 2057 & 0.41 \\
200 & 1 & 3281 & 0.66 \\
250 & 1 & 6312 & 1.26 \\
300 & 2 & 10519 & 2.1 \\
400 & 2 & 31401 & 6.28 \\
500 & 2 & 73207 & 14.64 \\
\hline
\end{tabular}

Develop annualized costs for each pipe size and flow rate using the following assumptions:

The capital cost of the pipe material is based on \(\$ 800\) per ton.
For pipe installation cost, use the following:

NPS 16: \(\$ 50\) per foot
NPS 18: \(\$ 60\) per foot
NPS 20: \(\$ 80\) per foot
For compressor station capital cost, use \(\$ 2000\) per installed HP.
Fuel gas can be assumed to be \(\$ 3\) per MCF.
The project life is 20 years, and the interest rate for discounting cash flow is \(8 \%\).
Add \(40 \%\) to the pipe and compressor capital costs to account for miscellaneous costs such as meter stations; valves; ROW; environmental, engineering, and construction management; and contingency. The pipeline is assumed to be operational 350 days a year.

\section*{Solution}

From the given hydraulic simulation data, using the assumptions listed, we develop the total capital cost of pipe, compressor station, and miscellaneous costs.

The pipe material cost is calculated from Equation 10.2, using \(\$ 800\) per ton for pipe material cost, as follows:

For NPS 16 pipe,

Pipe material cost \(=\frac{10.68(16-0.25) 0.25 \times 120 \times \$ 800 \times 5280}{2000}=\$ 10.66\) million

Similarly, for NPS 18 pipe,

Pipe material cost \(=\frac{10.68(18-0.25) 0.25 \times 120 \times \$ 800 \times 5280}{2000}=\$ 12.01\) million
and NPS 20 pipe material cost is

Pipe material cost \(=\frac{10.68(20-0.5) 0.5 \times 120 \times \$ 800 \times 5280}{2000}=\$ 26.39\) million
These costs are shown in Table 10.5 through Table 10.7.

Table 10.5 NPS 16 Pipe Cost Summary
\begin{tabular}{|c|c|c|c|c|c|c|c|c|c|c|c|c|c|c|}
\hline \multicolumn{15}{|c|}{NPS 16} \\
\hline Flow Rate & Number of Compressor Stations & Total HP & Fuel, MMSCFD & Fuel, \$/yr & Pipe Material, \$ & Pipe Labor, \$ & Total Pipe Cost, \$ & Compressor Station Cost, \$ & Misc Cost, \$ & Total Capital, \$ & O\&M Cost, \$/yr & Annualized Capital, \$/yr & Total Annual Cost, \$ & Annual Cost, \$/MCF \\
\hline 50 & 1 & 49 & 0.01 & 0.01 & 10.66 & 31.68 & 42.34 & 0.098 & 16.97 & 59.41 & 2.00 & 6.05 & 8.06 & 0.4607 \\
\hline 100 & 1 & 1072 & 0.21 & 0.22 & 10.66 & 31.68 & 42.34 & 2.144 & 17.79 & 62.27 & 2.00 & 6.34 & 8.56 & 0.2447 \\
\hline 150 & 1 & 3875 & 0.78 & 0.82 & 10.66 & 31.68 & 42.34 & 7.75 & 20.04 & 70.12 & 2.00 & 7.14 & 9.96 & 0.1897 \\
\hline 175 & 2 & 5705 & 1.14 & 1.20 & 10.66 & 31.68 & 42.34 & 11.41 & 21.50 & 75.25 & 3.00 & 7.66 & 11.86 & 0.1937 \\
\hline 200 & 2 & 9203 & 1.84 & 1.93 & 10.66 & 31.68 & 42.34 & 18.406 & 24.30 & 85.04 & 3.00 & 8.66 & 13.59 & 0.1942 \\
\hline
\end{tabular}

Notes:
Pipe material cost \(=\$ 800 /\) ton
Pipe labor cost = \$50/ft for NPS 16
Compressor station cost \(=\$ 2000\) per installed HP
Miscellaneous cost is \(40 \%\) of pipe and compressor station cost
Operating cost based on 350 days per year
Fuel cost is \(\$ 3\) per MCF
Capital cost is annualized at \(8 \%\) interest for 20-year project life
Table values in millions of dollars

Table 10.6 NPS 18 Pipe Cost Summary
\begin{tabular}{|c|c|c|c|c|c|c|c|c|c|c|c|c|c|c|}
\hline \multicolumn{15}{|c|}{NPS 18} \\
\hline \begin{tabular}{l}
Flow \\
Rate
\end{tabular} & Number of Compressor Stations & Total HP & Fuel, MMSCFD & Fuel, \$/yr & Pipe Material, \$ & Pipe Labor, \$ & Total Pipe Cost, \$ & Compressor Station Cost, \$ & Misc Cost, \$ & Total Capital, \$ & O\&M Cost, \$/yr & Annualized Capital, \$/yr & Total Annual Cost, \$ & Annual Cost, \$/MCF \\
\hline 50 & 1 & 49 & 0.01 & 0.01 & 12.01 & 38.02 & 50.03 & 0.098 & 20.05 & 70.18 & 2.00 & 7.15 & 9.16 & 0.5233 \\
\hline 100 & 1 & 209 & 0.04 & 0.04 & 12.01 & 38.02 & 50.03 & 0.418 & 20.18 & 70.62 & 2.00 & 7.19 & 9.24 & 0.2639 \\
\hline 150 & 1 & 2060 & 0.41 & 0.43 & 12.01 & 38.02 & 50.03 & 4.12 & 21.66 & 75.81 & 2.00 & 7.72 & 10.15 & 0.1934 \\
\hline 175 & 1 & 3394 & 0.68 & 0.71 & 12.01 & 38.02 & 50.03 & 6.788 & 22.73 & 79.54 & 2.00 & 8.10 & 10.82 & 0.1766 \\
\hline 200 & 1 & 4954 & 1.00 & 1.05 & 12.01 & 38.02 & 50.03 & 9.908 & 23.97 & 83.91 & 2.00 & 8.55 & 11.60 & 0.1657 \\
\hline 250 & 2 & 9348 & 1.87 & 1.96 & 12.01 & 38.02 & 50.03 & 18.696 & 27.49 & 96.21 & 3.00 & 9.80 & 14.76 & 0.1687 \\
\hline 300 & 2 & 17902 & 3.58 & 3.76 & 12.01 & 38.02 & 50.03 & 35.804 & 34.33 & 120.16 & 3.00 & 12.24 & 19.00 & 0.1809 \\
\hline
\end{tabular}

Notes:
Pipe material cost \(=\$ 800 /\) ton
Pipe labor cost \(=\$ 60 / f t\) for NPS 18
Compressor station cost = \$2000 per installed HP
Miscellaneous cost is \(40 \%\) of pipe and compressor station cost
Operating cost based on 350 days per year
Fuel cost is \(\$ 3\) per MCF
Capital cost is annualized at \(8 \%\) interest for 20 -year project life
Table values in millions of dollars

Table 10.7 NPS 20 Pipe Cost Summary
\begin{tabular}{|c|c|c|c|c|c|c|c|c|c|c|c|c|c|c|}
\hline \multicolumn{15}{|c|}{NPS 20} \\
\hline Flow Rate & Number of Compressor Stations & Total HP & Fuel, MMSCFD & Fuel, \$/yr & Pipe Material, \$ & Pipe Labor, \$ &  & Compressor Station Cost, \$ & Misc Cost, \$ & Total
Capital,
\(\$\) & O\&M Cost, \$/yr & Annualized Capital, \$/yr & Total Annual Cost, \$ & Annual Cost, \$/MCF \\
\hline 100 & 1 & 98 & 0.02 & 0.02 & 26.39 & 50.69 & 77.08 & 0.196 & 30.91 & 108.18 & 2.00 & 11.02 & 13.04 & 0.3726 \\
\hline 150 & 1 & 1053 & 0.21 & 0.22 & 26.39 & 50.69 & 77.08 & 2.106 & 31.67 & 110.86 & 2.00 & 11.29 & 13.51 & 0.2574 \\
\hline 175 & 1 & 2057 & 0.41 & 0.43 & 26.39 & 50.69 & 77.08 & 4.114 & 32.48 & 113.67 & 2.00 & 11.58 & 14.01 & 0.2287 \\
\hline 200 & 1 & 3281 & 0.66 & 0.69 & 26.39 & 50.69 & 77.08 & 6.562 & 33.46 & 117.10 & 2.00 & 11.93 & 14.62 & 0.2089 \\
\hline 250 & 1 & 6312 & 1.26 & 1.32 & 26.39 & 50.69 & 77.08 & 12.624 & 35.88 & 125.58 & 2.00 & 12.79 & 16.11 & 0.1842 \\
\hline 300 & 2 & 10519 & 2.1 & 2.21 & 26.39 & 50.69 & 77.08 & 21.038 & 39.25 & 137.36 & 3.00 & 13.99 & 19.20 & 0.1828 \\
\hline 400 & 2 & 31401 & 6.28 & 6.59 & 26.39 & 50.69 & 77.08 & 62.802 & 55.95 & 195.83 & 3.00 & 19.95 & 29.54 & 0.2110 \\
\hline 500 & 2 & 73207 & 14.64 & 15.37 & 26.39 & 50.69 & 77.08 & 146.414 & 89.40 & 312.89 & 3.00 & 31.87 & 50.24 & 0.2871 \\
\hline
\end{tabular}

\section*{Notes:}

Pipe material cost \(=\$ 800 /\) ton
Pipe labor cost \(=\$ 80 / \mathrm{ft}\) for NPS 20
Compressor station cost \(=\$ 2000\) per installed HP
Miscellaneous cost is \(40 \%\) of pipe and compressor station cost
Operating cost based on 350 days per year
Fuel cost is \(\$ 3\) per MCF
Capital cost is annualized at \(8 \%\) interest for 20 -year project life
Table values in millions of dollars

The labor cost for installing pipe is calculated as follows:

For NPS 16 pipe,
Pipe installation cost \(=\$ 50 \times 5280 \times 120=\$ 31.68\) million
Similarly, for NPS 18 pipe,
Pipe installation cost \(=\$ 60 \times 5280 \times 120=\$ 38.02\) million
and for NPS 20 pipe,
Pipe installation cost \(=\$ 80 \times 5280 \times 120=\$ 50.69\) million
Next, we calculate the installation cost of compressor stations using \(\$ 2000\) per installed HP.

For the NPS 16 pipe at 100 MMSCFD flow rate, the HP required is 1072 and the installation cost is
\[
\$ 2000 \times 1072=\$ 2.14 \text { million }
\]

Similarly, the installation costs of each compressor station for all cases are calculated and tabulated as shown in Table 10.5 through Table 10.7.

The miscellaneous cost is \(40 \%\) of the sum of the pipe cost and compressor station cost, as follows:

Pipe material cost \(=\$ 10.66\) million
Pipe installation cost \(=\$ 31.68\) million
Compressor station cost \(=\$ 2.14\) million

Thus, for NPS 16 pipe at 100 MMSCFD,
Miscellaneous cost \(=0.40 \times(10.66+31.68+2.14)=\$ 17.79\) million
The operation and maintenance costs are added to the annual fuel cost to obtain the total annual cost. The total capital cost is annualized at \(8 \%\) interest for 20 years and added to the O\&M and fuel costs. For example, for NPS 16 pipe at 100 MMSCFD flow rate, the total capital cost of \(\$ 62.27\) million is annualized at \(\$ 6.34\) million and added to the O\&M and fuel costs to obtain the total annual cost of \(\$ 8.56\) million. Dividing this annual cost by the gas transported per year, we obtain the annual cost per MCF as follows:
\[
\text { Annual cost per MCF }=\frac{8.56 \times 10^{6} \times 10^{3}}{100 \times 10^{6} \times 350}=\$ 0.2447
\]

Similarly, for each pipe size and flow rate, the values are tabulated as shown in Table 10.5 through Table 10.7.

Upon reviewing Table 10.5 for NPS 16 pipe, we see that the annual cost per MCF decreases from \(\$ 0.4607\) to \(\$ 0.1897\) as the flow rate increases from 50 to 150 MMSCFD .

After that, it increases with flow rate and reaches a value of \(\$ 0.1942\) at 200 MMSCFD. Therefore, for NPS 16 pipe, 150 MMSCFD is the optimum flow rate that results in the lowest transportation cost.

Similarly, from Table 10.6, for NPS 18 pipe, the annual cost per MCF decreases from \(\$ 0.5233\) to \(\$ 0.1657\) as the flow rate increases from 50 to 200 MMSCFD. After that, it increases with flow rate and reaches a value of \(\$ 0.1809\) at 300 MMSCFD. Therefore, for NPS 18 pipe, 200 MMSCFD is the optimum flow rate that results in the lowest transportation cost.

Finally, from Table 10.7, for NPS 20 pipe, the annual cost per MCF decreases from \(\$ 0.3726\) to \(\$ 0.1828\) as the flow rate increases from 100 to 300 MMSCFD. After that, it increases with flow rate and reaches a value of \(\$ 0.2871\) at 500 MMSCFD . Therefore, for NPS 20 pipe, 300 MMSCFD is the optimum flow rate that results in the lowest transportation cost.

A plot of the annual cost per MCF vs. flow rate for the three pipe sizes is shown in Figure 10.2.

In the preceding calculations, to simplify matters, several assumptions were made. Miscellaneous costs were estimated as a percentage of the pipeline and compressor station costs. Also, we considered the annual costs to be constant from year to year. A more nearly accurate calculation would be to escalate the annual costs by a percentage each year to account for inflation, using the Consumer Price Index (CPI). Nevertheless, the preceding calculations illustrate a methodology of economic analysis to determine the most optimum pipe size.

\section*{Example 5}

In Chapter 5, we compared expanding the capacity of the gas pipeline from Windsor to Cardiff using two options-installing intermediate compressor stations or installing pipe loops. Using the results of Example 1 in Chapter 5, compare the two options,


Figure 10.2 Annualized cost vs. flow rate for three pipe sizes.
taking into account the capital cost, operating cost, and fuel cost and considering a project life of 25 years. The capital will be financed with \(70 \%\) debt at \(8 \%\) interest. The regulatory return allowed on the \(30 \%\) equity is \(12 \%\). The tax rate can be assumed to be \(35 \%\). Fuel consumption is 0.2 MCF per day per HP and fuel gas cost is \(\$ 3\) per MCF. Assume 350 days of operation per year. Calculate the annualized cost of service and transportation tariff for both options. It is expected that the annual O\&M cost will increase by \(\$ 2\) million for phase 1 and an additional \(\$ 3\) million for phase 2 compressor station options. For the looping option, the incremental O\&M cost is \(\$ 0.5\) million for phase 1 and \(\$ 0.75\) million for phase 2.

\section*{Solution}

Phase 1 expansion

This expansion results in a flow rate of 238.41 MMSCFD, and the compressor station option requires installing the following HP:

Windsor compressor station-8468 HP

Avon compressor station-3659 HP
\[
\text { Total } H P=8468+3659=12,127 \mathrm{HP}
\]

The incremental HP for phase 1 was calculated as
\[
\Delta H P=12,127-7064=5063 \mathrm{HP}
\]

The cost of this incremental HP based on \(\$ 2000\) per installed HP is
\[
\Delta \text { Capital cost }=5063 \times 2000=\$ 10.13 \text { million }
\]

The incremental fuel cost for 5063 HP is
\[
\Delta \text { Fuel cost }=5063 \times 0.2 \times \$ 3 \times 350=\$ 1.06 \text { million per year } .
\]

The incremental capital cost of \(\$ 10.13\) million will be funded by \(70 \%\) debt and \(30 \%\) equity. The debt capital \(=10.13 \times 0.7=\$ 7.09\) million.
\[
\text { Loan amortization cost }=\frac{10.13 \times 0.7 \times 0.08}{1-\left(\frac{1}{1.08}\right)^{25}}=\$ 0.66 \text { million per year }
\]

The remaining capital of ( \(\$ 10.13-\$ 7.09\) ) million or \(\$ 3.04\) million is equity that, according to regulatory guidelines, can earn \(12 \%\) interest.

The annual revenue allowed on equity capital is
\[
3.04 \times 0.12=\$ 0.36 \text { million }
\]

Accounting for a \(35 \%\) tax rate, the adjusted annual revenue on equity capital is
\[
\frac{\$ 0.36}{1-0.35}=\$ 0.55 \text { million }
\]

Next, add the O\&M cost increase of \(\$ 2\) million per year and the fuel cost of \(\$ 1.06\) million to the annual costs for debt and equity just calculated to arrive at the annual cost of service as follows:

Annual payment to retire debt \(=\$ 0.66\) million
Annual revenue on equity capital \(=\$ 0.36\) million
Annual operating cost \(=\$ 2.0\) million
Annual fuel cost \(=\$ 1.06\) million

Therefore, the incremental annual cost of service for phase 1 expansion compressor station option is \(\$ 0.66+0.36+2.0+1.06=\$ 4.08\) million.

This amount is the incremental annual cost of service over and above the cost of service for the initial flow rate of 188.41 MMSCFD.

The incremental tariff for an incremental flow rate of 50 MMSCFD for phase 1 expansion is
\[
\text { Incremental tariff }=\frac{4.08 \times 10^{6} \times 10^{3}}{50 \times 10^{6} \times 350}=\$ 0.2331 \text { per MCF }
\]

Next, we calculate the cost of service and tariff considering the looping option.
In Example 1 of Chapter 5, for phase 1 expansion, we required installation of 50.03 mi of loop at a cost of \(\$ 25.02\) million. In addition to this cost of pipe loop, we must include the cost of the increased horsepower requirement at Windsor for the phase 1 flow rate, which was calculated at 1404 HP .

At \(\$ 2000\) per installed HP, the extra cost for incremental HP is \(\$ 2.81\) million. Thus, for phase 1 the total cost of looping pipe upstream of Cardiff and increased HP cost at the Windsor compressor station is calculated as
\[
\$ 25.02+\$ 2.81=\$ 27.83 \text { million }
\]

The incremental fuel cost for the extra 1404 HP is
\[
\Delta \text { Fuel cost }=1404 \times 0.2 \times \$ 3 \times 350=\$ 0.30 \text { million per year }
\]

The incremental capital of \(\$ 27.83\) million for the looping option would also be funded by \(70 \%\) debt and \(30 \%\) equity.
\[
\begin{gathered}
\text { Debt capital }=27.83 \times 0.7=\$ 19.48 \text { million } \\
\text { Loan amortization cost }=\frac{19.48 \times 0.08}{1-\left(\frac{1}{1.08}\right)^{25}}=\$ 1.82 \text { million per year }
\end{gathered}
\]

The remaining capital of ( \(\$ 27.83-\$ 19.48\) ) million or \(\$ 8.35\) million is equity that, according to regulatory guidelines, can earn \(12 \%\) interest.

The annual revenue allowed on equity capital is
\[
8.35 \times 0.12=\$ 1.0 \text { million }
\]

Accounting for a \(35 \%\) tax rate, the adjusted annual revenue on equity capital is
\[
\frac{\$ 1.0}{1-0.35}=\$ 1.54 \text { million }
\]

Next, add the O\&M cost increase of \(\$ 0.5\) million per year and the fuel cost of \(\$ 0.30\) million to the annual costs for debt and equity just calculated to arrive at the incremental annual cost of service as follows:

Annual payment to retire debt \(=\$ 1.82\) million
Annual revenue on equity capital \(=\$ 1.0\) million
Annual operating cost \(=\$ 0.5\) million
Annual fuel cost \(=\$ 0.3\) million

Therefore, the incremental annual cost of service for phase 1 expansion looping option is \(\$ 1.82+1.0+0.5+0.3=\$ 3.62\) million .

This amount is the incremental annual cost of service over and above the cost of service for the initial flow rate of 188.41 MMSCFD.

The incremental tariff for an incremental flow rate of 50 MMSCFD for the phase 1 expansion looping option is
\[
\text { Incremental tariff }=\frac{3.62 \times 10^{6} \times 10^{3}}{50 \times 10^{6} \times 350}=\$ 0.2069 \text { per MCF }
\]

We can summarize the calculations as follows:

For the phase 1 expansion, compressor station option:

Incremental annual cost of service \(=\$ 4.08\) million
Incremental tariff \(=\$ 0.2331\) per MCF

For the phase 1 expansion, looping option:

Incremental annual cost of service \(=\$ 3.62\) million
Incremental tariff \(=\$ 0.2069\) per MCF

It can be seen that the incremental annual cost of service and the incremental tariff for phase 1 expansion are less in the looping option than the compressor station option. Therefore, for phase 1 expansion, the looping option is the preferred choice.

For phase 2 expansion, the throughput increase of 50 MMSCFD will be on top of phase 1 expansion. Since the preferred choice for phase 1 expansion is the
looping option, we must consider the increase in facilities required for phase 2 with 50.03 mi of pipe loop already installed. In Example 1 of Chapter 5, for phase 2, the loop required was calculated to be 76.26 mi . The incremental HP at Windsor was calculated as 1775 HP . Also, the incremental looping required and cost of increased HP at Windsor over the phase 1 values were calculated to be \(\$ 16.66\) million.

The incremental fuel cost for the extra 1775 HP is
\[
\Delta \text { Fuel cost }=1775 \times 0.2 \times \$ 3 \times 350=\$ 0.37 \text { million per year }
\]

The incremental capital of \(\$ 16.66\) million for the phase 2 looping option would also be funded by \(70 \%\) debt and \(30 \%\) equity.
\[
\begin{gathered}
\text { Debt capital }=16.66 \times 0.7=\$ 11.66 \text { million } \\
\text { Loan amortization cost }=\frac{11.66 \times 0.08}{1-\left(\frac{1}{1.08}\right)^{25}}=\$ 1.09 \text { million per year }
\end{gathered}
\]

The remaining capital of ( \(\$ 16.66-\$ 11.66\) ) million or \(\$ 5.0\) million is equity that, according to regulatory guidelines, can earn \(12 \%\) interest.

The annual revenue allowed on equity capital is
\[
\$ 5.0 \times 0.12=\$ 0.6 \text { million }
\]

Accounting for a \(35 \%\) tax rate, the adjusted annual revenue on equity capital is
\[
\frac{\$ 0.6}{1-0.35}=\$ 0.92 \text { million }
\]

Next, add the O\&M cost increase of \(\$ 0.75\) million per year and the fuel cost of \(\$ 0.37\) million to the annual costs for debt and equity just calculated to arrive at the incremental annual cost of service for the phase 2 looping expansion as follows:

Annual payment to retire debt \(=\$ 1.09\) million
Annual revenue on equity capital \(=\$ 0.6\) million
Annual operating cost \(=\$ 0.75\) million
Annual fuel cost \(=\$ 0.37\) million
Therefore, the incremental annual cost of service for phase 2 expansion looping option is \(\$ 1.09+0.6+0.75+0.37=\$ 2.81\) million.

This amount is the incremental annual cost of service over and above the cost of service for the phase 1 flow rate of 238.41 MMSCFD.

The incremental tariff for an incremental flow rate of 50 MMSCFD for the phase 2 expansion looping option is
\[
\text { Incremental tariff }=\frac{2.81 \times 10^{6} \times 10^{3}}{50 \times 10^{6} \times 350}=\$ 0.1606 \text { per MCF }
\]

In summary,

For the phase 2 expansion, looping option:
Incremental annual cost of service \(=\$ 2.81\) million
Incremental tariff \(=\$ 0.1606\) per MCF
These incremental costs are over and above the phase 1 numbers.

It must be noted that we did not consider a compressor station option for phase 2 expansion. This is because the preferred option for phase 1 expansion was installing loop. Since approximately 50 mi of pipe loop was already installed for phase 1 , we simply looked at adding approximately 26 mi of extra loop for phase 2. For comparison, we could determine additional compressor station requirements for phase 2 instead of extending the loop. This is left as an exercise for the reader.

\subsection*{10.5 SUMMARY}

In this chapter the economic aspects of a natural gas pipeline transportation were reviewed. A method for determining the optimum pipe size necessary to transport a certain flow rate was discussed. We introduced concepts of the capital cost of pipeline and compressor stations and the annual operating and maintenance costs. The fuel consumption calculations were also explained. Taking into account time value of money and the rate of return allowed on an equity investment in pipeline facilities, we calculated an annual cost of transporting gas. From this annual cost, the transportation tariff was calculated. The economic pipe size for a particular application was illustrated using three different pipe sizes and estimating the initial capital cost and annual operating costs. A typical pipeline expansion scenario with the option of installing compressors vs. pipe loops was also explained using economic principles. Additionally, the major components of the capital cost of a typical pipeline system were reviewed.

\section*{PROBLEMS}
1. A natural gas pipeline transports 120 MMSCFD at a load factor of \(95 \%\). The capital cost is estimated at \(\$ 70\) million and the annual operating cost is \(\$ 6\) million. Amortizing the capital at \(8 \%\) for a project life of 20 years, calculate the cost of the service and transportation tariffs for this pipeline.
2. A new pipeline is being constructed to transport natural gas from a processing plant to a power plant 150 mi away. An initial phase and an expansion phase
are contemplated. During the initial phase lasting 10 years, the amount of gas shipped is expected to be a constant volume of 100 MMSCFD at a \(95 \%\) load factor. A pipe size of NPS 18, 0.250 in . wall thickness, is required to handle the volumes with two compressor stations of 5000 HP total. The total pipeline cost can be estimated at \(\$ 750,000\) per mi and the compressor station cost at \(\$ 2000\) per HP installed. The annual operating costs are estimated at \(\$ 6\) million. The construction project will be financed by borrowing \(75 \%\) of the required capital at an interest rate of \(6 \%\). The regulatory rate of return allowed on equity is \(13 \%\). Consider a project life of 25 years and an overall tax rate of \(36 \%\).
a. Calculate the annual cost of service for this pipeline and the transportation tariff in \$/MCF.
b. The second phase, lasting the next 10 years, is projected to increase throughput to 150 MMSCFD. Calculate the transportation tariff for the expansion phase, considering the capital cost to increase by \(\$ 30\) million and the annual cost to increase by \(\$ 4\) million, with the same load factor as before.
3. A gas pipeline is to be constructed to transport 200 MMSCFD of natural gas from Jackson to Columbus, 180 mi away. Consider three pipe sizes-NPS 18, NPS 20, and NPS 24-all constructed of API 5L-X52 pipe with suitable wall thickness for an MOP of 1400 psig. Determine the most economical pipe diameter, taking into account the pipe material cost, cost of compressor stations, and fuel costs. The selection of pipe size can be based on a 30 -year project life and a present value of discounted cash flow at \(6 \%\) per year. Use \(\$ 750\) per ton for pipe material and \(\$ 2000\) per installed HP for compressor station cost. Fuel gas can be estimated at \(\$ 3\) per MCF.

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2. Mohitpour, M., Golshan, H., and Murray, A., Pipeline Design and Construction, 2nd ed., ASME Press, New York, 2003.
3. Pipeline Design for Hydrocarbon Gases and Liquids, American Society of Civil Engineers, New York, 1975.
4. Katz, D.L. et al., Handbook of Natural Gas Engineering, McGraw-Hill, New York, 1959.

\section*{appendix A}

\section*{Units and Conversions}
USCS Units-U.S. Customary System of Units
SI Units-Systeme International Units (modified metric)
\begin{tabular}{|c|c|c|c|c|}
\hline Item & USCS Units & SI Units & USCS to SI Conversion & SI to USCS Conversion \\
\hline \multirow[t]{4}{*}{Mass} & slug (slug) & \multirow[t]{2}{*}{kilogram (kg)} & \(1 \mathrm{lb}=0.45359 \mathrm{~kg}\) & \(1 \mathrm{~kg}=0.0685\) slug \\
\hline & pound mass (lbm) & & \(1 \mathrm{slug}=14.594 \mathrm{~kg}\) & \(1 \mathrm{~kg}=2.205 \mathrm{lb}\) \\
\hline & 1 U.S. ton \(=2,000 \mathrm{lb}\) & \multirow[t]{2}{*}{metric tonne \((\mathrm{t})=1,000 \mathrm{~kg}\)} & \(1 \mathrm{U} . \mathrm{S}\). ton \(=0.9072 \mathrm{t}\) & \(1 \mathrm{t}=1.1023\) U.S. ton \\
\hline & 1 long ton \(=2,240 \mathrm{lb}\) & & 1 long ton \(=1.016 \mathrm{t}\) & \(1 \mathrm{t}=0.9842\) long ton \\
\hline Weight & pound (lb) & Newton (N) \(=\mathrm{kg}-\mathrm{m} / \mathrm{s}^{2}\) & \(1 \mathrm{lb}=4.4482 \mathrm{~N}\) & \(1 \mathrm{~N}=0.2248 \mathrm{lb}\) \\
\hline \multirow[t]{3}{*}{Length} & inch (in.) & millimeter (mm) & \(1 \mathrm{in}=25.4 \mathrm{~mm}\) & \(1 \mathrm{~mm}=0.0394 \mathrm{in}\) \\
\hline & 1 foot (ft) = 12 in . & 1 meter \((\mathrm{m})=1,000 \mathrm{~mm}\) & \(1 \mathrm{ft}=0.3048 \mathrm{~m}\) & \(1 \mathrm{~m}=3.2808 \mathrm{ft}\) \\
\hline & 1 mile ( mi ) \(=5,280 \mathrm{ft}\) & 1 kilometer \((\mathrm{km})=1,000 \mathrm{~m}\) & \(1 \mathrm{mi}=1.609 \mathrm{~km}\) & \(1 \mathrm{~km}=0.6214 \mathrm{mi}\) \\
\hline \multirow[t]{2}{*}{Area} & square foot ( \(\mathrm{ft}^{2}\) ) & square meter ( \(\mathrm{m}^{2}\) ) & \(1 \mathrm{ft}^{2}=0.0929 \mathrm{~m}^{2}\) & \(1 \mathrm{~m}^{2}=10.764 \mathrm{ft}^{2}\) \\
\hline & 1 acre \(=43,560 \mathrm{ft}^{2}\) & 1 hectare \(=10,000 \mathrm{~m}^{2}\) & 1 acre \(=0.4047\) hectare & 1 hectare \(=2.4711\) acre \\
\hline Volume & cubic foot ( \(\mathrm{ft}^{3}\) ) & cubic meter ( \(\mathrm{m}^{3}\) ) & \(1 \mathrm{ft}^{3}=0.02832 \mathrm{~m}^{3}\) & \(1 \mathrm{~m}^{3}=35.3134 \mathrm{ft}^{3}\) \\
\hline Density & slug per cubic foot (slug/ft \({ }^{3}\) ) & kilogram/cubic meter ( \(\mathrm{kg} / \mathrm{m}^{3}\) ) & 1 slug/ft \({ }^{3}=515.38 \mathrm{~kg} / \mathrm{m}^{3}\) & \(1 \mathrm{~kg} / \mathrm{m}^{3}=0.0019\) slug/ \(/ \mathrm{ft}^{3}\) \\
\hline Specific weight & pound per cubic foot ( \(\mathrm{lb} / \mathrm{ft}^{3}\) ) & Newton per cubic meter ( \(\mathrm{N} / \mathrm{m}^{3}\) ) & \(1 \mathrm{lb} / \mathrm{ft}^{3}=157.09 \mathrm{~N} / \mathrm{m}^{3}\) & \(1 \mathrm{~N} / \mathrm{m}^{3}=0.0064 \mathrm{lb} / \mathrm{ft}^{3}\) \\
\hline \multirow[t]{3}{*}{Viscosity (absolute or dynamic)} & \(\mathrm{lb} / \mathrm{ft}-\mathrm{s}\) & 1 Poise ( \(P\) ) = 0.1 Pa-s & & \(1 \mathrm{cP}=6.7197 \times 10^{-4} \mathrm{lb} / \mathrm{ft}-\mathrm{s}\) \\
\hline & \multirow[t]{2}{*}{\(\mathrm{lb}-\mathrm{s} / \mathrm{ft}^{2}\)} & 1 centiPoise (cP) \(=0.01 \mathrm{P}\) & \(1 \mathrm{lb}-\mathrm{s} / \mathrm{ft}^{2}=47.88 \mathrm{~N}-\mathrm{s} / \mathrm{m}^{2}\) & \(1 \mathrm{~N}-\mathrm{s} / \mathrm{m}^{2}=0.0209 \mathrm{lb}-\mathrm{s} / \mathrm{ft}^{2}\) \\
\hline & & \begin{tabular}{l}
1 Poise = 1 dyne-s \(/ \mathrm{cm}^{2}\) \\
1 Poise \(=0.1 \mathrm{~N}-\mathrm{s} / \mathrm{m}^{2}\)
\end{tabular} & \(1 \mathrm{lb}-\mathrm{s} / \mathrm{ft}^{2}=478.8\) Poise & 1 Poise \(=0.00209 \mathrm{lb}-\mathrm{s} / \mathrm{ft}^{2}\) \\
\hline \multirow[t]{2}{*}{Viscosity (kinematic)} & \multirow[t]{2}{*}{\(\mathrm{ft}^{2} / \mathrm{s}\)} & \(\mathrm{m}^{2} / \mathrm{s}\) & \multirow[t]{2}{*}{\(1 \mathrm{ft}^{2} / \mathrm{s}=0.092903 \mathrm{~m}^{2} / \mathrm{s}\)} & \(1 \mathrm{~m}^{2} / \mathrm{s}=10.7639 \mathrm{ft}^{2} / \mathrm{s}\) \\
\hline & & Stoke (S), centiStoke (cSt) & & \(1 \mathrm{cSt}=1.076 \times 10^{-5} \mathrm{ft}^{2} / \mathrm{s}\) \\
\hline Flow rate & cubic foot/hour (ft \({ }^{3} / \mathrm{h}\) ) & cubic meter/hour ( \(\mathrm{m}^{3} / \mathrm{h}\) ) & \(1 \mathrm{ft}^{3} / \mathrm{h}=0.02832 \mathrm{~m}^{3} / \mathrm{h}\) & \(1 \mathrm{~m}^{3} / \mathrm{h}=35.3134 \mathrm{ft}^{3} / \mathrm{h}\) \\
\hline & cubic foot/day ( \(\mathrm{ft}^{3} / \mathrm{day}\) ) & cubic meter/day ( \(\mathrm{m}^{3} / \mathrm{day}\) ) & & \\
\hline & million Std \(\mathrm{ft}^{3} /\) day (MMSCFD) & million std \(\mathrm{m}^{3} /\) day ( \(\mathrm{Mm}^{3} /\) day \()\) & & \\
\hline Force & pound (lb) & Newton (N) \(=\mathrm{kg}-\mathrm{m} / \mathrm{s}^{2}\) & \(1 \mathrm{lb}=4.4482 \mathrm{~N}\) & \(1 \mathrm{~N}=0.2248 \mathrm{lb}\) \\
\hline
\end{tabular}
\begin{tabular}{|c|c|c|c|c|}
\hline Pressure & pound/square inch, lb/in \({ }^{2}\) (psi) \(1 \mathrm{~b} / \mathrm{ft}^{2}=144 \mathrm{psi}\) & \[
\begin{aligned}
& \text { Pascal }(\mathrm{Pa})=\mathrm{N} / \mathrm{m}^{2} \\
& 1 \mathrm{kiloPascal}(\mathrm{kPa})=1,000 \mathrm{~Pa} \\
& 1 \mathrm{megaPascal}(\mathrm{MPa})=1,000 \mathrm{kPa} \\
& 1 \mathrm{Bar}=100 \mathrm{kPa} \\
& \text { kilogram } / \mathrm{sq} \text {. centimeter }\left(\mathrm{kg} / \mathrm{cm}^{2}\right)
\end{aligned}
\] & \[
\begin{aligned}
& 1 \mathrm{psi}=6.895 \mathrm{kPa} \\
& 1 \mathrm{psi}=0.069 \mathrm{Bar} \\
& 1 \mathrm{psi}=0.0703 \mathrm{~kg} / \mathrm{cm}^{2}
\end{aligned}
\] & \[
\begin{aligned}
& 1 \mathrm{kPa}=0.145 \mathrm{psi} \\
& \\
& 1 \mathrm{Bar}=14.5 \mathrm{psi} \\
& 1 \mathrm{~kg} / \mathrm{cm}^{2}=14.22 \mathrm{psi}
\end{aligned}
\] \\
\hline Velocity & \begin{tabular}{l}
foot/second (ft/s) \\
mile \(/\) hour ( \(\mathrm{m} / \mathrm{h}\) ) \(=1.4667 \mathrm{ft} / \mathrm{s}\)
\end{tabular} & meter/second (m/s) & \(1 \mathrm{ft} / \mathrm{s}=0.3048 \mathrm{~m} / \mathrm{s}\) & \(1 \mathrm{~m} / \mathrm{s}=3.281 \mathrm{ft} / \mathrm{s}\) \\
\hline Work and energy & \begin{tabular}{l}
foot-pound (ft-lb) \\
British thermal unit (Btu) \\
\(1 \mathrm{Btu}=778 \mathrm{ft}-\mathrm{lb}\)
\end{tabular} & Joule ( J ) \(=\mathrm{N}-\mathrm{m}\) & \(1 \mathrm{Btu}=1055.0 \mathrm{~J}\) & \(1 \mathrm{~kJ}=0.9478 \mathrm{Btu}\) \\
\hline Power & \begin{tabular}{l}
\(\mathrm{ft}-\mathrm{lb} / \mathrm{min}\) \\
Btu/hour \\
horsepower (HP) \\
\(1 \mathrm{HP}=33,000 \mathrm{ft}-\mathrm{lb} / \mathrm{min}\)
\end{tabular} & \[
\begin{aligned}
& \text { Joule/second }(\mathrm{J} / \mathrm{s}) \\
& \text { watt }(\mathrm{W})=\mathrm{J} / \mathrm{s} \\
& 1 \text { kilowatt }(\mathrm{kW})=1,000 \mathrm{~W}
\end{aligned}
\] & \[
\begin{aligned}
& 1 \mathrm{Btu} / \mathrm{hr}=0.2931 \mathrm{~W} \\
& 1 \mathrm{HP}=0.746 \mathrm{~kW}
\end{aligned}
\] & \[
\begin{aligned}
& 1 \mathrm{~W}=3.4121 \mathrm{Btu} / \mathrm{hr} \\
& 1 \mathrm{~kW}=1.3405 \mathrm{HP}
\end{aligned}
\] \\
\hline Temperature & \[
\begin{aligned}
& \text { degree Fahrenheit }\left({ }^{\circ} \mathrm{F}\right) \\
& 1 \text { degree Rankin }\left({ }^{\circ} \mathrm{R}\right)={ }^{\circ} \mathrm{F}+460
\end{aligned}
\] & \begin{tabular}{l}
degree Celsius ( \({ }^{\circ} \mathrm{C}\) ) \\
1 degree Kelvin \((K)={ }^{\circ} \mathrm{C}+273\)
\end{tabular} & \[
\begin{aligned}
& 1^{\circ} \mathrm{F}=9 / 5^{\circ} \mathrm{C}+32 \\
& 1^{\circ} \mathrm{R}=1.8 \mathrm{~K}
\end{aligned}
\] & \[
\begin{aligned}
& 1^{\circ} \mathrm{C}=\left({ }^{\circ} \mathrm{F}-32\right) / 1.8 \\
& 1 \mathrm{~K}={ }^{\circ} \mathrm{R} / 1.8
\end{aligned}
\] \\
\hline Thermal conductivity & \(\mathrm{Btu} / \mathrm{hr} / \mathrm{ft} /{ }^{\circ} \mathrm{F}\) & \(\mathrm{W} / \mathrm{m} /{ }^{\circ} \mathrm{C}\) & \[
\begin{aligned}
& 1 \mathrm{Btu} / \mathrm{hr} / \mathrm{ft} /{ }^{\circ} \mathrm{F}= \\
& 1.7307 \mathrm{~W} / \mathrm{m} /{ }^{\circ} \mathrm{C}
\end{aligned}
\] & \(1 \mathrm{~W} / \mathrm{m} /{ }^{\circ} \mathrm{C}=\) 0.5778 Btu/hr/ft/F \\
\hline Heat transfer coefficient & Btu/hr/ft \(/\) / \({ }^{\circ} \mathrm{F}\) & \(\mathrm{W} / \mathrm{m}^{2} /{ }^{\circ} \mathrm{C}\) & \[
\begin{aligned}
& 1 \mathrm{Btu} / \mathrm{hr} / \mathrm{ft}^{2} /{ }^{\circ} \mathrm{F}= \\
& 5.6781 \mathrm{~W} / \mathrm{m}^{2} /{ }^{\circ} \mathrm{C}
\end{aligned}
\] & \(1 \mathrm{~W} / \mathrm{m}^{2} /{ }^{\circ} \mathrm{C}=\) 0.1761 Btu/hr//t \(\mathrm{t}^{\circ} \mathrm{F}\) \\
\hline Specific heat & \(\mathrm{Btu} / \mathrm{lb} /{ }^{\circ} \mathrm{F}\) & kJ/kg/ \({ }^{\circ} \mathrm{C}\) & \(1 \mathrm{Btu} / \mathrm{b} /{ }^{\circ} \mathrm{F}=\) \(4.1869 \mathrm{~kJ} / \mathrm{kg} /{ }^{\circ} \mathrm{C}\) & \(1 \mathrm{~kJ} / \mathrm{kg} /{ }^{\circ} \mathrm{C}=\) \(0.2388 \mathrm{Btu} / \mathrm{b} / \mathrm{F}\) \\
\hline
\end{tabular}

\section*{Physical Properties of Various Gases}
\begin{tabular}{|c|c|c|c|c|c|c|c|c|c|}
\hline \multirow[t]{2}{*}{Gas} & \multirow[t]{2}{*}{Formula} & \multirow[t]{2}{*}{Molecular Weight} & \multirow[t]{2}{*}{Vapor Pressure, psia at \(100^{\circ} \mathrm{F}\)} & \multicolumn{3}{|l|}{Critical Constants} & \multicolumn{2}{|l|}{Ideal Gas, 14.696 psia, \(60^{\circ} \mathrm{F}\)} & \multirow[t]{2}{*}{Specific Heat, Btu/b/ \({ }^{\circ} \mathrm{F}\) 14.696 psia, \(60^{\circ} F\) Ideal Gas} \\
\hline & & & & Pressure, psia & Temp., \({ }^{\circ} \mathrm{F}\) & Volume, \(\mathrm{ft}^{3} / \mathrm{lb}\) & \[
\begin{gathered}
\text { Spgr } \\
(\text { air }=1.00)
\end{gathered}
\] & \(\mathrm{ft}^{3} / \mathrm{lb}\)-gas & \\
\hline Methane & \(\mathrm{CH}_{4}\) & 16.0430 & 5,000 & 666.0 & -116.66 & 0.0988 & 0.5539 & 23.654 & 0.52676 \\
\hline Ethane & \(\mathrm{C}_{2} \mathrm{H}_{6}\) & 30.0700 & 800 & 707.0 & 90.07 & 0.0783 & 1.0382 & 12.620 & 0.40789 \\
\hline Propane & \(\mathrm{C}_{3} \mathrm{H}_{8}\) & 44.0970 & 188.65 & 617.0 & 205.93 & 0.0727 & 1.5226 & 8.6059 & 0.38847 \\
\hline Isobutane & \(\mathrm{C}_{4} \mathrm{H}_{10}\) & 58.1230 & 72.581 & 527.9 & 274.4 & 0.0714 & 2.0068 & 6.5291 & 0.38669 \\
\hline n -butane & \(\mathrm{C}_{4} \mathrm{H}_{10}\) & 58.1230 & 51.706 & 548.8 & 305.52 & 0.0703 & 2.0068 & 6.5291 & 0.39500 \\
\hline Iso-pentane & \(\mathrm{C}_{5} \mathrm{H}_{12}\) & 72.1500 & 20.443 & 490.4 & 368.96 & 0.0684 & 2.4912 & 5.2596 & 0.38448 \\
\hline n -pentane & \(\mathrm{C}_{5} \mathrm{H}_{12}\) & 72.1500 & 15.575 & 488.1 & 385.7 & 0.0695 & 2.4912 & 5.2596 & 0.38831 \\
\hline Neo-pentane & \(\mathrm{C}_{5} \mathrm{H}_{12}\) & 72.1500 & 36.72 & 464.0 & 321.01 & 0.0673 & 2.4912 & 5.2596 & 0.39038 \\
\hline n -hexane & \(\mathrm{C}_{6} \mathrm{H}_{14}\) & 86.1770 & 4.9596 & 436.9 & 453.8 & 0.0688 & 2.9755 & 4.4035 & 0.38631 \\
\hline 2-methyl pentane & \(\mathrm{C}_{6} \mathrm{H}_{14}\) & 86.1770 & 6.769 & 436.6 & 435.76 & 0.0682 & 2.9755 & 4.4035 & 0.38526 \\
\hline 3 -methyl pentane & \(\mathrm{C}_{6} \mathrm{H}_{14}\) & 86.1770 & 6.103 & 452.5 & 448.2 & 0.0682 & 2.9755 & 4.4035 & 0.37902 \\
\hline Neo hexane & \(\mathrm{C}_{6} \mathrm{H}_{14}\) & 86.1770 & 9.859 & 446.7 & 419.92 & 0.0667 & 2.9755 & 4.4035 & 0.38231 \\
\hline 2,3-dimethylbutane & \(\mathrm{C}_{6} \mathrm{H}_{14}\) & 86.1770 & 7.406 & 454.0 & 440.08 & 0.0665 & 2.9755 & 4.4035 & 0.37762 \\
\hline n -Heptane & \(\mathrm{C}_{7} \mathrm{H}_{16}\) & 100.2040 & 1.621 & 396.8 & 512.8 & 0.0682 & 3.4598 & 3.7872 & 0.38449 \\
\hline 2-Methylhexane & \(\mathrm{C}_{7} \mathrm{H}_{16}\) & 100.2040 & 2.273 & 396.0 & 494.44 & 0.0673 & 3.4598 & 3.7872 & 0.38170 \\
\hline 3-Methylhexane & \(\mathrm{C}_{7} \mathrm{H}_{16}\) & 100.2040 & 2.13 & 407.6 & 503.62 & 0.0646 & 3.4598 & 3.7872 & 0.37882 \\
\hline 3-Ethylpentane & \(\mathrm{C}_{7} \mathrm{H}_{16}\) & 100.2040 & 2.012 & 419.2 & 513.16 & 0.0665 & 3.4598 & 3.7872 & 0.38646 \\
\hline 2,2-Dimethylpentane & \(\mathrm{C}_{7} \mathrm{H}_{16}\) & 100.2040 & 3.494 & 401.8 & 476.98 & 0.0665 & 3.4598 & 3.7872 & 0.38651 \\
\hline 2,4-Dimethylpentane & \(\mathrm{C}_{7} \mathrm{H}_{16}\) & 100.2040 & 3.294 & 397.4 & 475.72 & 0.0667 & 3.4598 & 3.7872 & 0.39627 \\
\hline 3,3-Dimethylpentane & \(\mathrm{C}_{7} \mathrm{H}_{16}\) & 100.2040 & 2.775 & 427.9 & 505.6 & 0.0662 & 3.4598 & 3.7872 & 0.38306 \\
\hline Triptane & \(\mathrm{C}_{7} \mathrm{H}_{16}\) & 100.2040 & 3.376 & 427.9 & 496.24 & 0.0636 & 3.4598 & 3.7872 & 0.37724 \\
\hline n -octane & \(\mathrm{C}_{88} \mathrm{H}_{18}\) & 114.2310 & 0.5371 & 360.7 & 564.15 & 0.0673 & 3.9441 & 3.322 & 0.38334 \\
\hline Di Isobutyl & \(\mathrm{C}_{8} \mathrm{H}_{18}\) & 114.2310 & 1.1020 & 361.1 & 530.26 & 0.0676 & 3.9441 & 3.322 & 0.37571 \\
\hline Isooctane & \(\mathrm{C}_{8} \mathrm{H}_{18}\) & 114.2310 & 1.7090 & 372.7 & 519.28 & 0.0657 & 3.9441 & 3.322 & 0.38222 \\
\hline n -Nonane & \(\mathrm{C}_{9} \mathrm{H}_{20}\) & 128.2580 & 0.17155 & 330.7 & 610.72 & 0.0693 & 4.4284 & 2.9588 & 0.38248 \\
\hline n -Decane & \(\mathrm{C}_{10} \mathrm{H}_{22}\) & 142.2850 & 0.06088 & 304.6 & 652.1 & 0.0702 & 4.9127 & 2.6671 & 0.38181 \\
\hline Cyclopentane & \(\mathrm{C}_{5} \mathrm{H}_{10}\) & 70.1340 & 9.917 & 653.8 & 461.1 & 0.0594 & 2.4215 & 5.411 & 0.27122 \\
\hline Methylcyclopentane & \(\mathrm{C}_{6} \mathrm{H}_{12}\) & 84.1610 & 4.491 & 548.8 & 499.28 & 0.0607 & 2.9059 & 4.509 & 0.30027 \\
\hline Cyclohexane & \(\mathrm{C}_{6} \mathrm{H}_{12}\) & 84.1610 & 3.267 & 590.7 & 536.6 & 0.0586 & 2.9059 & 4.509 & 0.29012 \\
\hline Methylcyclohexane & \(\mathrm{C}_{7} \mathrm{H}_{14}\) & 98.1880 & 1.609 & 503.4 & 570.2 & 0.0600 & 3.3902 & 3.8649 & 0.31902 \\
\hline
\end{tabular}


\section*{APPENDIX C}

\section*{Pipe Properties-U.S. Customary System of Units}
\begin{tabular}{|c|c|c|c|c|c|c|c|c|c|c|c|}
\hline \multirow[t]{2}{*}{Nominal Pipe Size, NPS} & \multirow[t]{2}{*}{Outside Dia, in.} & \multicolumn{3}{|l|}{Schedule} & \multirow[t]{2}{*}{Wall Thickness, in} & \multirow[t]{2}{*}{\begin{tabular}{l}
Inside \\
Dia, in
\end{tabular}} & \multirow[t]{2}{*}{Inside Area, in. \({ }^{2}\)} & \multirow[t]{2}{*}{\begin{tabular}{l}
Surface \\
Area, \(\mathrm{ft}^{2} / \mathrm{ft}\)
\end{tabular}} & \multirow[t]{2}{*}{Volume, \(\mathrm{ft}^{3} / \mathrm{ft}\)} & \multirow[t]{2}{*}{Pipe Weight, lb/ft} & \multirow[t]{2}{*}{Water Weight, lb/ft} \\
\hline & & a & b & c & & & & & & & \\
\hline \multirow[t]{6}{*}{1/2} & 0.84 & & & 5 S & 0.065 & 0.710 & 0.3957 & 0.22 & 0.0027 & 0.54 & 0.02 \\
\hline & 0.84 & & & 10S & 0.083 & 0.674 & 0.3566 & 0.22 & 0.0025 & 0.67 & 0.02 \\
\hline & 0.84 & 40 & Std & 40S & 0.109 & 0.622 & 0.3037 & 0.22 & 0.0021 & 0.85 & 0.02 \\
\hline & 0.84 & 80 & XS & 805 & 0.147 & 0.546 & 0.2340 & 0.22 & 0.0016 & 1.09 & 0.01 \\
\hline & 0.84 & 160 & & & 0.187 & 0.466 & 0.1705 & 0.22 & 0.0012 & 1.30 & 0.01 \\
\hline & 0.84 & & XXS & & 0.294 & 0.252 & 0.0499 & 0.22 & 0.0003 & 1.71 & 0.00 \\
\hline \multirow[t]{6}{*}{3/4} & 1.05 & & & 5 S & 0.065 & 0.920 & 0.6644 & 0.27 & 0.0046 & 0.68 & 0.04 \\
\hline & 1.05 & & & 10S & 0.083 & 0.884 & 0.6134 & 0.27 & 0.0043 & 0.86 & 0.04 \\
\hline & 1.05 & 40 & Std & 40 S & 0.113 & 0.824 & 0.5330 & 0.27 & 0.0037 & 1.13 & 0.03 \\
\hline & 1.05 & 80 & XS & 805 & 0.154 & 0.742 & 0.4322 & 0.27 & 0.0030 & 1.47 & 0.03 \\
\hline & 1.05 & 160 & & & 0.218 & 0.614 & 0.2959 & 0.27 & 0.0021 & 1.94 & 0.02 \\
\hline & 1.05 & & xxs & & 0.308 & 0.434 & 0.1479 & 0.27 & 0.0010 & 2.44 & 0.01 \\
\hline \multirow[t]{6}{*}{1} & 1.315 & & & 5 S & 0.065 & 1.185 & 1.1023 & 0.34 & 0.0077 & 0.87 & 0.06 \\
\hline & 1.315 & & & 10 S & 0.109 & 1.097 & 0.9447 & 0.34 & 0.0066 & 1.40 & 0.05 \\
\hline & 1.315 & 40 & Std & 40S & 0.330 & 0.655 & 0.3368 & 0.34 & 0.0023 & 3.47 & 0.02 \\
\hline & 1.315 & 80 & XS & 805 & 0.179 & 0.957 & 0.7189 & 0.34 & 0.0050 & 2.17 & 0.04 \\
\hline & 1.315 & 160 & & & 0.250 & 0.815 & 0.5214 & 0.34 & 0.0036 & 2.84 & 0.03 \\
\hline & 1.315 & & XXS & & 0.358 & 0.599 & 0.2817 & 0.34 & 0.0020 & 3.66 & 0.02 \\
\hline \multirow[t]{6}{*}{\(11 / 2\)} & 1.900 & & & 5 S & 0.065 & 1.770 & 2.4593 & 0.50 & 0.0171 & 1.27 & 0.14 \\
\hline & 1.900 & & & 10 S & 0.109 & 1.682 & 2.2209 & 0.50 & 0.0154 & 2.08 & 0.13 \\
\hline & 1.900 & 40 & Std & 40 S & 0.145 & 1.610 & 2.0348 & 0.50 & 0.0141 & 2.72 & 0.12 \\
\hline & 1.900 & 80 & XS & 805 & 0.200 & 1.500 & 1.7663 & 0.50 & 0.0123 & 3.63 & 0.10 \\
\hline & 1.900 & 160 & & & 0.281 & 1.338 & 1.4053 & 0.50 & 0.0098 & 4.86 & 0.08 \\
\hline & 1.900 & & xXS & & 0.400 & 1.100 & 0.9499 & 0.50 & 0.0066 & 6.41 & 0.06 \\
\hline \multirow[t]{6}{*}{2} & 2.375 & & & 5 S & 0.065 & 2.245 & 3.9564 & 0.62 & 0.0275 & 1.60 & 0.23 \\
\hline & 2.375 & & & 10 S & 0.109 & 2.157 & 3.6523 & 0.62 & 0.0254 & 2.64 & 0.21 \\
\hline & 2.375 & 40 & Std & 40S & 0.154 & 2.067 & 3.3539 & 0.62 & 0.0233 & 3.65 & 0.19 \\
\hline & 2.375 & 80 & XS & 805 & 0.218 & 1.939 & 2.9514 & 0.62 & 0.0205 & 5.02 & 0.17 \\
\hline & 2.375 & 160 & & & 0.343 & 1.689 & 2.2394 & 0.62 & 0.0156 & 7.44 & 0.13 \\
\hline & 2.375 & & xXS & & 0.436 & 1.503 & 1.7733 & 0.62 & 0.0123 & 9.03 & 0.10 \\
\hline \multirow[t]{2}{*}{\(2^{1 / 2}\)} & 2.875 & & & 5 S & 0.083 & 2.709 & 5.7609 & 0.75 & 0.0400 & 2.47 & 0.33 \\
\hline & 2.875 & & & 10S & 0.12 & 2.635 & 5.4504 & 0.75 & 0.0379 & 3.53 & 0.32 \\
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\begin{tabular}{|c|c|c|c|c|c|c|c|c|c|c|c|}
\hline \multirow[t]{2}{*}{Nominal Pipe Size, NPS} & \multirow[t]{2}{*}{Outside Dia, in.} & \multicolumn{3}{|l|}{Schedule} & \multirow[t]{2}{*}{\begin{tabular}{l}
Wall \\
Thickness, in.
\end{tabular}} & \multirow[t]{2}{*}{Inside Dia, in.} & \multirow[t]{2}{*}{\begin{tabular}{l}
Inside \\
Area, in. \({ }^{2}\)
\end{tabular}} & \multirow[t]{2}{*}{Surface Area, \(\mathrm{ft}^{2} / \mathrm{ft}\)} & \multirow[t]{2}{*}{Volume, \(\mathrm{ft}^{3} / \mathrm{ft}\)} & \multirow[t]{2}{*}{Pipe Weight, lb/ft} & \multirow[t]{2}{*}{Water Weight, lb/ft} \\
\hline & & a & b & c & & & & & & & \\
\hline & 8.625 & & XXS & & 0.875 & 6.875 & 37.1035 & 2.26 & 0.2577 & 72.42 & 2.15 \\
\hline & 8.625 & 160 & & & 0.906 & 6.813 & 36.4373 & 2.26 & 0.2530 & 74.69 & 2.11 \\
\hline 10 & 10.75 & & & 5 S & 0.134 & 10.482 & 86.2498 & 2.81 & 0.5990 & 15.19 & 5.00 \\
\hline & 10.75 & & & 10 S & 0.165 & 10.420 & 85.2325 & 2.81 & 0.5919 & 18.65 & 4.94 \\
\hline & 10.75 & 20 & & & 0.250 & 10.250 & 82.4741 & 2.81 & 0.5727 & 28.04 & 4.78 \\
\hline & 10.75 & & & & 0.279 & 10.192 & 81.5433 & 2.81 & 0.5663 & 31.20 & 4.72 \\
\hline & 10.75 & 30 & & & 0.307 & 10.136 & 80.6497 & 2.81 & 0.5601 & 34.24 & 4.67 \\
\hline & 10.75 & 40 & Std & 40 S & 0.365 & 10.020 & 78.8143 & 2.81 & 0.5473 & 40.48 & 4.57 \\
\hline & 10.75 & 60 & XS & 80 S & 0.500 & 9.750 & 74.6241 & 2.81 & 0.5182 & 54.74 & 4.32 \\
\hline & 10.75 & 80 & & & 0.593 & 9.564 & 71.8040 & 2.81 & 0.4986 & 64.33 & 4.16 \\
\hline & 10.75 & 100 & & & 0.718 & 9.314 & 68.0992 & 2.81 & 0.4729 & 76.93 & 3.94 \\
\hline & 10.75 & 120 & & & 0.843 & 9.064 & 64.4925 & 2.81 & 0.4479 & 89.20 & 3.74 \\
\hline & 10.75 & 140 & & & 1.000 & 8.750 & 60.1016 & 2.81 & 0.4174 & 104.13 & 3.48 \\
\hline & 10.75 & 160 & & & 1.125 & 8.500 & 56.7163 & 2.81 & 0.3939 & 115.64 & 3.29 \\
\hline 12 & 12.75 & & & 5 S & 0.156 & 12.438 & 121.4425 & 3.34 & 0.8434 & 20.98 & 7.03 \\
\hline & 12.75 & & & 10 S & 0.180 & 12.390 & 120.5070 & 3.34 & 0.8369 & 24.16 & 6.98 \\
\hline & 12.75 & 20 & & & 0.250 & 12.250 & 117.7991 & 3.34 & 0.8181 & 33.38 & 6.82 \\
\hline & 12.75 & 30 & & & 0.330 & 12.090 & 114.7420 & 3.34 & 0.7968 & 43.77 & 6.65 \\
\hline & 12.75 & & Std & 40 S & 0.375 & 12.000 & 113.0400 & 3.34 & 0.7850 & 49.56 & 6.55 \\
\hline & 12.75 & 40 & & & 0.406 & 11.938 & 111.8749 & 3.34 & 0.7769 & 53.52 & 6.48 \\
\hline & 12.75 & & xs & 80S & 0.500 & 11.750 & 108.3791 & 3.34 & 0.7526 & 65.42 & 6.28 \\
\hline & 12.75 & 60 & & & 0.562 & 11.626 & 106.1036 & 3.34 & 0.7368 & 73.15 & 6.15 \\
\hline & 12.75 & 80 & & & 0.687 & 11.376 & 101.5895 & 3.34 & 0.7055 & 88.51 & 5.88 \\
\hline & 12.75 & 100 & & & 0.843 & 11.064 & 96.0935 & 3.34 & 0.6673 & 107.20 & 5.57 \\
\hline & 12.75 & 120 & & & 1.000 & 10.750 & 90.7166 & 3.34 & 0.6300 & 125.49 & 5.26 \\
\hline & 12.75 & 140 & & & 1.125 & 10.500 & 86.5463 & 3.34 & 0.6010 & 139.67 & 5.01 \\
\hline & 12.75 & 160 & & & 1.312 & 10.126 & 80.4907 & 3.34 & 0.5590 & 160.27 & 4.66 \\
\hline 14 & 14.00 & & & 5 S & 0.156 & 13.688 & 147.0787 & 3.67 & 1.0214 & 23.07 & 8.52 \\
\hline & 14.00 & & & 10 S & 0.188 & 13.624 & 145.7065 & 3.67 & 1.0119 & 27.73 & 8.44 \\
\hline & 14.00 & 10 & & & 0.250 & 13.500 & 143.0663 & 3.67 & 0.9935 & 36.71 & 8.29 \\
\hline & 14.00 & 20 & & & 0.312 & 13.376 & 140.4501 & 3.67 & 0.9754 & 45.61 & 8.14 \\
\hline & 14.00 & 30 & Std & & 0.375 & 13.250 & 137.8166 & 3.67 & 0.9571 & 54.57 & 7.98 \\
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\begin{tabular}{|c|c|c|c|c|c|c|c|c|c|c|c|}
\hline \multirow[t]{2}{*}{Nominal Pipe Size, NPS} & \multirow[t]{2}{*}{Outside Dia, in.} & \multicolumn{3}{|l|}{Schedule} & \multirow[t]{2}{*}{\begin{tabular}{l}
Wall \\
Thickness, in.
\end{tabular}} & \multirow[t]{2}{*}{Inside Dia, in.} & \multirow[t]{2}{*}{\begin{tabular}{l}
Inside \\
Area, in. \({ }^{2}\)
\end{tabular}} & \multirow[t]{2}{*}{Surface Area, \(\mathrm{ft}^{2} / \mathrm{ft}\)} & \multirow[t]{2}{*}{Volume, \(\mathrm{ft}^{3} / \mathrm{ft}\)} & \multirow[t]{2}{*}{\begin{tabular}{l}
Pipe \\
Weight, lb/ft
\end{tabular}} & \multirow[t]{2}{*}{Water Weight, lb/ft} \\
\hline & & a & b & c & & & & & & & \\
\hline & 18.00 & 20 & & & 0.312 & 17.376 & 237.0114 & 4.71 & 1.6459 & 58.94 & 13.73 \\
\hline & 18.00 & & Std & & 0.375 & 17.250 & 233.5866 & 4.71 & 1.6221 & 70.59 & 13.53 \\
\hline & 18.00 & 30 & & & 0.437 & 17.126 & 230.2404 & 4.71 & 1.5989 & 81.97 & 13.34 \\
\hline & 18.00 & & XS & & 0.500 & 17.000 & 226.8650 & 4.71 & 1.5755 & 93.45 & 13.14 \\
\hline & 18.00 & 40 & & & 0.562 & 16.876 & 223.5675 & 4.71 & 1.5526 & 104.67 & 12.95 \\
\hline & 18.00 & & & & 0.625 & 16.750 & 220.2416 & 4.71 & 1.5295 & 115.98 & 12.76 \\
\hline & 18.00 & & & & 0.687 & 16.626 & 216.9927 & 4.71 & 1.5069 & 127.03 & 12.57 \\
\hline & 18.00 & 60 & & & 0.750 & 16.500 & 213.7163 & 4.71 & 1.4841 & 138.17 & 12.38 \\
\hline & 18.00 & & & & 0.875 & 16.250 & 207.2891 & 4.71 & 1.4395 & 160.03 & 12.01 \\
\hline & 18.00 & 80 & & & 0.937 & 16.126 & 204.1376 & 4.71 & 1.4176 & 170.75 & 11.83 \\
\hline & 18.00 & 100 & & & 1.156 & 15.688 & 193.1990 & 4.71 & 1.3417 & 207.96 & 11.19 \\
\hline & 18.00 & 120 & & & 1.375 & 15.250 & 182.5616 & 4.71 & 1.2678 & 244.14 & 10.58 \\
\hline & 18.00 & 140 & & & 1.562 & 14.876 & 173.7169 & 4.71 & 1.2064 & 274.22 & 10.06 \\
\hline & 18.00 & 160 & & & 1.781 & 14.438 & 163.6378 & 4.71 & 1.1364 & 308.50 & 9.48 \\
\hline 20 & 20.00 & & & 5 S & 0.188 & 19.624 & 302.3046 & 5.24 & 2.0993 & 39.78 & 17.51 \\
\hline & 20.00 & & & 10S & 0.218 & 19.564 & 300.4588 & 5.24 & 2.0865 & 46.06 & 17.41 \\
\hline & 20.00 & 10 & & & 0.250 & 19.500 & 298.4963 & 5.24 & 2.0729 & 52.73 & 17.29 \\
\hline & 20.00 & & & & 0.312 & 19.376 & 294.7121 & 5.24 & 2.0466 & 65.60 & 17.07 \\
\hline & 20.00 & 20 & Std & & 0.375 & 19.250 & 290.8916 & 5.24 & 2.0201 & 78.60 & 16.85 \\
\hline & 20.00 & & & & 0.437 & 19.126 & 287.1560 & 5.24 & 1.9941 & 91.30 & 16.63 \\
\hline & 20.00 & 30 & XS & & 0.500 & 19.000 & 283.3850 & 5.24 & 1.9680 & 104.13 & 16.42 \\
\hline & 20.00 & & & & 0.562 & 18.876 & 279.6982 & 5.24 & 1.9424 & 116.67 & 16.20 \\
\hline & 20.00 & 40 & & & 0.593 & 18.814 & 277.8638 & 5.24 & 1.9296 & 122.91 & 16.10 \\
\hline & 20.00 & & & & 0.625 & 18.750 & 275.9766 & 5.24 & 1.9165 & 129.33 & 15.99 \\
\hline & 20.00 & & & & 0.687 & 18.626 & 272.3384 & 5.24 & 1.8912 & 141.70 & 15.78 \\
\hline & 20.00 & & & & 0.750 & 18.500 & 268.6663 & 5.24 & 1.8657 & 154.19 & 15.56 \\
\hline & 20.00 & 60 & & & 0.812 & 18.376 & 265.0767 & 5.24 & 1.8408 & 166.40 & 15.36 \\
\hline & 20.00 & & & & 0.875 & 18.250 & 261.4541 & 5.24 & 1.8157 & 178.72 & 15.15 \\
\hline & 20.00 & 80 & & & 1.031 & 17.938 & 252.5909 & 5.24 & 1.7541 & 208.87 & 14.63 \\
\hline & 20.00 & 100 & & & 1.281 & 17.438 & 238.7058 & 5.24 & 1.6577 & 256.10 & 13.83 \\
\hline & 20.00 & 120 & & & 1.500 & 17.000 & 226.8650 & 5.24 & 1.5755 & 296.37 & 13.14 \\
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\begin{tabular}{|c|c|c|c|c|c|c|c|c|c|c|c|}
\hline \multirow[t]{2}{*}{Nominal Pipe Size, NPS} & \multirow[t]{2}{*}{Outside Dia, in.} & \multicolumn{3}{|l|}{Schedule} & \multirow[t]{2}{*}{\begin{tabular}{l}
Wall \\
Thickness, in.
\end{tabular}} & \multirow[t]{2}{*}{Inside Dia, in.} & \multirow[t]{2}{*}{\begin{tabular}{l}
Inside \\
Area, in. \({ }^{2}\)
\end{tabular}} & \multirow[t]{2}{*}{Surface Area, \(\mathrm{ft}^{2} / \mathrm{ft}\)} & \multirow[t]{2}{*}{Volume, \(\mathrm{ft}^{3} / \mathrm{ft}\)} & \multirow[t]{2}{*}{Pipe Weight, lb/ft} & \multirow[t]{2}{*}{Water Weight, lb/ft} \\
\hline & & a & b & c & & & & & & & \\
\hline & 26.00 & & Std & & 0.375 & 25.250 & 500.4866 & 6.81 & 3.4756 & 102.63 & 28.99 \\
\hline & 26.00 & 20 & XS & & 0.500 & 25.000 & 490.6250 & 6.81 & 3.4071 & 136.17 & 28.42 \\
\hline & 26.00 & & & & 0.625 & 24.750 & 480.8616 & 6.81 & 3.3393 & 169.38 & 27.86 \\
\hline & 26.00 & & & & 0.750 & 24.500 & 471.1963 & 6.81 & 3.2722 & 202.25 & 27.30 \\
\hline & 26.00 & & & & 0.875 & 24.250 & 461.6291 & 6.81 & 3.2058 & 234.79 & 26.74 \\
\hline & 26.00 & & & & 1.000 & 24.000 & 452.1600 & 6.81 & 3.1400 & 267.00 & 26.19 \\
\hline & 26.00 & & & & 1.125 & 23.750 & 442.7891 & 6.81 & 3.0749 & 298.87 & 25.65 \\
\hline 28 & 28.00 & & & & 0.250 & 27.500 & 593.6563 & 7.33 & 4.1226 & 74.09 & 34.39 \\
\hline & 28.00 & 10 & & & 0.312 & 27.376 & 588.3146 & 7.33 & 4.0855 & 92.26 & 34.08 \\
\hline & 28.00 & & Std & & 0.375 & 27.250 & 582.9116 & 7.33 & 4.0480 & 110.64 & 33.77 \\
\hline & 28.00 & 20 & XS & & 0.500 & 27.000 & 572.2650 & 7.33 & 3.9741 & 146.85 & 33.15 \\
\hline & 28.00 & 30 & & & 0.625 & 26.750 & 561.7166 & 7.33 & 3.9008 & 182.73 & 32.54 \\
\hline & 28.00 & & & & 0.750 & 26.500 & 551.2663 & 7.33 & 3.8282 & 218.27 & 31.93 \\
\hline & 28.00 & & & & 0.875 & 26.250 & 540.9141 & 7.33 & 3.7564 & 253.48 & 31.33 \\
\hline & 28.00 & & & & 1.000 & 26.000 & 530.6600 & 7.33 & 3.6851 & 288.36 & 30.74 \\
\hline & 28.00 & & & & 1.125 & 25.750 & 520.5041 & 7.33 & 3.6146 & 322.90 & 30.15 \\
\hline 30 & 30.00 & & & 5 S & 0.250 & 29.500 & 683.1463 & 7.85 & 4.7441 & 79.43 & 39.57 \\
\hline & 30.00 & 10 & & 10 S & 0.312 & 29.376 & 677.4153 & 7.85 & 4.7043 & 98.93 & 39.24 \\
\hline & 30.00 & & Std & & 0.375 & 29.250 & 671.6166 & 7.85 & 4.6640 & 118.65 & 38.91 \\
\hline & 30.00 & 20 & XS & & 0.500 & 29.000 & 660.1850 & 7.85 & 4.5846 & 157.53 & 38.24 \\
\hline & 30.00 & 30 & & & 0.625 & 28.750 & 648.8516 & 7.85 & 4.5059 & 196.08 & 37.59 \\
\hline & 30.00 & 40 & & & 0.750 & 28.500 & 637.6163 & 7.85 & 4.4279 & 234.29 & 36.94 \\
\hline & 30.00 & & & & 0.875 & 28.250 & 626.4791 & 7.85 & 4.3506 & 272.17 & 36.29 \\
\hline & 30.00 & & & & 1.000 & 28.000 & 615.4400 & 7.85 & 4.2739 & 309.72 & 35.65 \\
\hline & 30.00 & & & & 1.125 & 27.750 & 604.4991 & 7.85 & 4.1979 & 346.93 & 35.02 \\
\hline 32 & 32.00 & & & & 0.250 & 31.500 & 778.9163 & 8.38 & 5.4092 & 84.77 & 45.12 \\
\hline & 32.00 & 10 & & & 0.312 & 31.376 & 772.7959 & 8.38 & 5.3667 & 105.59 & 44.77 \\
\hline & 32.00 & & Std & & 0.375 & 31.250 & 766.6016 & 8.38 & 5.3236 & 126.66 & 44.41 \\
\hline & 32.00 & 20 & XS & & 0.500 & 31.000 & 754.3850 & 8.38 & 5.2388 & 168.21 & 43.70 \\
\hline & 32.00 & 30 & & & 0.625 & 30.750 & 742.2666 & 8.38 & 5.1546 & 209.43 & 43.00 \\
\hline & 32.00 & 40 & & & 0.688 & 30.624 & 736.1961 & 8.38 & 5.1125 & 230.08 & 42.65 \\
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\end{tabular}



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\section*{APPENDIX D}

\section*{GASMOD Output Report}

The following is a report from GASMOD—Gas Pipeline Hydraulics Simulation software (www.systek.us)-for a pipeline transporting natural gas in an NPS 16 pipeline 450 miles long from Compton to Harvard.
```

    ******* GASMOD - GAS PIPELINE HYDRAULIC SIMULATION ********
    ************ 32-bit Version 5.00.200
    DATE:
21-September-2004 TIME: 09:45:06
PROJECT DESCRIPTION:
Pipeline from Compton to Harvard
16" pipeline
Case Number: 1264
Pipeline data file: C:\GASMOD32\MYPIPE001.TOT
Pressure drop formula: AGA Turbulent
Pipeline efficiency: 1.00
Compressibility Factor Method: AGA NX19
Inlet Gas Gravity(Air=1.0): 0.67883
Inlet Gas Viscosity:
CALCULATION OPTIONS:
Branch pipe calculations: NO
Loop pipe calculations: NO
Compressor Fuel Calculated: YES
Joule Thompson effect included: NO
Customized Output: NO
Holding Delivery Pressure
at terminus

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```

****************** Calculations Based on Specified Thermal
Conductivities of Pipe, Soil and Insulation **************
Origin suction temperature:
Base temperature:
Base pressure:
Origin suction pressure:
Delivery pressure:
Minimum pressure:
Gas specific heat ratio:
Maximum gas velocity:
Inlet Flow rate:
Outlet Flow rate: 128.2744(MMSCFD)
70.00(degF)
60.00 (degF)
50.0000 (MMSCFD)

```

\section*{PIPELINE PROFILE DATA}
\begin{tabular}{rrrcc}
\begin{tabular}{c} 
Distance \\
(mi)
\end{tabular} & \begin{tabular}{c} 
Elevation \\
(ft)
\end{tabular} & \begin{tabular}{c} 
Diameter \\
(in)
\end{tabular} & \begin{tabular}{c} 
Thickness \\
(in)
\end{tabular} & \begin{tabular}{c} 
Roughness \\
(in)
\end{tabular} \\
0.00 & 620.00 & 16.000 & 0.375 & 0.000700 \\
15.00 & 969.70 & 16.000 & 0.375 & 0.000700 \\
20.00 & 1086.26 & 16.000 & 0.375 & 0.000700 \\
30.00 & 1319.39 & 16.000 & 0.375 & 0.000700 \\
42.00 & 1599.14 & 16.000 & 0.375 & 0.000700 \\
45.00 & 1669.08 & 16.000 & 0.375 & 0.000700 \\
48.90 & 1760.00 & 16.000 & 0.375 & 0.000700 \\
85.00 & 2929.00 & 16.000 & 0.375 & 0.000700 \\
128.00 & 1260.00 & 16.000 & 0.375 & 0.000700 \\
130.00 & 1338.46 & 16.000 & 0.375 & 0.000700 \\
140.00 & 1730.77 & 16.000 & 0.375 & 0.000700 \\
150.00 & 2123.08 & 16.000 & 0.375 & 0.000700 \\
154.00 & 2280.00 & 16.000 & 0.375 & 0.000700 \\
155.00 & 2263.53 & 16.000 & 0.375 & 0.000700 \\
160.00 & 2181.18 & 16.000 & 0.375 & 0.000700 \\
180.00 & 1851.80 & 16.000 & 0.375 & 0.000700 \\
238.40 & 890.00 & 16.000 & 0.375 & 0.000700 \\
240.00 & 883.18 & 16.000 & 0.375 & 0.000700 \\
250.00 & 840.54 & 16.000 & 0.375 & 0.000700 \\
260.00 & 797.91 & 16.000 & 0.375 & 0.000700 \\
290.00 & 670.00 & 16.000 & 0.375 & 0.000700 \\
292.00 & 727.00 & 16.000 & 0.375 & 0.000700 \\
300.00 & 955.00 & 16.000 & 0.375 & 0.000700 \\
320.00 & 1525.00 & 16.000 & 0.375 & 0.000700 \\
330.00 & 1563.33 & 16.000 & 0.375 & 0.000700 \\
346.00 & 1624.67 & 16.000 & 0.375 & 0.000700 \\
350.00 & 1640.00 & 16.000 & 0.375 & 0.000700 \\
356.00 & 1543.40 & 16.000 & 0.375 & 0.000700 \\
360.00 & 1479.00 & 16.000 & 0.375 & 0.000700 \\
400.00 & 835.00 & 16.000 & 0.375 & 0.000700 \\
420.00 & 881.00 & 16.000 & 0.375 & 0.000700 \\
450.00 & 950.00 & 16.000 & 0.375 & 0.000700
\end{tabular}
\begin{tabular}{|c|c|c|c|c|c|c|}
\hline \multirow[b]{2}{*}{\[
\begin{gathered}
\text { Distance } \\
\text { (mi) }
\end{gathered}
\]} & \multirow[b]{2}{*}{\begin{tabular}{l}
Cover \\
(in)
\end{tabular}} & \multicolumn{3}{|l|}{Thermal Conductivity (Btu/hr/ft/degF)} & \multirow[t]{2}{*}{\[
\begin{aligned}
& \text { Insul.Thk } \\
& \text { (in) }
\end{aligned}
\]} & \multirow[t]{2}{*}{Soil Temp (degF)} \\
\hline & & Pipe & Soil & Insulation & & \\
\hline 0.000 & 36.000 & 29.000 & 0.800 & 0.200 & 0.000 & 80.00 \\
\hline 15.000 & 36.000 & 29.000 & 0.800 & 0.200 & 0.000 & 80.00 \\
\hline 20.000 & 36.000 & 29.000 & 0.800 & 0.200 & 0.000 & 80.00 \\
\hline 30.000 & 36.000 & 29.000 & 0.800 & 0.200 & 0.000 & 80.00 \\
\hline 42.000 & 36.000 & 29.000 & 0.800 & 0.200 & 0.000 & 80.00 \\
\hline 45.000 & 36.000 & 29.000 & 0.800 & 0.200 & 0.000 & 80.00 \\
\hline 48.900 & 36.000 & 29.000 & 0.800 & 0.200 & 0.000 & 80.00 \\
\hline 85.000 & 36.000 & 29.000 & 0.800 & 0.200 & 0.000 & 80.00 \\
\hline 128.000 & 36.000 & 29.000 & 0.800 & 0.200 & 0.000 & 80.00 \\
\hline 130.000 & 36.000 & 29.000 & 0.800 & 0.200 & 0.000 & 80.00 \\
\hline 140.000 & 36.000 & 29.000 & 0.800 & 0.200 & 0.000 & 80.00 \\
\hline 150.000 & 36.000 & 29.000 & 0.800 & 0.200 & 0.000 & 80.00 \\
\hline 154.000 & 36.000 & 29.000 & 0.800 & 0.200 & 0.000 & 80.00 \\
\hline 155.000 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline 160.000 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline 180.000 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline 238.400 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline 240.000 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline 250.000 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline 260.000 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline 290.000 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline 292.000 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline 300.000 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline 320.000 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline 330.000 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline 346.000 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline 350.000 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline 356.000 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline 360.000 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline 400.000 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline 420.000 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline 450.000 & 36.000 & 29.000 & 0.750 & 0.200 & 0.000 & 80.00 \\
\hline
\end{tabular}
**************** COMPRESSOR STATION DATA

FLOW RATES, PRESSURES AND TEMPERATURES:
\begin{tabular}{|c|c|c|c|c|c|c|c|c|c|}
\hline Name & \[
\begin{gathered}
\text { Flow } \\
\text { Rate } \\
\text { (MMSCFD) }
\end{gathered}
\] & Suct. Press. (psig) & \begin{tabular}{l}
Disch. \\
Press. \\
(psig)
\end{tabular} & Compr. Ratio & \begin{tabular}{l}
Suct. \\
Loss. (psig)
\end{tabular} & Disch. Loss. (psig) & \begin{tabular}{l}
Suct. \\
Temp. \\
(degF)
\end{tabular} & Disch. Temp (degF) & MaxPipe Temp (degF) \\
\hline Compton & 149.39 & 795.00 & 1210.00 & 1.5125 & 5.00 & 10.00 & 70.00 & 129.17 & 140.00 \\
\hline Sta-2 & 98.90 & 752.05 & 1210.00 & 1.5973 & 5.00 & 10.00 & 80.00 & 148.62 & 140.00 \\
\hline Sta-4 & 98.45 & 768.45 & 1210.00 & 1.5638 & 5.00 & 10.00 & 80.00 & 145.37 & 140 \\
\hline Sta-5 & 98.27 & 953.88 & 1149.13 & 1.2016 & 5.00 & 10.00 & 80.00 & 106.12 & 14 \\
\hline
\end{tabular}

INACTIVE COMPRESSOR STATIONS:
\begin{tabular}{lc} 
& \begin{tabular}{c} 
Distance \\
(mi)
\end{tabular} \\
Name & \\
Dimpton & 180 \\
Jackson & 420
\end{tabular}
********* COMPRESSOR EFFICIENCY, HP AND FUEL USED *********
\begin{tabular}{|c|c|c|c|c|c|c|c|}
\hline \multirow[b]{2}{*}{Name} & \multirow[b]{2}{*}{\[
\begin{gathered}
\text { Distance } \\
\text { (mi) }
\end{gathered}
\]} & & & & \multicolumn{3}{|c|}{Fuel} \\
\hline & & Compr Effy. (\%) & Mech. Effy. (\%) & \begin{tabular}{l}
Overall Effy. \\
(\%)
\end{tabular} & Horse Power & \[
\begin{gathered}
\text { Factor } \\
\text { (MCF/ } \\
\text { day/HP) }
\end{gathered}
\] & \[
\begin{gathered}
\text { Fuel } \\
\text { Used } \\
\text { (MMSCFD) }
\end{gathered}
\] \\
\hline Compton & 0.00 & 80.00 & 98.00 & 78.40 & 3,059.55 & 0.2000 & 0.6119 \\
\hline Sta-2 & 85.00 & 80.00 & 98.00 & 78.40 & 2,419.97 & 0.2000 & 0.4840 \\
\hline Sta-4 & 250.00 & 80.00 & 98.00 & 78.40 & 2,284.08 & 0.2000 & 0.4568 \\
\hline Sta-5 & 330.00 & 80.00 & 98.00 & 78.40 & 864.48 & 0.2000 & 0.1729 \\
\hline Total C & mpressor & Station & Horse & ower: & 8,628.08 & & \\
\hline
\end{tabular}

\section*{LOCATIONS AND FLOW RATES}
\begin{tabular}{|c|c|c|c|c|c|c|c|}
\hline Location & Distance (mi) & \[
\begin{aligned}
& \text { Flow } \\
& \text { in/out } \\
& \text { (MMSCFD) }
\end{aligned}
\] & Gravity & \begin{tabular}{l}
Viscosity \\
(lb/ft-sec)
\end{tabular} & Pressure (psig) & GasTemp. (degF) & GasName \\
\hline Compton & 0.00 & 150.0000 & 0.6788 & 0.00000700 & 1200.00 & 129.17 & \[
\begin{aligned}
& \text { SAN JUAN } \\
& \text { GAS }
\end{aligned}
\] \\
\hline & 30.00 & -50.0000 & 0.6788 & 0.00000700 & 986.29 & 83.52 & \\
\hline & 350.00 & 30.0000 & 0.6000 & 0.00000800 & 1083.08 & 82.05 & \\
\hline Harvard & 450.00 & -128.2744 & 0.6606 & 0.00000723 & 500.33 & 80.00 & \\
\hline
\end{tabular}
```

****** REYNOLD'S NUMBER AND HEAT TRANSFER COEFFICIENT *****

```
\begin{tabular}{crcccc}
\begin{tabular}{c} 
Distance \\
(mi)
\end{tabular} & Reynold'sNum. & \multicolumn{4}{c}{\begin{tabular}{c} 
FrictFactor \\
(Darcy)
\end{tabular}} \\
& & & \begin{tabular}{c} 
TratTrans- \\
Coeff \\
Factor
\end{tabular} & \begin{tabular}{c} 
Compressibility- \\
(Btu/hr/ \\
ft2/degF)
\end{tabular} & \begin{tabular}{c} 
Factor \\
(AGA NX19)
\end{tabular} \\
0.000 & \(12,831,042\). & 0.0104 & 19.63 & 0.4999 & 0.8027 \\
15.000 & \(12,831,042\). & 0.0104 & 19.63 & 0.4999 & 0.7887 \\
20.000 & \(12,831,042\). & 0.0104 & 19.63 & 0.4999 & 0.7913 \\
30.000 & \(8,536,437\). & 0.0104 & 19.63 & 0.4992 & 0.7979 \\
42.000 & \(8,536,437\). & 0.0104 & 19.63 & 0.4992 & 0.8024 \\
45.000 & \(8,536,437\). & 0.0104 & 19.63 & 0.4992 & 0.8052 \\
48.900 & \(8,536,437\). & 0.0104 & 19.63 & 0.4992 & 0.8241 \\
85.000 & \(8,493,784\). & 0.0104 & 19.63 & 0.4991 & 0.7672 \\
128.000 & \(8,493,784\). & 0.0104 & 19.63 & 0.4991 & 0.7518 \\
130.000 & \(8,493,784\). & 0.0104 & 19.63 & 0.4991 & 0.7574 \\
140.000 & \(8,493,784\). & 0.0104 & 19.63 & 0.4991 & 0.7670
\end{tabular}
\begin{tabular}{rrrrrr}
150.000 & \(8,493,784\). & 0.0104 & 19.63 & 0.4991 & 0.7738 \\
154.000 & \(8,493,784\). & 0.0104 & 19.63 & 0.4991 & 0.7761 \\
155.000 & \(8,493,784\). & 0.0104 & 19.63 & 0.4681 & 0.7777 \\
160.000 & \(8,493,784\). & 0.0104 & 19.63 & 0.4681 & 0.7849 \\
180.000 & \(8,493,784\). & 0.0104 & 19.63 & 0.4681 & 0.8094 \\
238.400 & \(8,493,784\). & 0.0104 & 19.63 & 0.4681 & 0.8298 \\
240.000 & \(8,493,784\). & 0.0104 & 19.63 & 0.4681 & 0.8345 \\
250.000 & \(8,451,888\). & 0.0104 & 19.63 & 0.4681 & 0.8069 \\
260.000 & \(8,451,888\). & 0.0104 & 19.63 & 0.4681 & 0.7629 \\
290.000 & \(8,451,888\). & 0.0104 & 19.63 & 0.4681 & 0.7635 \\
292.000 & \(8,451,888\). & 0.0104 & 19.63 & 0.4681 & 0.7676 \\
300.000 & \(8,451,888\). & 0.0104 & 19.63 & 0.4681 & 0.7797 \\
320.000 & \(8,451,888\). & 0.0104 & 19.63 & 0.4681 & 0.7923 \\
330.000 & \(8,435,063\). & 0.0104 & 19.63 & 0.4681 & 0.7808 \\
346.000 & \(8,435,063\). & 0.0104 & 19.63 & 0.4681 & 0.7706 \\
350.000 & \(10,716,727\). & 0.0104 & 19.63 & 0.4685 & 0.7892 \\
356.000 & \(10,716,727\). & 0.0104 & 19.63 & 0.4685 & 0.7925 \\
360.000 & \(10,716,727\). & 0.0104 & 19.63 & 0.4685 & 0.8124 \\
400.000 & \(10,716,727\). & 0.0104 & 19.63 & 0.4685 & 0.8451 \\
420.000 & \(10,716,727\). & 0.0104 & 19.63 & 0.4685 & 0.8805 \\
450.000 & \(10,716,727\). & 0.0104 & 19.63 & 0.4685 & 0.8805
\end{tabular}
\begin{tabular}{|c|c|c|c|c|c|c|c|c|}
\hline \[
\begin{gathered}
\text { Distance } \\
\text { (mi) }
\end{gathered}
\] & \[
\begin{gathered}
\text { Diameter } \\
\text { (in) }
\end{gathered}
\] & \begin{tabular}{l}
Flow \\
(MMSCFD)
\end{tabular} & \begin{tabular}{l}
Velocity \\
(ft/sec)
\end{tabular} & \begin{tabular}{l}
Press. \\
(psig)
\end{tabular} & \begin{tabular}{l}
GasTemp. \\
(degF)
\end{tabular} & \begin{tabular}{l}
SoilTemp. \\
(degF)
\end{tabular} & MAOP (psig) & Location \\
\hline 0.00 & 16.000 & 149.3881 & 16.53 & 1200.00 & 129.17 & 80.00 & 1400.00 & Compton \\
\hline 15.00 & 16.000 & 149.3881 & 18.08 & 1095.78 & 93.61 & 80.00 & 1400.00 & \\
\hline 20.00 & 16.000 & 149.3881 & 18.68 & 1060.08 & 88.70 & 80.00 & 1400.00 & \\
\hline 30.00 & 16.000 & 99.3881 & 13.34 & 986.29 & 83.52 & 80.00 & 1400.00 & \\
\hline 42.00 & 16.000 & 99.3881 & 13.97 & 940.95 & 80.68 & 80.00 & 1400.00 & \\
\hline 45.00 & 16.000 & 99.3881 & 14.15 & 929.42 & 80.45 & 80.00 & 1400.00 & Branch1 \\
\hline 48.90 & 16.000 & 99.3881 & 14.38 & 914.30 & 80.26 & 80.00 & 1400.00 & \\
\hline 85.00 & 16.000 & 98.9041 & 17.22 & 757.05 & 80.00 & 80.00 & 1400.00 & Sta-2 \\
\hline 85.00 & 16.000 & 98.9041 & 10.94 & 1200.00 & 140.00 & 80.00 & 1400.00 & Sta-2 \\
\hline 128.00 & 16.000 & 98.9041 & 11.42 & 1149.26 & 80.23 & 80.00 & 1400.00 & \\
\hline 130.00 & 16.000 & 98.9041 & 11.50 & 1141.23 & 80.17 & 80.00 & 1400.00 & \\
\hline 140.00 & 16.000 & 98.9041 & 11.91 & 1100.95 & 80.04 & 80.00 & 1400.00 & \\
\hline 150.00 & 16.000 & 98.9041 & 12.36 & 1060.36 & 80.01 & 80.00 & 1400.00 & \\
\hline 154.00 & 16.000 & 98.9041 & 12.55 & 1044.01 & 80.01 & 80.00 & 1400.00 & \\
\hline 155.00 & 16.000 & 98.9041 & 12.58 & 1041.70 & 80.01 & 80.00 & 1400.00 & \\
\hline 160.00 & 16.000 & 98.9041 & 12.72 & 1030.05 & 80.00 & 80.00 & 1400.00 & \\
\hline 180.00 & 16.000 & 98.9041 & 13.34 & 981.36 & 80.00 & 80.00 & 1400.00 & Dimpton \\
\hline 238.40 & 16.000 & 98.9041 & 16.01 & 815.32 & 80.00 & 80.00 & 1400.00 & \\
\hline 240.00 & 16.000 & 98.9041 & 16.12 & 809.68 & 80.00 & 80.00 & 1400.00 & \\
\hline 250.00 & 16.000 & 98.4473 & 16.78 & 773.45 & 80.00 & 80.00 & 1400.00 & Sta-4 \\
\hline 250.00 & 16.000 & 98.4473 & 10.89 & 1200.00 & 140.00 & 80.00 & 1400.00 & Sta-4 \\
\hline 260.00 & 16.000 & 98.4473 & 11.12 & 1175.10 & 98.11 & 80.00 & 1400.00 & \\
\hline 290.00 & 16.000 & 98.4473 & 11.85 & 1101.69 & 80.39 & 80.00 & 1400.00 & \\
\hline 292.00 & 16.000 & 98.4473 & 11.93 & 1094.39 & 80.30 & 80.00 & 1400.00 & \\
\hline 300.00 & 16.000 & 98.4473 & 12.25 & 1065.01 & 80.11 & 80.00 & 1400.00 & \\
\hline 320.00 & 16.000 & 98.4473 & 13.17 & 990.04 & 80.01 & 80.00 & 1400.00 & \\
\hline 330.00 & 16.000 & 98.2744 & 13.56 & 958.88 & 80.00 & 80.00 & 1400.00 & Sta-5 \\
\hline 330.00 & 16.000 & 98.2744 & 11.44 & 1139.13 & 106.12 & 80.00 & 1400.00 & Sta-5 \\
\hline 346.00 & 16.000 & 98.2744 & 11.91 & 1094.37 & 83.45 & 80.00 & 1400.00 & \\
\hline
\end{tabular}
\begin{tabular}{lrrrrrrrl}
350.00 & 16.000 & 128.2744 & 15.70 & 1083.08 & 82.05 & 80.00 & 1400.00 \\
356.00 & 16.000 & 128.2744 & 16.05 & 1058.92 & 81.11 & 80.00 & 1400.00 \\
360.00 & 16.000 & 128.2744 & 16.30 & 1042.45 & 80.74 & 80.00 & 1400.00 \\
400.00 & 16.000 & 128.2744 & 19.78 & 856.54 & 80.01 & 80.00 & 1400.00 \\
420.00 & 16.000 & 128.2744 & 23.01 & 734.40 & 80.00 & 80.00 & 1400.00 & Jackson \\
450.00 & 16.000 & 128.2744 & 33.47 & 500.33 & 80.00 & 80.00 & 1400.00 Harvard
\end{tabular}

\section*{LINE PACK VOLUMES AND PRESSURES}
\begin{tabular}{|c|c|c|}
\hline \[
\begin{gathered}
\text { Distance } \\
\text { (mi) }
\end{gathered}
\] & Pressure (psig) & Line Pack (million std.cu.ft) \\
\hline 0.00 & 1200.00 & 0.0000 \\
\hline 15.00 & 1095.78 & 9.6000 \\
\hline 20.00 & 1060.08 & 3.0086 \\
\hline 30.00 & 986.29 & 5.7524 \\
\hline 42.00 & 940.95 & 6.4677 \\
\hline 45.00 & 929.42 & 1.5581 \\
\hline 48.90 & 914.30 & 1.9909 \\
\hline 85.00 & 757.05 & 16.4027 \\
\hline 128.00 & 1149.26 & 23.2986 \\
\hline 130.00 & 1141.23 & 1.3544 \\
\hline 140.00 & 1100.95 & 6.5846 \\
\hline 150.00 & 1060.36 & 6.2702 \\
\hline 154.00 & 1044.01 & 2.4211 \\
\hline 155.00 & 1041.70 & 0.5982 \\
\hline 160.00 & 1030.05 & 2.9652 \\
\hline 180.00 & 981.36 & 11.4175 \\
\hline 238.40 & 815.32 & 28.9974 \\
\hline 240.00 & 809.68 & 0.7002 \\
\hline 250.00 & 773.45 & 4.2420 \\
\hline 260.00 & 1175.10 & 5.1998 \\
\hline 290.00 & 1101.69 & 20.1395 \\
\hline 292.00 & 1094.39 & 1.2790 \\
\hline 300.00 & 1065.01 & 5.0078 \\
\hline 320.00 & 990.04 & 11.7426 \\
\hline 330.00 & 958.88 & 5.4816 \\
\hline 346.00 & 1094.37 & 9.2580 \\
\hline 350.00 & 1083.08 & 2.5092 \\
\hline 356.00 & 1058.92 & 3.6990 \\
\hline 360.00 & 1042.45 & 2.4088 \\
\hline 400.00 & 856.54 & 21.2972 \\
\hline 420.00 & 734.40 & 8.5521 \\
\hline 450.00 & 500.33 & 9.6571 \\
\hline
\end{tabular}

Total line pack in main pipeline \(=239.8614(m i l l i o n ~ s t d . c u . f t)\)

Started simulation at: 09:44:17
Finished simulation at: 09:45:06
Time elapsed: 49 seconds
DATE: 21-September-2004

\section*{appendix E}

\section*{Summary of Formulas}

\section*{CHAPTER 1}
1. Density
\[
\begin{equation*}
\rho=\frac{m}{V} \tag{1.1}
\end{equation*}
\]
where
\(\rho=\) density of gas
\(m=\) mass of gas
\(V=\) volume of gas
2. Gas gravity
\[
\begin{equation*}
G=\frac{\rho_{g}}{\rho_{\text {air }}} \tag{1.2}
\end{equation*}
\]
where
\(G=\) gas gravity, dimensionless
\(\rho_{g}=\) density of gas
\(\rho_{\text {air }}=\) density of air
\[
\begin{equation*}
G=\frac{M_{g}}{29} \tag{1.4}
\end{equation*}
\]
where
\(M_{g}=\) molecular weight of gas
\(M_{\text {air }}=\) molecular weight of air \(=28.9625\)
3. Kinematic viscosity
\[
\begin{equation*}
v=\frac{\mu}{\rho} \tag{1.5}
\end{equation*}
\]
where, in USCS units,
\(v=\) kinematic viscosity, \(\mathrm{ft}^{2} / \mathrm{s}\)
\(\mu=\) dynamic viscosity, lb/ft-s
\(\rho=\) density, \(\mathrm{lb} / \mathrm{ft}^{3}\)
and, in SI units,
\(v=\) kinematic viscosity, cSt
\(\mu=\) dynamic viscosity, cP
\(\rho=\) density, \(\mathrm{kg} / \mathrm{m}^{3}\)
4. Viscosity of mixture
\[
\begin{equation*}
\mu=\frac{\Sigma\left(\mu_{i} y_{i} \sqrt{M_{i}}\right)}{\Sigma\left(y_{i} \sqrt{M_{i}}\right)} \tag{1.6}
\end{equation*}
\]
where
\(\mu=\) dynamic viscosity of gas mixture
\(\mu_{i}=\) dynamic viscosity of gas component \(i\)
\(y_{i}=\) mole fraction or percent of gas component \(i\)
\(M_{i}=\) molecular weight of gas component \(i\)
5. Ideal gas law or perfect gas equation
\[
\begin{equation*}
P V=n R T \tag{1.8}
\end{equation*}
\]
where
\(P=\) absolute pressure, pounds per square inch absolute (psia)
\(V=\) gas volume, \(\mathrm{ft}^{3}\)
\(n=\) number of lb moles as defined in Equation 1.7
\(R=\) universal gas constant, \(\mathrm{psia}_{\mathrm{ft}}{ }^{3} / \mathrm{lb}\) mole \({ }^{\circ} \mathrm{R}\)
\(T=\) absolute temperature of gas, \({ }^{\circ} \mathrm{R}\left({ }^{\circ} \mathrm{F}+460\right)\)
6. Absolute pressure
\[
\begin{equation*}
P_{\mathrm{abs}}=P_{\mathrm{gauge}}+P_{\mathrm{atm}} \tag{1.10}
\end{equation*}
\]
7. Boyle's law
\[
\begin{equation*}
\frac{P_{1}}{P_{2}}=\frac{V_{2}}{V_{1}} \text { or } P_{1} V_{1}=P_{2} V_{2} \tag{1.13}
\end{equation*}
\]
8. Charles's law
\[
\begin{align*}
& \frac{V_{1}}{V_{2}}=\frac{T_{1}}{T_{2}} \quad \text { at constant pressure }  \tag{1.14}\\
& \frac{P_{1}}{P_{2}}=\frac{T_{1}}{T_{2}} \quad \text { at constant volume } \tag{1.15}
\end{align*}
\]
9. Modified ideal gas equation
\[
\begin{equation*}
P V=Z n R T \quad \text { (USCS units) } \tag{1.16}
\end{equation*}
\]
where
\(P=\) absolute pressure of gas, psia
\(V=\) volume of gas, \(\mathrm{ft}^{3}\)
\(Z=\) gas compressibility factor, dimensionless
\(T=\) absolute temperature of gas, \({ }^{\circ} \mathrm{R}\)
\(n=\) number of lb moles as defined in Equation 1.7
\(R=\) universal gas constant, \(10.73 \mathrm{psia}^{3}{ }^{3} / \mathrm{lb}^{2}\) mole \({ }^{\circ} \mathrm{R}\)
10. Reduced temperature and reduced pressure
\[
\begin{align*}
& T_{r}=\frac{T}{T_{c}}  \tag{1.17}\\
& P_{r}=\frac{P}{P_{c}} \tag{1.18}
\end{align*}
\]
11. Pseudo-reduced temperature and pseudo-reduced pressure
\[
\begin{align*}
& T_{p r}=\frac{T}{T_{p c}}  \tag{1.19}\\
& P_{p r}=\frac{P}{P_{p c}} \tag{1.20}
\end{align*}
\]
where
\(P=\) absolute pressure of gas mixture, psia
\(T=\) absolute temperature of gas mixture, \({ }^{\circ} \mathrm{R}\)
\(T_{p r}=\) pseudo-reduced temperature, dimensionless
\(P_{p r}=\) pseudo-reduced pressure, dimensionless
\(T_{p c}=\) pseudo-critical temperature, \({ }^{\circ} \mathrm{R}\)
\(P_{p c}=\) pseudo-critical pressure, psia
12. Apparent molecular weight of gas mixture
\[
\begin{equation*}
M_{a}=\Sigma y_{i} M_{i} \tag{1.21}
\end{equation*}
\]
where
\(M_{a}=\) apparent molecular weight of gas mixture
\(y_{i}=\) mole fraction of gas component \(i\)
\(M_{i}=\) molecular weight of gas component \(i\)
13. Kay's rule to calculate the average pseudo-critical properties of the gas mixture
\[
\begin{align*}
& T_{p c}=\Sigma y_{i} T_{c}  \tag{1.22}\\
& P_{p c}=\Sigma y_{i} P_{c} \tag{1.23}
\end{align*}
\]
14. Pseudo-critical properties from gas gravity
\[
\begin{align*}
& T_{p c}=170.491+307.344 G  \tag{1.24}\\
& P_{p c}=709.604-58.718 G \tag{1.25}
\end{align*}
\]
where
\(G=\) gas gravity (air = 1.00)
\(T_{p c}=\) pseudo-critical temperature, \({ }^{\circ} \mathrm{R}\)
\(P_{p c}=\) pseudo-critical pressure, psia
15. Supercompressibility factor
\[
\begin{equation*}
Z=\frac{1}{\left(F_{p v}\right)^{2}} \tag{1.30}
\end{equation*}
\]
16. Dranchuk, Purvis, and Robinson method
\[
\begin{equation*}
Z=1+\left(A+\frac{A_{2}}{T_{p r}}+\frac{A_{3}}{T_{p r}^{3}}\right) \rho_{r}+\left(A_{4}+\frac{A_{5}}{T_{p r}}\right) \rho_{r}^{2}+\frac{A_{5} A_{6} \rho_{r}^{5}}{T_{p r}}+\frac{A_{7} \rho_{r}^{3}}{T_{p r}^{3}\left(1+A_{8} \rho_{r}^{2}\right) e^{\left(-A_{8} \rho_{r}^{2}\right)}} \tag{1.31}
\end{equation*}
\]
where
\[
\begin{equation*}
\rho_{r}=\frac{0.27 P_{p r}}{Z T_{p r}} \tag{1.32}
\end{equation*}
\]
and
\(A_{1}=0.31506237 ; \quad A_{2}=-1.04670990 ;\)
\(A_{3}=-0.57832729 ; \quad A_{4}=0.53530771\);
\(A_{5}=-0.61232032 ; \quad A_{6}=-0.10488813 ;\)
\(A_{7}=0.68157001 ; \quad A_{8}=0.68446549\);
\(P_{p r}=\) pseudo-reduced pressure
\(T_{p r}=\) pseudo-reduced temperature
17. CNGA method
\[
\begin{equation*}
Z=\frac{1}{\left[1+\left(\frac{P_{\text {avg }} 344,400(10)^{1.785 G}}{T_{f}^{3.825}}\right)\right]} \tag{1.34}
\end{equation*}
\]
for the average gas pressure \(P_{\text {avg }}>100 \mathrm{psig}\).
For \(P_{\text {avg }}<100 \mathrm{psig}, \quad Z=1.00\)
where
\(P_{\text {avg }}=\) average gas pressure, psig
\(T_{f}=\) average gas temperature, \({ }^{\circ} \mathrm{R}\)
\(G=\) gas gravity (air =1.00)
18. Average pressure in a pipe segment
\[
\begin{equation*}
P_{\mathrm{avg}}=\frac{2}{3}\left(\frac{P_{1}^{3}-P_{2}^{3}}{P_{1}^{2}-P_{2}^{2}}\right) \tag{1.36}
\end{equation*}
\]
19. Heating value
\[
\begin{equation*}
H_{m}=\Sigma\left(y_{i} H_{i}\right) \tag{1.37}
\end{equation*}
\]
where
\(H_{m}=\) gross heating value of mixture, Btu/ft \({ }^{3}\)
\(y_{i}=\) mole fraction or percent of gas component \(i\)
\(H_{i}=\) heating value of gas component, Btu/ft \({ }^{3}\)

\section*{CHAPTER 2}
1. General Flow equation using friction factor
\[
\begin{equation*}
Q=77.54\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-P_{2}^{2}}{G T_{f} L Z f}\right)^{0.5} D^{2.5} \quad \text { (USCS units) } \tag{2.2}
\end{equation*}
\]
where
\(Q=\) gas flow rate, measured at standard conditions, \(\mathrm{ft}^{3} /\) day (SCFD)
\(f=\) friction factor, dimensionless
\(P_{b}=\) base pressure, psia
\(T_{b}=\) base temperature, \({ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)\)
\(P_{1}=\) upstream pressure, psia
\(P_{2}=\) downstream pressure, psia
\(G=\) gas gravity (air = 1.00)
\(T_{f}=\) average gas flowing temperature, \({ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)\)
\(L=\) pipe segment length, mi
\(Z=\) gas compressibility factor at the flowing temperature, dimensionless
\(D=\) pipe inside diameter, in.
\[
\begin{equation*}
Q=1.1494 \times 10^{-3}\left(\frac{T_{b}}{P_{b}}\right)\left[\frac{\left(P_{1}^{2}-P_{2}^{2}\right)}{G T_{f} L Z f}\right]^{0.5} D^{2.5} \quad \text { (SI units) } \tag{2.3}
\end{equation*}
\]
where
\(Q=\) gas flow rate, measured at standard conditions, \(\mathrm{m}^{3} /\) day
\(f=\) friction factor, dimensionless
\(P_{b}=\) base pressure, kPa
\(T_{b}=\) base temperature, \(\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)\)
\(P_{1}=\) upstream pressure, kPa
\(P_{2}=\) downstream pressure, kPa
\(G=\) gas gravity (air =1.00)
\(T_{f}=\) average gas flowing temperature, \(\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)\)
\(L=\) pipe segment length, km
\(Z=\) gas compressibility factor at the flowing temperature, dimensionless
\(D=\) pipe inside diameter, mm
2. General Flow equation using transmission factor
\[
\begin{gather*}
Q=38.77 F\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G T_{f} L_{e} Z}\right)^{0.5} D^{2.5} \quad \text { (USCS units) }  \tag{2.7}\\
Q=5.747 \times 10^{-4} F\left(\frac{T_{b}}{P_{b}}\right)\left[\frac{\left(P_{1}^{2}-e^{s} P_{2}^{2}\right)}{G T_{f} L_{e} Z}\right]^{0.5} D^{2.5} \quad \text { (SI units) } \tag{2.8}
\end{gather*}
\]
where the elevation correction is as follows:
\[
\begin{gather*}
L_{e}=\frac{L\left(e^{s}-1\right)}{s}  \tag{2.9}\\
s=0.0375 G\left(\frac{H_{2}-H_{1}}{T_{f} Z}\right) \quad \text { (USCS units) } \tag{2.10}
\end{gather*}
\]
where
\(s\) = elevation adjustment parameter, dimensionless
\(H_{1}=\) upstream elevation, ft
\(H_{2}=\) downstream elevation, ft
\(e=\) base of natural logarithms
and
\[
\begin{equation*}
s=0.0684 G\left(\frac{H_{2}-H_{1}}{T_{f} Z}\right) \quad \text { (SI units) } \tag{2.11}
\end{equation*}
\]
where
\(H_{1}=\) upstream elevation, m
\(H_{2}=\) downstream elevation, m
3. The equivalent length
\[
\begin{equation*}
L_{e}=j_{1} L_{1}+j_{2} L_{2} e^{s 1}+j_{3} L_{3} e^{s 2}+\cdots \tag{2.13}
\end{equation*}
\]
4. The gas velocity
\[
\begin{equation*}
u=0.002122\left(\frac{Q_{b}}{D^{2}}\right)\left(\frac{P_{b}}{T_{b}}\right)\left(\frac{Z T}{P}\right) \tag{2.28}
\end{equation*}
\]
where
\(u=\) gas velocity, \(\mathrm{ft} / \mathrm{s}\)
\(Q_{b}=\) gas flow rate, measured at standard conditions, \(\mathrm{ft}^{3} / \mathrm{day}\) (SCFD)
\(D=\) pipe inside diameter, in.
\(P_{b}=\) base pressure, psia
\(T_{b}=\) base temperature, \({ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)\)
\(P=\) upstream pressure, psia
\(T=\) upstream gas temperature, \({ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)\)
\(Z=\) gas compressibility factor at upstream conditions, dimensionless
In SI units, the gas velocity at any point in a gas pipeline is given by
\[
\begin{equation*}
u=14.7349\left(\frac{Q_{b}}{D^{2}}\right)\left(\frac{P_{b}}{T_{b}}\right)\left(\frac{Z T}{P}\right) \quad \text { (SI units) } \tag{2.29}
\end{equation*}
\]
where
\(u=\) gas velocity, \(\mathrm{m} / \mathrm{s}\)
\(Q_{b}=\) gas flow rate, measured at standard conditions, \(\mathrm{m}^{3} /\) day
\(D=\) pipe inside diameter, mm
\(P_{b}=\) base pressure, kPa
\(T_{b}=\) base temperature, \(\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)\)
\(P=\) pressure, kPa
\(T=\) average gas flowing temperature, \(\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)\)
\(Z=\) gas compressibility factor at the flowing temperature, dimensionless
5. Maximum velocity
\[
\begin{equation*}
u_{\max }=100 \sqrt{\frac{Z R T}{29 G P}} \quad \text { (USCS units) } \tag{2.31}
\end{equation*}
\]
where
\(Z=\) compressibility factor of gas, dimensionless
\(R=\) gas constant \(=10.73 \mathrm{ft}^{3} \mathrm{psia} / \mathrm{lb}-\) moleR
\(T=\) gas temperature, \({ }^{\circ} \mathrm{R}\)
\(G=\) gas gravity (air \(=1.00\) )
\(P=\) gas pressure, psia
6. Reynolds number
\[
\begin{equation*}
\operatorname{Re}=0.0004778\left(\frac{P_{b}}{T_{b}}\right)\left(\frac{G Q}{\mu D}\right) \quad(\mathrm{USCS} \text { units }) \tag{2.34}
\end{equation*}
\]
where
\(P_{b}=\) base pressure, psia
\(T_{b}=\) base temperature, \({ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)\)
\(G=\) specific gravity of gas (air \(=1.0\) )
\(Q=\) gas flow rate, standard \(\mathrm{ft}^{3} /\) day (SCFD)
\(D=\) pipe inside diameter, in.
\(\mu=\) viscosity of gas, lb/ft-s

In SI units, the Reynolds number is
\[
\begin{equation*}
\operatorname{Re}=0.5134\left(\frac{P_{b}}{T_{b}}\right)\left(\frac{G Q}{\mu D}\right) \quad(\text { SI units }) \tag{2.35}
\end{equation*}
\]
where
\(P_{b}=\) base pressure, kPa
\(T_{b}=\) base temperature, \({ }^{\circ} \mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)\)
\(G=\) specific gravity of gas (air \(=1.0\) )
\(Q=\) gas flow rate, \(\mathrm{m}^{3} /\) day (standard conditions)
\(D=\) pipe inside diameter, mm
\(\mu=\) viscosity of gas, Poise
7. Colebrook-White equation
\[
\begin{equation*}
\frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{e}{3.7 D}+\frac{2.51}{\operatorname{Re} \sqrt{f}}\right) \text { for } \operatorname{Re}>4000 \tag{2.39}
\end{equation*}
\]
where
\(f=\) friction factor, dimensionless
\(D=\) pipe inside diameter, in.
\(e=\) absolute pipe roughness, in.
\(\mathrm{Re}=\) Reynolds number of flow, dimensionless
\[
\begin{align*}
& \frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{2.51}{\operatorname{Re} \sqrt{f}}\right) \text { for turbulent flow in smooth pipes }  \tag{2.40}\\
& \frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{e}{3.7 D}\right) \text { for turbulent flow in fully rough pipes } \tag{2.41}
\end{align*}
\]

The transmission factor \(F\) is related to the friction factor \(f\) as follows:
\[
\begin{equation*}
F=\frac{2}{\sqrt{f}} \tag{2.42}
\end{equation*}
\]

Therefore,
\[
\begin{equation*}
f=\frac{4}{F^{2}} \tag{2.43}
\end{equation*}
\]
where
\(f=\) friction factor
\(F=\) transmission factor
8. Colebrook equation in terms of transmission factor \(F\)
\[
\begin{equation*}
F=-4 \log _{10}\left(\frac{e}{3.7 D}+\frac{1.255 F}{\operatorname{Re}}\right) \tag{2.45}
\end{equation*}
\]
9. Modified Colebrook-White equation for turbulent flow using friction factor
\[
\begin{equation*}
\frac{1}{\sqrt{f}}=-2 \log _{10}\left(\frac{e}{3.7 D}+\frac{2.825}{\operatorname{Re} \sqrt{f}}\right) \tag{2.46}
\end{equation*}
\]

Modified Colebrook-White equation in terms of the transmission factor
\[
\begin{equation*}
F=-4 \log _{10}\left(\frac{e}{3.7 D}+\frac{1.4125 F}{\operatorname{Re}}\right) \quad(\mathrm{USCS} \text { and SI units) } \tag{2.47}
\end{equation*}
\]
10. AGA equation
\[
\begin{equation*}
F=4 \log _{10}\left(\frac{3.7 D}{e}\right) \tag{2.48}
\end{equation*}
\]
11. Bend index
\[
\begin{equation*}
B I=\frac{\text { total degrees of all bends in pipe section }}{\text { total length of pipe section }} \tag{2.51}
\end{equation*}
\]
12. Weymouth equation
\[
\begin{equation*}
Q=433.5 E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G T_{f} L_{e} Z}\right)^{0.5} D^{2.667} \quad \text { (in USCS units) } \tag{2.52}
\end{equation*}
\]
where
\(Q=\) volume flow rate, standard \(\mathrm{ft}^{3} /\) day (SCFD)
\(E=\) pipeline efficiency, a decimal value less than or equal to 1.0
\(P_{b}=\) base pressure, psia
\(T_{b}=\) base temperature, \({ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)\)
\(P_{1}=\) upstream pressure, psia
\(P_{2}=\) downstream pressure, psia
\(G=\) gas gravity (air =1.00)
\(T_{f}=\) average gas flow temperature, \({ }^{\circ} \mathrm{R}\left(460+{ }^{\circ} \mathrm{F}\right)\)
\(L_{e}=\) equivalent length of pipe segment, mi
\(Z=\) gas compressibility factor, dimensionless
\(D=\) pipe inside diameter, in.
Weymouth transmission factor
\[
\begin{equation*}
F=11.18(\mathrm{D})^{1 / 6} \quad \text { (in USCS units) } \tag{2.53}
\end{equation*}
\]

Weymouth equation
\[
\begin{equation*}
Q=3.7435 \times 10^{-3} E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G T_{f} L_{e} Z}\right)^{0.5} D^{2.667} \quad \text { (in SI units) } \tag{2.54}
\end{equation*}
\]
where
\(Q=\) gas flow rate, standard m \(3 /\) day
\(T_{b}=\) base temperature, \(\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)\)
\(P_{b}=\) base pressure, kPa
\(T_{f}=\) average gas flow temperature, \(\mathrm{K}\left(273+{ }^{\circ} \mathrm{C}\right)\)
\(P_{1}=\) upstream pressure, kPa
\(P_{2}=\) downstream pressure, kPa
\(L_{e}=\) equivalent length of pipe segment, km
Other symbols are as defined previously.
Weymouth transmission factor
\[
\begin{equation*}
F=6.521(D)^{1 / 6} \quad \text { (in SI units) } \tag{2.53a}
\end{equation*}
\]
13. Panhandle A equation
\[
\begin{align*}
& Q=435.87 E\left(\frac{T_{b}}{P_{b}}\right)^{1.0788}\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G^{0.8539} T_{f} L_{e} Z}\right)^{0.5394} D^{2.6182} \quad \text { (USCS units) }  \tag{2.55}\\
& Q=4.5965 \times 10^{-3} E\left(\frac{T_{b}}{P_{b}}\right)^{1.0788}\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G^{0.8539} T_{f} L_{e} Z}\right)^{0.5394} D^{2.6182} \quad \text { (SI units) } \tag{2.56}
\end{align*}
\]

The equivalent transmission factor for Panhandle A equation
\[
\begin{equation*}
F=7.2111 E\left(\frac{Q G}{D}\right)^{0.07305} \quad(\mathrm{USCS}) \tag{2.57}
\end{equation*}
\]
and in SI units, it is
\[
\begin{equation*}
F=11.85 E\left(\frac{Q G}{D}\right)^{0.07305} \tag{2.58}
\end{equation*}
\]
13. Panhandle B equation
\[
\begin{align*}
& Q=737 E\left(\frac{T_{b}}{P_{b}}\right)^{1.02}\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G^{0.961} T_{f} L_{e} Z}\right)^{0.51} D^{2.53} \quad \text { (USCS units) }  \tag{2.59}\\
& Q=1.002 \times 10^{-2} E\left(\frac{T_{b}}{P_{b}}\right)^{1.02}\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G^{0.961} T_{f} L_{e} Z}\right)^{0.51} D^{2.53} \quad \text { (SI units) } \tag{2.60}
\end{align*}
\]

The equivalent transmission factor for Panhandle B equation is
\[
\begin{align*}
F & =16.7 E\left(\frac{Q G}{D}\right)^{0.01961} \quad \text { (USCS units) }  \tag{2.61}\\
F & =19.08 E\left(\frac{Q G}{D}\right)^{0.01961} \quad \text { (SI units) } \tag{2.62}
\end{align*}
\]
14. IGT equation
\[
\begin{align*}
& Q=136.9 E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G^{0.8} T_{f} L_{e} \mu^{0.2}}\right)^{0.555} D^{2.667} \quad \text { (USCS units) }  \tag{2.63}\\
& Q=1.2822 \times 10^{-3} E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G^{0.8} T_{f} L_{e} \mu^{0.2}}\right)^{0.555} D^{2.667} \quad \text { (SI units) } \tag{2.64}
\end{align*}
\]
15. Spitzglass equation

The low pressure (less than or equal to 1 psig ) version
\[
\begin{equation*}
Q=3.839 \times 10^{3} E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}-P_{2}}{G T_{f} L_{e} Z\left(1+\frac{3.6}{D}+0.03 D\right)}\right)^{0.5} D^{2.5} \quad \text { (USCS units) } \tag{2.65}
\end{equation*}
\]

The high pressure (more than 1 psig ) version
\[
\begin{equation*}
Q=729.6087 E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G T_{f} L_{e} Z\left(1+\frac{3.6}{D}+0.03 D\right)}\right)^{0.5} D^{2.5} \quad \text { (USCS units) } \tag{2.67}
\end{equation*}
\]

The low pressure (less than 6.9 kPa ) version
\[
\begin{equation*}
Q=5.69 \times 10^{-2} E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}-P_{2}}{G T_{f} L_{e} Z\left(1+\frac{91.44}{D}+0.0012 D\right)}\right)^{0.5} D^{2.5} \quad \text { (SI units) } \tag{2.66}
\end{equation*}
\]

The high pressure (more than 6.9 kPa ) version
\[
\begin{equation*}
Q=1.0815 \times 10^{-2} E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G T_{f} L_{e} Z\left(1+\frac{91.44}{D}+0.0012 D\right)}\right)^{0.5} D^{2.5} \quad \text { (SI units) } \tag{2.68}
\end{equation*}
\]
16. The Mueller equation
\[
\begin{align*}
& Q=85.7368 E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G^{0.7391} T_{f} L_{e} \mu^{0.2609}}\right)^{0.575} D^{2.725} \quad \text { (USCS units) }  \tag{2.69}\\
& Q=3.0398 \times 10^{-2} E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G^{0.7391} T_{f} L_{e} \mu^{0.2609}}\right)^{0.575} D^{2.725} \quad \text { (SI units) } \tag{2.70}
\end{align*}
\]
17. Fritzsche formula
\[
\begin{align*}
& Q=410.1688 E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-P_{2}^{2}}{G^{0.8587} T_{f} L_{e}}\right)^{0.538} D^{2.69} \quad \text { (USCS units) }  \tag{2.71}\\
& Q=2.827 E\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{1}^{2}-e^{s} P_{2}^{2}}{G^{0.8587} T_{f} L_{e}}\right)^{0.538} D^{2.69} \quad \text { (SI units) } \tag{2.72}
\end{align*}
\]

\section*{CHAPTER 3}
1. Total equivalent length-series piping
\[
\begin{equation*}
L e=L_{1}+L_{2}\left(\frac{D_{1}}{D_{2}}\right)^{5}+L_{3}\left(\frac{D_{1}}{D_{3}}\right)^{5} \tag{3.6}
\end{equation*}
\]
2. Equivalent diameter-parallel pipes
\[
\begin{equation*}
D_{e}=D_{1}\left[\left(\frac{1+\text { Const } 1}{\text { Const } 1}\right)^{2}\right]^{1 / 5} \tag{3.17}
\end{equation*}
\]
where
\[
\begin{equation*}
\text { Const } 1=\sqrt{\left(\frac{D_{1}}{D_{2}}\right)^{5}\left(\frac{L_{2}}{L_{1}}\right)} \tag{3.18}
\end{equation*}
\]

Flow rates \(Q_{1}\) and \(Q_{2}\) are calculated from
\[
\begin{equation*}
Q_{1}=\frac{Q \text { Const } 1}{1+\text { Const } 1} \tag{3.19}
\end{equation*}
\]
and
\[
\begin{equation*}
Q_{2}=\frac{Q}{1+\text { Const } 1} \tag{3.20}
\end{equation*}
\]
3. Temperature profile of gas in a pipe segment
\[
\begin{gather*}
T_{2}=T_{s}+\left(T_{1}-T_{s}\right) e^{-\theta}  \tag{3.29}\\
\theta=\frac{\pi U D \Delta L}{m C p} \tag{3.28}
\end{gather*}
\]
where
\(U=\) overall heat transfer coefficient, Btu/h/ft \({ }^{2} /{ }^{\circ} \mathrm{F}\)
\(\Delta L=\) length of pipe segment
\(\Delta A=\) surface area of pipe for heat transfer \(=\pi D \Delta L\)
\(T_{1}=\) gas temperature upstream of pipe segment, \({ }^{\circ} \mathrm{F}\)
\(T_{2}=\) gas temperature downstream of pipe segment, \({ }^{\circ} \mathrm{F}\)
\(T_{s}=\) average soil temperature surrounding pipe segment, \({ }^{\circ} \mathrm{F}\)
\(D=\) pipe inside diameter, ft
\(m=\) mass flow rate of gas, \(\mathrm{lb} / \mathrm{s}\)
4. Line pack
\[
\begin{equation*}
V_{b}=28.798\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{\text {avg }}}{Z_{\text {avg }} T_{\text {avg }}}\right)\left(D^{2} L\right) \quad \text { (USCS units) } \tag{3.34}
\end{equation*}
\]
where
\(V_{b}=\) line pack in pipe segment, standard \(\mathrm{ft}^{3}\)
\(D=\) pipe inside diameter, in.
\(L=\) pipe segment length, mi
\[
\begin{equation*}
V_{b}=7.855 \times 10^{-4}\left(\frac{T_{b}}{P_{b}}\right)\left(\frac{P_{\text {avg }}}{Z_{\text {avg }} T_{\text {avg }}}\right)\left(D^{2} L\right) \quad \text { (SI units) } \tag{3.35}
\end{equation*}
\]
where
\(V_{b}=\) line pack in pipe segment, standard \(\mathrm{m}^{3}\)
\(D=\) pipe inside diameter, mm
\(L=\) pipe segment length, km

\section*{CHAPTER 4}
1. Compression ratio
\[
\begin{equation*}
r=\frac{P_{d}}{P_{s}} \tag{4.1}
\end{equation*}
\]
2. Isothermal work done
\[
\begin{equation*}
W i=\frac{53.28}{G} T_{1} \log _{e}\left(\frac{P_{2}}{P_{1}}\right) \quad \text { (USCS units) } \tag{4.4}
\end{equation*}
\]
where
\(W i=\) isothermal work done, \(\mathrm{ft}-\mathrm{lb} / \mathrm{lb}\) of gas
\(G \quad=\) gas gravity, dimensionless
\(T_{1}=\) suction temperature of gas, \({ }^{\circ} \mathrm{R}\)
\(P_{1} \quad=\) suction pressure of gas, psia
\(P_{2}=\) discharge pressure of gas, psia
\(\log _{e}=\) natural logarithm to base \(e(e=2.718)\)
The ratio \(\left(\frac{P_{2}}{P_{1}}\right)\) is also called the compression ratio.
\[
\begin{equation*}
W i=\frac{286.76}{G} T_{1} \log _{e}\left(\frac{P_{2}}{P_{1}}\right) \quad \text { (SI units) } \tag{4.5}
\end{equation*}
\]
where
\(W i=\) isothermal work done, \(\mathrm{J} / \mathrm{kg}\) of gas
\(T_{1}=\) suction temperature of gas, K
\(P_{1}=\) suction pressure of gas, kPa absolute
\(P_{2}=\) discharge pressure of gas, kPa absolute
Other symbols are as defined earlier.
3. Adiabatic work done
\[
\begin{equation*}
W a=\frac{53.28}{G} T_{1}\left(\frac{\gamma}{\gamma-1}\right)\left[\left(\frac{P_{2}}{P_{1}}\right)^{\frac{\gamma-1}{\gamma}}-1\right] \quad \text { (USCS units) } \tag{4.8}
\end{equation*}
\]
where
\(W a=\) adiabatic work done, ft-lb/lb of gas
\(G=\) gas gravity, dimensionless
\(T_{1}=\) suction temperature of gas, \({ }^{\circ} \mathrm{R}\)
\(\gamma=\) ratio of specific heats of gas, dimensionless
\(P_{1}=\) suction pressure of gas, psia
\(P_{2}=\) discharge pressure of gas, psia
\[
\begin{equation*}
W a=\frac{286.76}{G} T_{1}\left(\frac{\gamma}{\gamma-1}\right)\left[\left(\frac{P_{2}}{P_{1}}\right)^{\frac{\gamma-1}{\gamma}}-1\right] \quad \text { (SI units) } \tag{4.9}
\end{equation*}
\]
where
\(W a=\) adiabatic work done, \(\mathrm{J} / \mathrm{kg}\) of gas
\(T_{1}=\) suction temperature of gas, K
\(P_{1}=\) suction pressure of gas, kPa absolute
\(P_{2}=\) discharge pressure of gas, kPa absolute

Other symbols are as defined earlier.
4. Horsepower
\[
\begin{equation*}
H P=0.0857\left(\frac{\gamma}{\gamma-1}\right) Q T_{1}\left(\frac{Z_{1}+Z_{2}}{2}\right)\left(\frac{1}{\eta_{a}}\right)\left[\left(\frac{P_{2}}{P_{1}}\right)^{\frac{\gamma-1}{\gamma}}-1\right] \tag{4.15}
\end{equation*}
\]
where
\(H P=\) compression horsepower
\(\gamma=\) ratio of specific heats of gas, dimensionless
\(Q=\) gas flow rate, MMSCFD
\(T_{1}=\) suction temperature of gas, \({ }^{\circ} \mathrm{R}\)
\(P_{1}=\) 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
\(\eta_{a}=\) compressor adiabatic (isentropic) efficiency, decimal value

In SI units, the Power equation is as follows:
\[
\begin{equation*}
\text { Power }=4.0639\left(\frac{\gamma}{\gamma-1}\right) Q T_{1}\left(\frac{Z_{1}+Z_{2}}{2}\right)\left(\frac{1}{\eta_{a}}\right)\left[\left(\frac{P_{2}}{P_{1}}\right)^{\frac{\gamma-1}{\gamma}}-1\right] \tag{4.16}
\end{equation*}
\]
where
Power \(=\) compression power, kW
\(\gamma \quad=\) ratio of specific heats of gas, dimensionless
\(Q \quad=\) gas flow rate, \(\mathrm{Mm}^{3} /\) day (standard)
\(T_{1} \quad=\) suction temperature of gas, K
\(P_{1} \quad=\) suction pressure of gas, kPa
\(P_{2}=\) discharge pressure of gas, kPa
\(Z_{1} \quad=\) compressibility of gas at suction conditions, dimensionless
\(Z_{2} \quad=\) compressibility of gas at discharge conditions, dimensionless
\(\eta_{a} \quad=\) compressor adiabatic (isentropic) efficiency, decimal value
\[
\begin{equation*}
B H P=\frac{H P}{\eta_{m}} \tag{4.17}
\end{equation*}
\]
5. Compression ratio
\[
\begin{equation*}
r=\left(r_{t}\right)^{\frac{1}{n}} \tag{4.25}
\end{equation*}
\]
where
\(r=\) compression ratio, dimensionless
\(r_{t}=\) overall compression ratio, dimensionless
\(n=\) number of compressors in series

\section*{CHAPTER 6}
1. Barlow's equation
\[
\begin{equation*}
S_{h}=\frac{P D}{2 t} \tag{6.1}
\end{equation*}
\]
where
\(S_{h}=\) hoop or circumferential stress in pipe material, psi
\(P=\) internal pressure, psi
\(D=\) pipe outside diameter, in.
\(t=\) pipe wall thickness, in.
Axial or longitudinal stress
\[
\begin{equation*}
S_{a}=\frac{P D}{4 t} \tag{6.2}
\end{equation*}
\]
2. Internal design pressure
\[
\begin{equation*}
P=\frac{2 t S E F T}{D} \tag{6.8}
\end{equation*}
\]
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
\(F=\) design factor, usually 0.72 for cross-country gas pipelines, but can be as low as 0.4 , depending on class location and type of construction
\(T=\) temperature deration factor \(=1.00\) for temperatures below \(250^{\circ} \mathrm{F}\)
3. Blowdown calculations
\[
\begin{equation*}
T=\frac{0.0588 P_{1}^{\frac{1}{3}} G^{\frac{1}{2}} D^{2} L F_{c}}{d^{2}} \quad \text { (USCS units) } \tag{6.9}
\end{equation*}
\]
where
\(T\) = blowdown time, min
\(P_{1}=\) initial pressure, psia
\(G=\) gas gravity (air = 1.00)
\(D=\) pipe inside diameter, in.
\(L=\) length of pipe section, mi
\(d\) = inside diameter of blowdown pipe, in.
\(F_{c}=\) choke factor (as follows)

Choke factor list
Ideal nozzle \(=1.0\)
Through gate \(=1.6\)
Regular gate \(=1.8\)
Regular lube plug \(=2.0\)
Venturi lube plug \(=3.2\)
In SI units,
\[
\begin{equation*}
T=\frac{0.0192 P_{1}^{\frac{1}{3}} G^{\frac{1}{2}} D^{2} L F_{c}}{d^{2}} \quad \text { (SI units) } \tag{6.10}
\end{equation*}
\]
where
\(P_{1}=\) initial pressure, kPa
\(D=\) pipe inside diameter, mm
\(L=\) length of pipe section, km
\(d\) = pipe inside diameter of blowdown, mm
Other symbols are as defined before.
4. Pipe weight
\[
\begin{equation*}
w=10.68 \times t \times(D-t) \quad(\text { USCS units }) \tag{6.11}
\end{equation*}
\]
where
\(w=\) pipe weight, \(\mathrm{lb} / \mathrm{ft}\)
\(D=\) pipe outside diameter, in.
\(t=\) pipe wall thickness, in.
\[
\begin{equation*}
w=0.0246 \times t \times(D-t) \quad \text { (SI units) } \tag{6.12}
\end{equation*}
\]
where
\(w=\) pipe weight, \(\mathrm{kg} / \mathrm{m}\)
\(D=\) pipe outside diameter, mm
\(t\) = pipe wall thickness, mm

\section*{CHAPTER 9}
1. The discharge through the orifice meter
\[
\begin{equation*}
Q=C_{c} C_{v} A_{o} \sqrt{\frac{2\left[\left(p_{1}-p_{2}\right) / \rho+g\left(z_{1}-z_{2}\right)\right]}{1-C_{c}^{2}\left(A_{o} / A\right)^{2}}} \tag{9.2}
\end{equation*}
\]
where
\(Q=\) flow rate, \(\mathrm{ft}^{3} / \mathrm{s}\)
\(C_{c}=\) contraction coefficient, dimensionless
\(C_{v}=\) discharge coefficient, dimensionless
\(A_{o}=\) cross-sectional area of the orifice, in. \({ }^{2}\)
\(A=\) cross-sectional area of pipe containing the orifice, in. \({ }^{2}\)
\(p_{1}=\) upstream pressure, psig
\(p_{2}=\) downstream pressure, psig
\(\rho=\) density of gas, \(\mathrm{lb} / \mathrm{ft}^{3}\)
\(z_{1}=\) upstream elevation, ft
\(z_{2}=\) downstream elevation, ft
\(g=\) acceleration due to gravity
2. Sharp-crested orifice
\[
\begin{equation*}
C_{c}=0.595+0.29\left(\frac{A_{o}}{A}\right)^{\frac{5}{2}} \tag{9.4}
\end{equation*}
\]
3. The fundamental orifice meter flow equation described in the ANSI 2530/AGA Report No. 3 is as follows:
\[
\begin{equation*}
q_{m}=\frac{C}{\left(1-\beta^{4}\right)^{0.5}} Y \frac{\pi}{4} d^{2}\left(2 g \rho_{f} \Delta P\right)^{0.5} \tag{9.6}
\end{equation*}
\]
or
\[
\begin{gather*}
q_{m}=K Y \frac{\pi}{4} d^{2}\left(2 g \rho_{f} \Delta P\right)^{0.5}  \tag{9.7}\\
\beta=\frac{d}{D}  \tag{9.8}\\
K=\frac{C}{\left(1-\beta^{4}\right)^{0.5}}=\frac{C D^{2}}{\left(D^{4}-d^{4}\right)^{0.5}} \tag{9.9}
\end{gather*}
\]
where
\(q_{m}=\) mass flow rate of gas, \(\mathrm{lb} / \mathrm{s}\)
\(\rho_{f}=\) density of gas, \(\mathrm{lb} / \mathrm{ft}^{3}\)
\(C=\) discharge coefficient
\(\beta=\) beta ratio, dimensionless
\(d=\) orifice diameter, in.
\(D=\) meter tube diameter, in.
\(Y=\) expansion factor, dimensionless
\(g=\) acceleration due to gravity, \(\mathrm{ft} / \mathrm{s}^{2}\)
\(\Delta P=\) pressure drop across the orifice, psi
\(K=\) flow coefficient, dimensionless
4. Buckingham and Bean equation endorsed by the National Bureau of Standards (NBS) and listed in AGA Report No. 3

For flange taps:
\[
\begin{align*}
K_{e}= & 0.5993+\frac{0.007}{D}+\left(0.364+\frac{0.076}{D^{0.5}}\right) \beta^{4}+0.4\left(1.6-\frac{1}{D}\right)^{5}\left[\left(0.07+\frac{0.5}{D}\right)-\beta\right]^{2.5} \\
& -\left(0.009+\frac{0.034}{D}\right)(0.5-\beta)^{1.5}+\left(\frac{65}{D^{2}}+3\right)(\beta-0.7)^{2.5} \tag{9.12}
\end{align*}
\]
where
\(K_{e}=\) flow coefficient for Reynolds number \(R_{d}=d\left(10^{6} / 15\right)\), dimensionless
\(D=\) meter tube diameter, in.
\(d=\) orifice diameter, in.
\(\beta=\) beta ratio, dimensionless

For pipe taps:
\[
\begin{align*}
K_{e}= & 0.5925+\frac{0.0182}{D}+\left(0.440-\frac{0.06}{D}\right) \beta^{2}+\left(0.935+\frac{0.225}{D}\right) \beta^{5} \\
& +1.35 \beta^{14}+\frac{1.43}{D^{0.5}}(0.25-\beta)^{2.5} \tag{9.13}
\end{align*}
\]
where all symbols are as defined before.
5. Expansion factor

For flange taps:
\[
\begin{equation*}
Y_{1}=1-\left(0.41+0.35 \beta^{4}\right) \frac{x_{1}}{k} \tag{9.20}
\end{equation*}
\]

For pipe taps:
\[
\begin{equation*}
Y_{1}=1-\left[0.333+1.145\left(\beta^{2}+0.7 \beta^{5}+12 \beta^{13}\right)\right] \frac{x_{1}}{k} \tag{9.21}
\end{equation*}
\]
and the pressure ratio \(x_{1}\) is
\[
\begin{equation*}
x_{1}=\frac{P_{f 1}-P_{f 2}}{P_{f 1}}=\frac{h_{w}}{27.707 P_{f 1}} \tag{9.22}
\end{equation*}
\]
where
\(Y_{1}=\) expansion factor based on upstream pressure
\(x_{1}=\) ratio of differential pressure to absolute upstream static pressure
\(h_{w}=\) differential pressure between upstream and downstream taps in in. of water at \(60^{\circ} \mathrm{F}\)
\(P_{f 1}=\) static pressure at upstream tap, psia
\(P_{f 2}=\) static pressure at downstream tap, psia
\(x_{1} / k=\) acoustic ratio, dimensionless
\(k=\) ratio of specific heats of gas, dimensionless
6. Supercompressibility factor
\[
\begin{equation*}
C=F_{b} F_{r} F_{p b} F_{t b} F_{t f} F_{g r} F_{p v} Y \tag{9.34}
\end{equation*}
\]
where the dimensionless factors are
\(F_{b}=\) basic orifice factor
\(F_{r}=\) Reynolds number factor
\(F_{p b}=\) pressure base factor
\(F_{t b}=\) temperature base factor
\(F_{t f}=\) flowing temperature factor
\(F_{g r}=\) gas relative density factor
\(F_{p v}=\) supercompressibility factor
\(Y=\) expansion factor

\section*{CHAPTER 10}
1. Present value
\[
\begin{equation*}
P V=\frac{R}{i}\left(1-\frac{1}{(1+i)^{n}}\right) \tag{10.1}
\end{equation*}
\]
where
\(P V=\) present value, \(\$\)
\(R=\) series of cash flows, \(\$\)
\(i=\) interest rate, decimal value
\(n\) = number of periods, years
2. Pipe material cost
\[
\begin{equation*}
P M C=\frac{10.68(D-T) T L C \times 5280}{2000} \tag{10.2}
\end{equation*}
\]
where
\(P M C=\) pipe material cost, \(\$\)
\(L \quad=\) length of pipe, mi
\(D \quad=\) pipe outside diameter, in.
\(T\) = pipe wall thickness, in.
\(C=\) pipe material cost, \$/ton

In SI units,
\[
\begin{equation*}
P M C=0.0246(D-T) T L C \tag{10.3}
\end{equation*}
\]
where
\(P M C=\) pipe material cost, \(\$\)
\(L \quad=\) length of pipe, km
\(D \quad=\) pipe outside diameter, mm
\(T\) = pipe wall thickness, mm
\(C=\) pipe material cost, \$/metric ton```


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