

OPERATOR'S GUIDE TO PROCESS COMPRESSORS



ROBERT X. PEREZ

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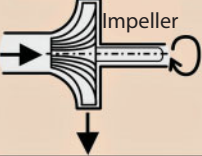
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Operator's Guide to Process Compressors

Frontispiece



Compressor type	Capacity type		Centrifugal type
	Screw	Reciprocating	Turbo
Compression principle	Compresses by rotating screw rotor  M rotor F rotor	Compresses through the reciprocating motion of the piston  Piston	Compresses through centrifugal force using impeller rotation  Impeller

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Robert X. Perez



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Dedication

I would like to dedicate this book to process operators around the world. In my view, they never get the recognition they deserve for their long hours and dedication to their critical machinery. Without them, safe and profitable unit operations would not be possible.

Acknowledgements

I would like to thank my wife Elaine for carefully reading and editing the various versions of my manuscript. I also want to acknowledge David Lawhon for performing a thorough technical review of my manuscript and providing valuable feedback that enabled me to improve the book's overall content and readability. Special thanks go out to Julien LeBleu for providing content for "The Importance of Lubrication" and "Inspection Technique Available to Operators and Field Personnel" book chapters.

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Preface

Gas compressors are installed in most large processing facilities. They are designed to transport gases between different locations in processing units by compressing them from a lower pressure to a higher one using some type of driver, such as an electric motor, a steam turbine, a gas turbine, etc. The process of gas compression requires that compressors be designed to handle high gas pressures, high operating temperatures, high rotational speeds, and the high component stress levels. Through their gradual technological evolution, compressors have become highly reliable and safe machines, when properly maintained and operated as intended by their designers.

Gas compressors tend to be the largest, most costly, and most critical machines employed in chemical and gas transfer processes. The most common types of compressors are centrifugal compressors, reciprocating compressors, and screw compressors. Since they tend to have the greatest effect on the reliability of processes they power, compressors typically receive the most scrutiny of all the machinery among the general population of processing equipment.

Today, there appears to be a need in industry for a review of the best in class operating methods and procedures for compressors. As the previous generation of operators retire or move on, much of the knowledge that was gained over the past years has been forgotten or lost. The attrition of experience we have all experienced in recent years has resulted in the recurrence of reliability problems that have already been solved. To prevent

unwanted compressor failures from occurring, operators must be taught how the equipment should operate and how each is different from one another.

The ultimate purpose of this book is to teach those who work in process settings more about gas compressors, so they can start them up and operate them correctly and monitor them with more confidence. Some may regard compressor technology as too broad and complex a topic for operating personnel to fully understand, but I have tried to address this concern by distilling this vast body of knowledge into some key, easy to understand lessons for the reader to study at his or her own pace. My hope is that learning more about how compressors work and the factors that are key to their reliability, compressor operators can keep them running longer and more reliably.

The main goals of this book are to:

- Explain important theories and concepts about gases and compression processes with a minimum of mathematics
- Identify key compressor components and explain how they affect reliability
- Explain how the different types of compressors function
- Explain key operating factors that affect reliability
- Introduce the reader to basic troubleshooting methodologies
- Introduce operators to proven field inspection techniques

I hope that readers find this book useful as they progress through their careers. I recommend that occasionally readers review the book's content to refresh their knowledge of compressors.

Always keep learning and questioning your assumptions and paradigms. I think the following quotation explains why it's important to change your point of view from time to time:

*"Your assumptions are your windows on the world.
Scrub them off every once in a while, or the light won't
come in."*

— Isaac Asimov

Robert X. Perez
Spring, 2019

1

Introduction to Gases

Gases represent a state of matter that has no fixed shape or fixed volume, which consist of tiny, energetic particles, i.e., atoms or molecules, that are widely spaced (Figure 1.1). Compared to the other states of matter, solids and liquids, gases have a much lower density, i.e., they have a small mass per unit volume, because there is a great deal of empty space between gas particles. At room temperature and pressure, the gas inside a container occupies only 0.1% of the total container volume. The other 99.9% of the total volume is empty space (whereas in liquids and solids, about 70% of the volume is occupied by particles). Gas particles move very fast and collide with one another, causing them to diffuse, or spread out, until they are evenly distributed throughout the volume of their container. You will never see only half of a balloon filled with air.

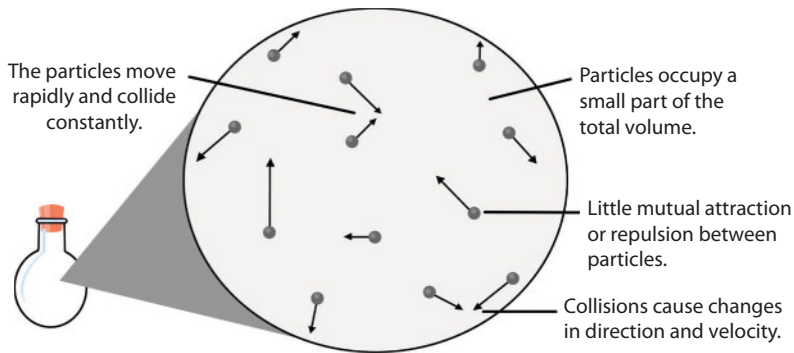


Figure 1.1 Gas atoms or molecules are constantly moving and colliding with one another.

Although both liquids and gases take the shape of their containers, gases differ from liquids in that there is so much space between gas molecules that they offer little resistance to motion and can be compressed to smaller and smaller volumes. As seen in Figure 1.2, as a gas is compressed, the molecules making up the gas get closer together and create a higher internal pressure.

Hydrogen is the lightest known gas. Any balloon filled with hydrogen gas will float in air if the total mass of its container is not too great. Helium gas is also lighter than air and has 92% of the lifting power of hydrogen. Today all airships, i.e., blimps, use helium instead of hydrogen because it offers almost the same lifting power and is not flammable.

Gases can be monatomic, diatomic, and polyatomic. Monatomic gases are gases composed of single atoms, diatomic gases are those composed of two atom molecules, and polyatomic gases are those made up of molecules with more than two atoms. Noble gases such as helium, neon, argon, etc., are normally found as single atoms, since they are chemically inert. Gases such as nitrogen (N_2), oxygen (O_2), and carbon monoxide (CO) tend to be found as diatomic molecules (Figure 1.3). Carbon dioxide (CO_2), and methane (CH_4) are examples of polyatomic gas molecules (Figure 1.3).

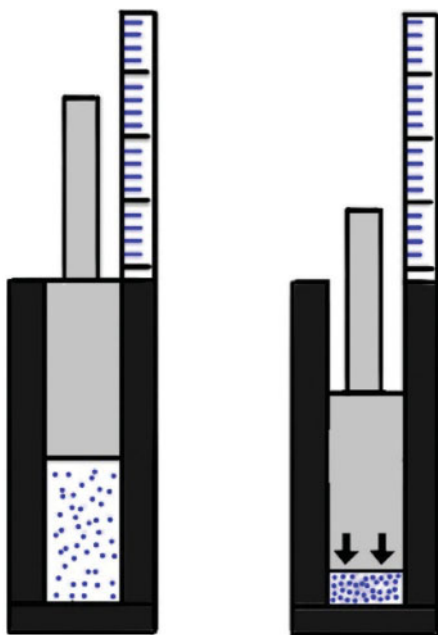


Figure 1.2 As gas is compressed, the gas molecules get closer together.

Gases can be found all around us. In fact, the earth's atmosphere is a blanket of gases composed of nitrogen (78%), oxygen (21%), argon (1%), and then trace amounts of carbon dioxide, neon, helium, methane, krypton, hydrogen, nitrous oxide, xenon, ozone, iodine, carbon monoxide, and ammonia.

Because of the large distances between gas particles, the attractions or repulsions among them are weak. The particles in a gas are in rapid and continuous motion. For example, the average velocity of nitrogen molecules, N_2 , at 68 °F is about 1640 ft/s. As the temperature of a gas increases, the particles' velocity increases. The average velocity of nitrogen molecules at 212 °F is about 1886 ft/s. The particles in a gas are constantly colliding with the walls of the container and with each other. Because of these collisions, the gas particles are constantly changing their direction of motion and their velocity. In a typical situation, a gas particle moves a very short distance between collisions. For example, oxygen, O_2 , molecules at normal temperatures

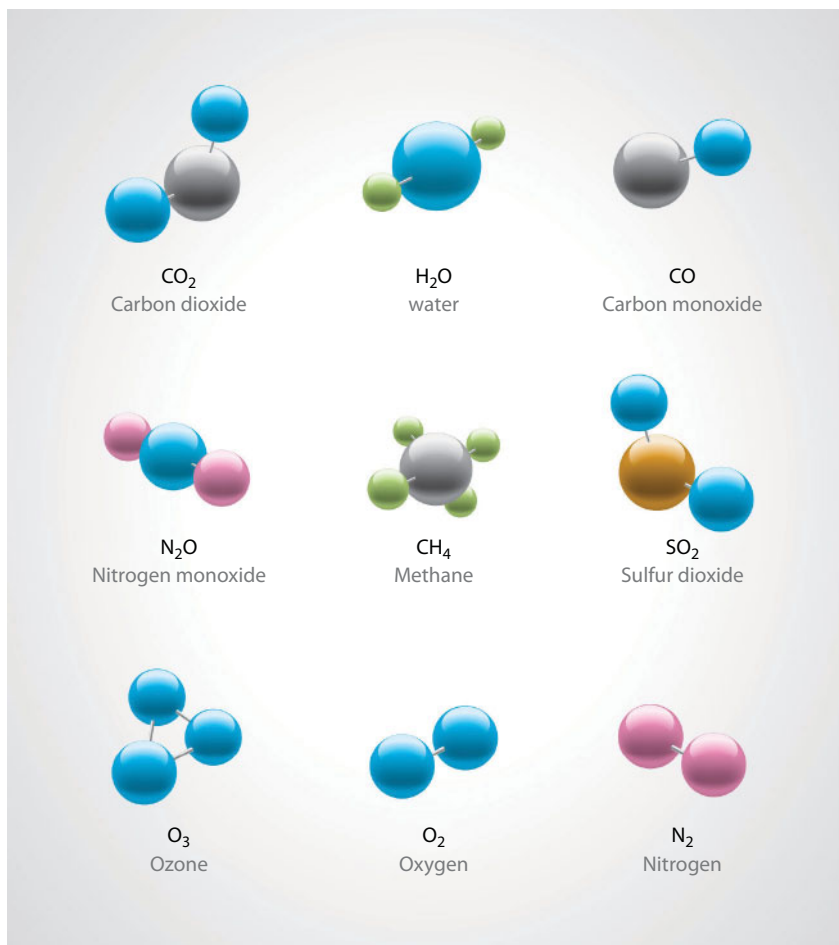


Figure 1.3 Oxygen, nitrogen, and carbon monoxide are examples of diatomic molecules. Carbon dioxide, water, nitrogen monoxide, methane, sulfur dioxide, and ozone are examples of polyatomic molecules.

and pressures move an average of 0.000003937 inches between collisions.

1.1 Ideal Gases

Scientists often simplify the model of gases by imagining the behavior of an ideal gas. An ideal gas differs from a real gas in

that the particles are assumed to be point masses, that is, particles that have a mass but occupy no volume. It is also assumed that there are no attractive or repulsive forces at all between the particles. When all these assumptions are incorporated into a gas model, the “ideal gas model” is obtained. As the name implies, the ideal gas model describes an “ideal” of gas behavior that is only approximated by reality. Nevertheless, the model has been proven to reasonably explain and predict the behavior of typical gases under typical conditions.

Note: Under ordinary conditions, the properties of gases predicted by the ideal gas law are within 5% of their actual values.

1.2 Properties of Gases

The ideal gas model is used to predict changes in four related gas properties: volume, number of particles, temperature, and pressure. Volumes of gases are usually described in cubic feet, ft^3 , or cubic meters, m^3 , and numbers of particles are usually described in moles.

1.3 Temperature

Temperature is a physical quantity expressing how hot or cold a system of atoms or physical object is. Technically, temperature is the proportional measure of the average kinetic energy related to the random motions of the constituent particles of matter in a system. Temperature is an important property of a system because it is an indication of the direction in which heat energy will spontaneously flow. Remember that heat energy always flows from a hotter body (one at a higher temperature) to a colder body (one at a lower temperature).

Temperature is a measure of the total heat energy in a system.

Gas temperatures can be measured with thermometers, infrared guns, and thermocouples. Readings can be reported

in degrees Fahrenheit, °F, or Celsius, °C. However, engineers generally use Rankine, or Kelvin temperatures for calculations.

1.4 Pressure

Remember that gases have no definite shape or volume; they tend to fill whatever container they are in. They can compress and expand and have extremely low densities when compared to a liquid or solid. Combinations of gases tend to mix together spontaneously; that is, they form gas mixtures. Air, for example, is a solution of mostly nitrogen and oxygen. Any understanding of the properties of gases must be able to explain the properties of gas mixture.

The kinetic theory of gases indicates that gas particles are always in motion and are colliding with other particles and the walls of the container holding them. Although collisions with container walls are elastic (i.e., there is no net energy gain or loss because of the collision), a gas particle does exert a force on the wall during the collision. Each time a gas particle collides with and ricochets off one of the walls of its container, it exerts a tiny force against the wall. The accumulation of all these forces distributed over the area of the walls of the container causes something we call pressure. Pressure (P) is defined as the force of all the gas particle-wall collisions divided by the area of the wall:

$$\text{Pressure} = \text{force/area}$$

In English units, pressure is measured in psi, or pounds per square in. The formal, SI-approved unit of pressure is the pascal (Pa), which is defined as 1 N/m² (one newton of force over an area of one square meter). However, this is usually too small in magnitude to be useful. A common unit of pressure is the atmosphere (atm), which was originally defined as the average atmospheric pressure at sea level.

1.5 Gas Laws

When seventeenth-century scientists began studying the physical properties of gases, they noticed simple relationships between some of the measurable properties of gases. For example, scientists noted that for a given quantity of gas, usually expressed in units of moles, i.e., number of molecules $[n]$ in a system, if the temperature (T) of the gas is kept constant, pressure and volume are related: as one variable increases, the other variable decreases. Conversely, as one variable decreases, the other variable increases. Therefore, we say that *pressure and volume are inversely related*.

take pressure (P) and volume (V), for example:

There is more to it, however: pressure and volume of a given amount of gas at a constant temperature are numerically related. If you take the pressure value and multiply it by the volume value, the product is a constant for a given amount of gas at a constant temperature:

$$P \times V = \text{constant at constant } n(\text{moles}) \text{ and } T(\text{temperature}) \quad (1.1)$$

If either volume or pressure changes while the amount and temperature stays the same, then the other property must change so that the product of the two properties still equals that same constant. That is, if the original conditions are labelled P_1 and V_1 and the new conditions are labelled P_2 and V_2 , we have

$$P_1 V_1 = \text{constant} = P_2 V_2 \quad (1.2)$$

where the properties are assumed to be multiplied together. Leaving out the middle part, we have simply:

$$P_1 V_1 = P_2 V_2 \text{ at constant } n \text{ and } T \quad (1.3)$$

This equation is an example of a gas law. A gas law is a simple mathematical formula that allows you to model, or predict, the behavior of a gas. This particular gas law is called *Boyle's Law*, after the English scientist Robert Boyle, who first announced it in 1662. Figure 1.4 shows two representations of what Boyle's Law describes.

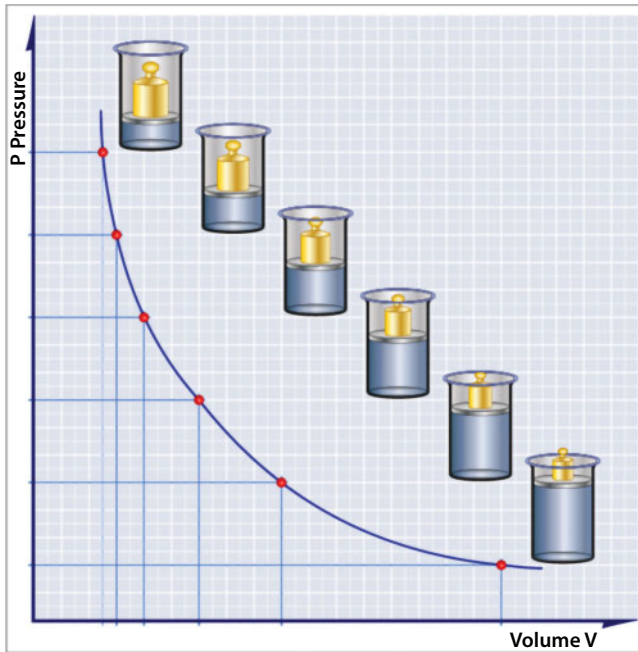


Figure 1.4 Starting with a piston having a given pressure and volume (far right piston), the volume continuously decreases as the applied pressure increases. If you plot pressure (P) as a function of the volume (V) for a given amount of gas at a certain temperature, you will get a plot that looks like the one shown here.

Boyle's law example:

A tire with a volume of 11.41 L (0.4029 ft³) reads 44 psia (pounds per square inch absolute) on the tire gauge. What is the new tire pressure if you compress the tire to a new volume of 10.6 L (0.3743 ft³)?

Answer:

First, we write out Boyle's Law:

$$P_1 V_1 = P_2 V_2 \quad (1.4)$$

Solving for P₂ we get:

$$P_2 = \frac{P_1 V_1}{V_2} = \frac{44 \text{ psi} \times 0.4029 \text{ ft}^3}{0.3743 \text{ ft}^3} = 47.36 \text{ psi}$$

$$P_2 = 47.39 \text{ psia}$$

Here is a listing of all the ideal gas laws along with what each one means:

Avogadro's Law states that equal volumes of all ideal gases (at the same temperature and pressure) contain the same number of molecules.

Boyle's Law states that equal pressure is inversely proportional to volume (when temperature is constant).

Charles's Law states that volume is proportional to temperature (when pressure is constant). Remember that temperature must be measured in Kelvin.

Gay-Lussac's Law states that pressure is proportional to temperature (when volume is constant).

Combining Charles's Law, Boyle's Law, and Gay-Lussac's Law gives us the combined law:

$$\frac{P \times V}{T} = \text{constant} \quad (1.5)$$

$$\frac{P_1 \times V_1}{T_1} = \frac{P_2 \times V_2}{T_2} \quad (1.6)$$

If we consider Avogadro's Law, we can combine all four gas properties into one equation:

$$\frac{P \times V}{n \times T} = R \quad (1.7)$$

The constant R in this equation is called the Universal Gas Constant.

The ideal gas law in this form is the most useful and should be memorized by those dealing with gases on a regular basis.

n = number of gas particles in a container

P = Pressure

T = Temperature

V = volume of the container

Keep in mind that these gas laws only apply during isothermal compression, i.e., compression occurring at a constant temperature, inside lab devices that move slowly and are poorly insulated. We'll discuss why this is important a little later in this chapter.

1.6 Gas Mixtures

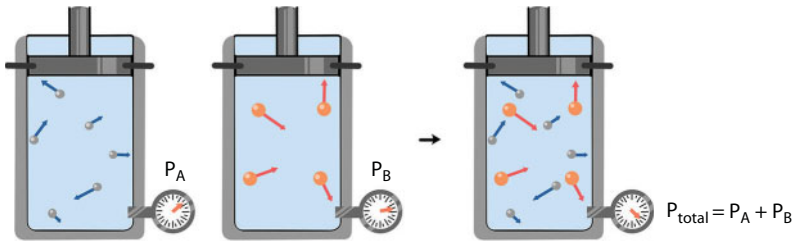
1.6.1 Dalton's Law of Partial Pressures

Most gaseous systems contain a mixture of gases. For example, air is a mixture of nitrogen gas, oxygen gas, xenon gas, carbon dioxide gas, and many others. A typical “neon” light on a Las Vegas marquee contains argon gas as well as neon. When working with a mixture of gases, we are sometimes interested in the total pressure exerted by all the gases together, and sometimes we are interested in the portion of the total pressure that is exerted by only one of the gases in the mixture. The portion of the total pressure that one gas in a mixture of gases contributes is called the partial pressure of the gas. Partial pressure is defined as the pressure exerted by one of the gases in a mixture if it occupied the same volume on its own.

One of the properties of gases is that they mix with each other. When they do so, they become a solution—a homogeneous mixture. Some of the properties of gas solutions are easy to determine if we know the composition of the gases in the mixture.

In gas mixtures, each component in the gas phase can be treated separately. Each component of the mixture shares the same temperature and volume. (Remember that gases expand to fill the volume of their container and all gases in a mixture do that as well.) However, each gas has its own pressure. The partial pressure of a gas, P_i , is the pressure that an individual gas in a mixture has. Partial pressures are expressed in psi, millimeters of mercury, or atmospheres like any other gas pressure; however, we use the term *pressure* when talking about pure gases and the term *partial pressure* when talking about the individual gas components in a mixture.

Dalton's law of partial pressures states that the total pressure of a gas mixture, P_{tot} , is equal to the sum of the partial pressures of the components, P_i (see Figure 1.5):



The situation is summarized in **Dalton's Law of Partial Pressures**: *the total pressure of a mixture of gases is equal to the sum of the partial pressure of all the gases.*

Figure 1.5 Dalton's law of partial pressures.

$$P_{\text{tot}} = P_1 + P_2 + P_3 + \dots = \sum_{\text{\# of gases}} P_i \quad (1.8)$$

$$P_{\text{tot}} = P_A + P_B \quad (1.9)$$

Although this may seem to be a trivial law, it reinforces the idea that gases behave independently of each other.

1.7 Molecular Weight of a Gas Mixture

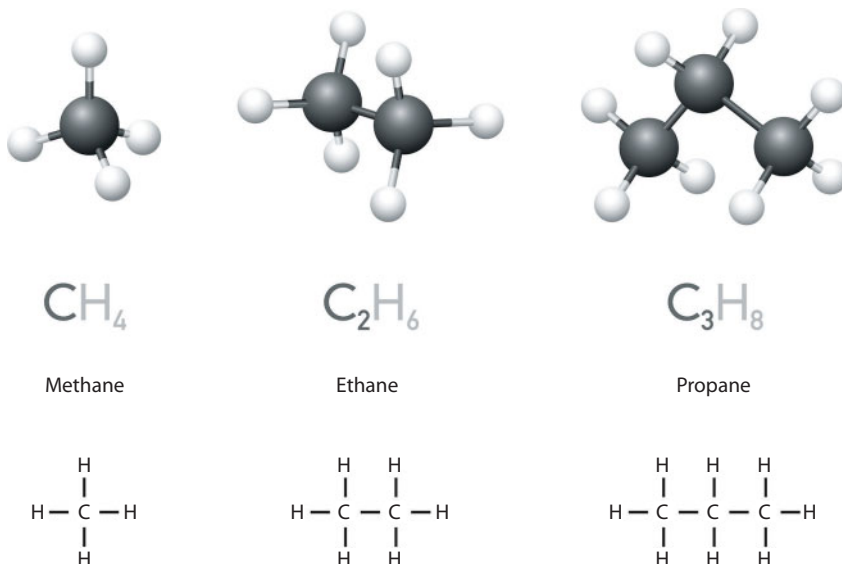
Because most industrial gases are composed of a mixture of gas, it is important to be able to determine the molecular weight of a gas mixture. To accurately calculate the molecular weight of a gas mixture, you need to know the molecular weights and percentage makeup of the mixture's constituent gases. The molecular weight of a gas mixture can affect a compressor's performance in several ways: for example, in dynamic compressors, a gas's molecular weight directly affects the compressor's horsepower consumption and the generated differential pressure.

Molecular weight (M) of a pure substance is the total mass of one of its molecules. It is calculated in practice by summing the atomic weights of the molecules making up the substance's molecular formula. For example, the molecular weight of a hydrogen molecule (chemical formula H_2) is 2 (after rounding

Table 1.1 Composition of methane.

Element	Symbol	Atomic mass	Number of atoms in molecule	Mass percent
Hydrogen	H	1.00794	4	25.132%
Carbon	C	12.0107	1	74.686%

Molecular weight of methane = $4 \times 1.00794 + 12.0107 = 16.04246$

**Figure 1.6** Molecular structure of methane, ethane, and propane.

off) and the molecular weight of methane is 16.04246 g/mol (see Table 1.1 below).

Hydrocarbons, such as methane, are substances composed entirely of carbon and hydrogen atoms. The three-dimensional molecular structures of the three common hydrocarbon molecules can be seen in Figure 1.6. You will notice that the methane molecule is tetrahedral in shape. The molecular structures seen in Figure 1.6 are dependent on the preferred orientation of covalent bonds between their atoms. (A covalent bond is a chemical

Table 1.2 Molecular weights of some common hydrocarbons. Compare the chemical formulas in this table with the molecular structures shown in Figure 1.6.

Gas	Chemical formula	Molecular weight
Methane	CH ₄	16.4
Ethane	C ₂ H ₆	30.07
Propane	C ₃ H ₈	44.09

bond that involves the sharing of electron pairs between atoms.) Table 1.2 lists the chemical formulas and molecular weights of these three common hydrocarbon molecules.

The average molecular weight of a gas mixture is determined by multiplying the molecular weights of each component by its mole fraction (Mole %/ 100) and then adding up all the values. (Note: The mole fraction or molar fraction (x_i) is defined as the quantity of a constituent (expressed in moles), n_i , divided by the total quantity of all constituents in a mixture (also expressed in moles), n_{tot} . In other words, the mole fraction of each constituent is determined as follows:

$$x_i = \frac{n_i}{n_{\text{tot}}} \quad (1.10)$$

To fully understand how to calculate the average molecular weight of a gas mixture, let's go through the following example (Table 1.3):

Note that we need to know the mole fraction and molecular weight of each component gas to determine the proportional molecular weight. The four-gas mixture has an average molecular weight of 14.84.

1.8 Gas Density

The density of a gas can be determined by utilizing a variation of the ideal gas law where density and molar mass replace moles

Table 1.3 Calculating the molecular weight of a gas mixture.

Gas	Mole % or Vol %	Molecular weight	Proportional MW
H ₂	61.4 x	2	1.228
N ₂	19.7 x	28	5.516
CO ₂	17.5 x	44	7.700
CO	1.4 x	28	0.392
Total %	100.0		14.84 is the MW of the total gas mixture

and volume. Where the original ideal gas law uses the formula $PV = nRT$, the density version of the ideal gas law is $PM = dRT$, where M is the molar mass. The molar mass of a gas composed of single atoms is given by the standard atomic weight of the element multiplied by the molar mass constant, $M_u = 1 \times 10^{-3} \text{ kg/mol} = 1 \text{ g/mol}$. For a molecule, the molar mass is the combined mass of all the atoms in a molecule in grams per mole. To calculate the molar mass of a molecule, first obtain the atomic weights from the individual elements in a periodic table. Then count the number of atoms and multiply it by the individual atomic masses. This sum represents the molecular weight of each molecule. Multiplying the molecules molecular weight by the molar mass constant ensures that the calculation is dimensionally correct.

1.9 Density of Mixtures

We learned from the ideal gas law that the density of a gas is directly proportional to its molecular mass (M), which is related to its molecular weight:

$$P \times M_A = \rho \times RT \tag{1.11}$$

where M_A is the molecular weight of the gas, ρ is the density of the gas, and R is the universal gas constant. If we know the density, total pressure and temperature of the gas the molecular weight can be easily determined from this equation.

What happens when we have a gas mixture? We use the density of the gas to determine the composition with the following equation: Density = mass (m) divided by the volume.

$$\rho = m/V \quad (1.12)$$

The density of a mixture of gases A and B is obtained knowing the total mass of the gases, which is the sum of the masses of each.

$$\rho = \frac{m_A + m_B}{V} = \frac{m_A}{V} + \frac{m_B}{V} \quad (1.13)$$

The ideal gas law indicates that the mass of A divided by the total volume is equal to $P_A \times M_A / RT$; therefore, if we replace the right side of the above equation with this relationship for each component we get:

$$\rho = \frac{P_A \times M_A}{R \times T} + \frac{P_B \times M_B}{R \times T} \quad (1.14)$$

Where P_A is the partial pressure of gas A and P_B is the partial pressure of gas B. Therefore, by knowing the partial pressure of each as in the mixture, we can determine the overall density of the mixture.

1.10 Heat of Compression

If you compress a gas adiabatically (without allowing heat to enter or leave) you are doing work on the gas, which increases its internal energy. Here's another way to look at the heat of compression:

compression occurs $\rightarrow$ gas volume decreases $\rightarrow$ the frequency of gas particle collisions increases $\rightarrow$ the gas temperature increases



Figure 1.7 Air fin cooler, which is located on the far right of the skid, provides interstage cooling for the reciprocating compressor (far left).

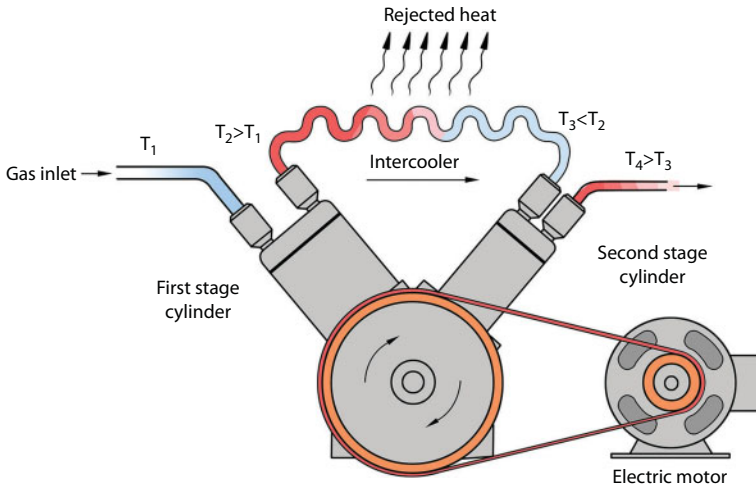


Figure 1.8 Schematic of a two-stage reciprocating compressor with an intercooler between stages. Notice that gas entering the second stage of compression has been cooled ($T_3 < T_2$) with an air-cooled heat exchanger before entering the second stage of compression.

Remember that the temperature is a measure of the internal energy (kinetic energy of the molecules for an ideal gas). Thus, compression tends to increase a gas's temperature.

Another way to explain the heat of compression is by remembering Boyle's Law: temperature increases because temperature and volume of gas are inversely proportional to each other according to gas laws. Therefore, when a gas is compressed rapidly, its volume decreases leading to increase in temperature.

The heat of compression effect is why we often need cooling between compression stages or downstream of the discharge of a compressor. In the processing industry, air fin coolers (Figure 1.7) are the most common means of cooling discharge gas. Air fin coolers, which can either be configured vertically or horizontally, force air across finned tubes filled with hot compressor discharge gas. The air coolers can either be driven off the driver shaft or powered by one or more electric motors. If designed properly, air fin coolers will drop the gas temperature to a manageable level, preferably between 80 to 100 °F, before it enters the next stage of compression (Figure 1.8) or the gas distribution piping.

2

Commonly Used Compressor Flow Terms

When pumping liquids, we typically use flows terms, such as gallons per minute, gallons per hour, barrels per day, etc. Because liquids are considered incompressible, pressure corrections are rarely required for liquid flow terms. We normally consider that the volumetric flow entering a pump remains essentially constant as it moves through a pump.

However, when dealing with gases, due to their compressible nature, a different set of terms is required to define flow. The purpose of this chapter is to introduce readers new to compression technology to the commonly used flow terms related to compressed gases. These terms are:

- Standard cubic feet per minute, SCFM
- Inlet cubic feet per minute, ICFM



Figure 2.1 Centrifugal Compressor in a Petrochemical Facility.

- Cubic feet per minute, CFM
- Actual cubic feet per minute, ACFM, which is used to define the flow of gases and vapors under pressure.

Here I will attempt to demystify this alphabet soup by explaining what these terms mean and how they are commonly used by practicing compressor professionals.

2.1 Ideal Gas Law

Before we can discuss flow definitions in more detail, let's quickly review some basic theory.

An ideal gas is defined as one in which all collisions between atoms or molecules are perfectly elastic and in which there are no intermolecular attractive forces. One can visualize it as a collection of perfectly hard spheres that collide, but otherwise do not interact with each other. In such an ideal gas, all the internal energy is in the form of kinetic energy and any change in internal energy is accompanied by a change in temperature.

An ideal gas can be characterized by three state variables: absolute pressure (P), volume (V), and absolute temperature (T). The relationship between them may be deduced from kinetic theory and is called the ideal gas law: $PV = nRT$

- n = number of moles
- R = universal gas constant = 8.3145 J/mol K

If the moles or weight in a given gas flow stream remains constant, we can write:

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2}, \quad (2.1)$$

where V can be in volume or volumetric flow for any two points in the flow stream.

Starting with equation 1, we can determine gas volumes at different conditions, such as standard conditions and actual process conditions. Flow at **standard conditions** (SCFM) is defined as flow in cubic feet per minutes of gas or air flow at standard conditions, which are typically set at 14.696 Pounds per Square Inch (psia), 60 Degrees Fahrenheit (°F) (520°R), and 0% Relative Humidity (RH). **Actual cubic feet per minute** (ACFM) stands for the cubic feet per minute of gas or air flow at the actual conditions inside the piping. A special distinction is given to flow at the conditions at the compressor inlet flange or nozzle. This flow is called **inlet cubic feet per minute** (ICFM) often used by compressor designers to determine actual gas or air velocities inside their machines.

If the process was moving a gas at exactly the "standard" condition, then ACFM would equal SCFM. Unfortunately, this

usually is not the case as process temperatures and pressure change from one point in the process to another. To move a gas, a positive pressure or a vacuum must be created. When positive pressure is applied to a standard cubic foot of gas, it gets smaller, and conversely when a vacuum is applied to a standard cubic foot of gas, it expands. The volume of gas after it is pressurized or rarefied is referred to as its "actual" volume.

Beware that some compressor manufacturers will use the term CFM for the flow of air out of the compressor discharge to the receiver or to the plant air mains. Others will use the acronym ACFM (Actual CFM) as the measure for air from their compressors.

Now we can begin calculating. Here is the equation to convert from SCFM to ACFM, assuming an ideal gas, i.e., $Z = 1$:

$$ACFM = SCFM \times \left(\frac{14.696}{P_{actual}} \right) \times \left(\frac{T_{actual}}{520} \right), \quad (2.2)$$

where P_{actual} is in psia and T_{actual} is in °R

Here is the equation to convert from ACFM to SCFM assuming an ideal gas:

$$SCFM = ACFM \times \left(\frac{P_{actual}}{14.696} \right) \times \left(\frac{520}{T_{actual}} \right), \quad (2.3)$$

where P_{actual} is in psia and T_{actual} is in °R.

Here is the equation to convert from SCFM to ICFM assuming an ideal gas:

$$ICFM = SCFM \times \left(\frac{14.696}{P_{inlet}} \right) \times \left(\frac{T_{inlet}}{520} \right), \quad (2.4)$$

where P_{inlet} is in psia and T_{inlet} is in °R.

2.1.1 Example of How to Convert from SCFM to ACFM

Let's say you want to know what volume of gas 2500 scfm represents at 200 psig and 250 °F. First, make sure you convert gauge

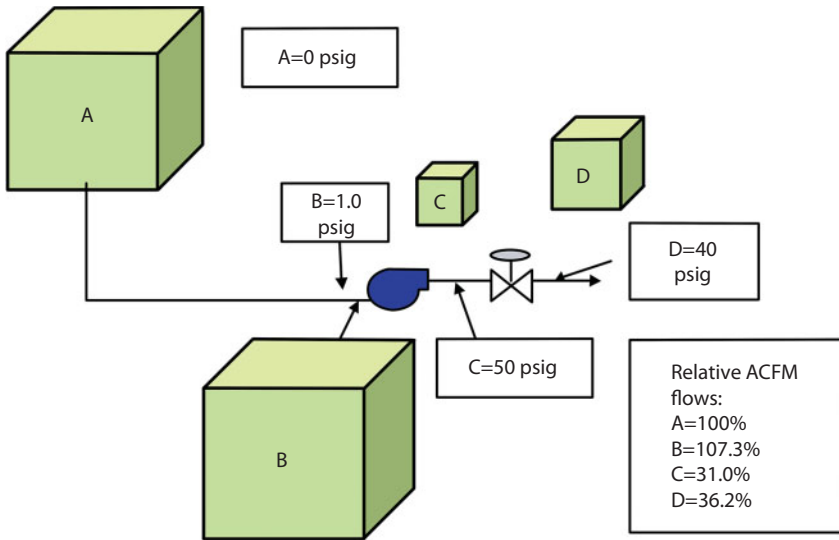


Figure 2.2 Hypothetical compressor piping system.

pressure into absolute pressure units, i.e., 200 psig = 214.7 psia and temperature into absolute units, i.e., 250 F = 710 R. Now we can readily calculate ACFM using equation (3) and obtain a value of 233.1 acfm.

$$\begin{aligned}
 ACFM &= SCFM \times \left(\frac{14.7}{P_{actual}} \right) \times \left(\frac{T_{actual}}{520} \right) \\
 ACFM &= 2500 \times \left(\frac{14.7}{(200+14.7)} \right) \times \left(\frac{(250+459.67)}{520} \right) \\
 &= 233.1 \text{ cubic feet / minute}
 \end{aligned}$$

2.2 Visualizing Gas Flow

To better understand the relationship between these different flow terms, let's study the simple compression system depicted in Figure 2.2. Here we have a compressor taking suction from the atmosphere (standard conditions) and discharging the gas into a higher pressure system of 50 psig. Eventually, the gas flows across a control valve, causing the

Table 2.1 Below Summarizes the Change in Volumetric Flow Rates at These Conditions.

Point	Note	Psia	Temp (F)	Cubic feet/minute
A	SCFM	14.7	60	1000
B	ICFM	13.7	60	1073.0
C	ACFM	64.7	250	310.2
D	ACFM	54.7	245	364.3

pressure to drop to 40 psig. If we follow the air flow through the compression system below, we can see how each of these flow terms apply.

Let’s assume we:

1. Start with 1000 scfm at standard conditions at the point “A”
2. Then we see a drop in pressure to –1.0 psig (13.7 psia) at the compressor suction at point “B”
3. Then we see an increase in pressure at point C to 50 psig.
4. Finally we see a drop in pressure to 10 psig at point B.

At the piping entrance (A), let’s say the gas volume is 100%. Because there is some pressure drop in the suction piping, the inlet volume (ICFM) at B will be slightly greater (107.3%) than the volume at A. After the gas is compressed, the volumetric flow at the compressor discharge, designated as actual cubic feet per minute (ACFM) will be less (31.02%) than the volumetric flow at A. Finally, after the control valve, the ACFM increases slightly (36.17%) compared to the flow at C because the pressure has been reduced.

2.3 Compressibility Factor (Z)

Keep in mind that the flow conversion equations presented above assume you are dealing with an ideal gas. In general, deviation from ideal gas behavior becomes more significant the closer a gas is to a phase change, the lower the temperature, or the higher the pressure. The compressibility factor (Z), also known as the compression factor, is the ratio of the expected volume of a gas to the volume of an ideal gas at the same temperature and pressure. The compressibility factor is a useful thermodynamic property for correcting the ideal gas law to account for the gas applications encountered in the real world.

If we insert the compressibility term into equation 1, we get:

$$\frac{P_1 V_1}{Z_1 T_1} = \frac{P_2 V_2}{Z_2 T_2} \quad (2.5)$$

After a little manipulation of equation 2.5, we arrive at the ACFM equation with the compressibility factor z:

$$ACFM = SCFM \times \left(\frac{14.696}{P_{actual}} \right) \times \left(\frac{T_{actual}}{520} \right) \times Z_{actual} \quad (2.6)$$

Let's look briefly at how the compressibility factor changes as a function of pressure for natural gas with a specific gravity of 0.6. (Specific gravity for a gas is defined as the ratio of the density of the gas to the density of dry air both measured at the same specific conditions of temperature and pressure.) Notice that for z to significantly reduce the idealized flow, the pressure must exceed 1000 psia for natural gas at 100 °F. As a rule of thumb for natural gas, we can say that z will vary from 1.00 only a few percent if your suction pressure is only a few hundred psia. However, if your suction pressure is much greater than a few hundred psi, you need to include z in your calculation of the ACFM.

Compressibility Factors for Natural Gas

Suction pressure (psia)	Compressibility factor (Z) at 100 °F
0	1
100	0.99
200	0.975
300	0.962
400	0.95
500	0.94
600	0.93
700	0.92
800	0.908
900	0.9
1000	0.89
1100	0.88
1200	0.87
1300	0.862
1400	0.857
1500	0.85
1600	0.84
1700	0.8375
1800	0.83
1900	0.8275
2000	0.82

Always remember that the compressibility factor is a function of pressure, temperature, and the actual mixture of gas analyzed, which means you’ll need to get an accurate set of

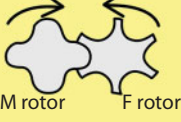
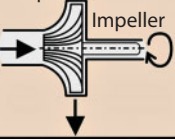
Compressor type	Capacity type		Centrifugal type
	Screw	Reciprocating	Turbo
Compression principle	Compresses by rotating screw rotor 	Compresses through the reciprocating motion of the piston 	Compresses through centrifugal force using impeller rotation 

Figure 2.3 The three main categories of gas compressors: Screw, Reciprocating, and Centrifugal Compressors.

compressibility tables for gases you plan to analyze. The “GPSA Engineering Data” books contain very detailed compressibility tables for natural gas mixture.

With time, these equations will become second nature to you as you apply them to real-world applications. You will eventually come to understand the relationships between volumetric gas flow, pressure and temperature. Next time someone asks you how much gas flow will you need, ask them: Do you want the flow in SCFM, ACFM, or ICFM?

2.4 Sizing Compressors

Regardless of whether the type of compressor you are trying to select is a screw, reciprocating, or centrifugal compressor (see Figure 2.3), they are all sized by knowing the inlet cubic feet of gas per minute (ICFM) entering the compressor’s suction. The reason ICFM is so important is that it can be converted to the gas velocity at the inlet conditions. As seen in Figure 2.2 the actual cubic feet per minute (ACFM) is constantly changing, therefore the velocity, as gas is moving through a compressor. The compressor designer must be aware of the ACFM of the gas as it moves through the compressor to ensure the compressor’s performance is optimized.

Table 2.2 Compressor Coverage Table (English Units)⁽¹⁾.

Compressor type	Inlet flow range (ICFM)	Maximum discharge pressure (psig)
Rotary Screw (oil free)	0–35000	175
Rotary Screw (oil flooded)	0–35000	290
Reciprocating (single stage)	0–2000	250
Reciprocating (multistage)	0–3500	80000
Centrifugal (single stage)	90–2000	2000
Centrifugal (multistage)	900–20000	10000
Axial	80000–800000	90

Table 2.3 Compressor Coverage Table (Metric Units).

Compressor type	Inlet flow range (m ³ /hr)	Maximum discharge pressure (bars)
Rotary Screw (oil free)	0–60000	12
Rotary Screw (oil flooded)	0–60000	20
Reciprocating (single stage)	0–3400	17
Reciprocating (multistage)	0–6000	5516
Centrifugal (single stage)	34–6000	138
Centrifugal (multistage)	1500–34000	689
Axial	136000–1360000	6

A simple way to select a compressor is to use the compressor coverage Table 2.2 and Table 2.3. If you know your inlet flow range in ICFM and the discharge pressure in psig, you

can simply locate the compressor types that can handle your conditions. For example, according to Table 2.2, applications requiring a flow of 50 ICFM and a discharge pressure of 150 psig can be handled by both oil free and oil flooded rotary screw compressors and reciprocating compressors.

3

Compression Processes

It takes power to compress a gas from a lower pressure to a higher pressure. Power is defined as force applied over a distance. In reciprocating compressor terminology, the distance traveled by a piston is called the piston stroke and the force required to move the piston is the difference in the pressure across the piston times the piston area ($P \times \text{Piston Area}$). During the compression process, the volume of the gas decreases and its temperature increases as the total internal energy of the system increases. In other words, during the compression cycle, energy is being converted from mechanical energy into gas energy in the form of increased temperature and pressure.

Recall, that when we studied Boyle's Law ($P_1 V_1 = \text{constant} = P_2 V_2$), it was stated that for this law to be valid the temperature (T) of the gas must be kept constant. This means for Boyle's Law to be accurate, the compression and



Figure 3.1 The ideal gas laws cannot be used to predict compressor performance.

expansion process must occur very slowly so that the gas remains at a constant temperature. In fact, all the gas laws we have discussed earlier in this book only apply during isothermal compression, i.e., compression occurring at a constant temperature, inside lab devices that move slowly and are poorly insulated. In reality, these conditions never occur in compressors due to their speeds and construction. It would be impractical to build compressors that moved slowly and compressed isothermally. Therefore, to properly analyze the performance of gas compressors, we must understand the various types of gas compression processes found in industry. The two most common types are the adiabatic compression process and polytropic compression process.



Figure 3.2 Reciprocating compressors' performance can be approximated using the adiabatic compression formula.

3.1 Adiabatic Compression

A thermodynamic process in which there is no heat transfer into or out of a system is called an adiabatic process. To perform an ideal adiabatic process, a system would have to be surrounded by a perfect heat insulator. If a compression or expansion of a gas takes place in a short time, it can be considered a nearly adiabatic, such as the compression stroke of a gasoline or a diesel engine.

In an adiabatic process, change is occurring within a system because of transfer of energy to or from the system in the form of work only; i.e., no heat is transferred. The adiabatic process obeys the relation:

$$pV^k = C = \text{constant} \quad (3.1)$$

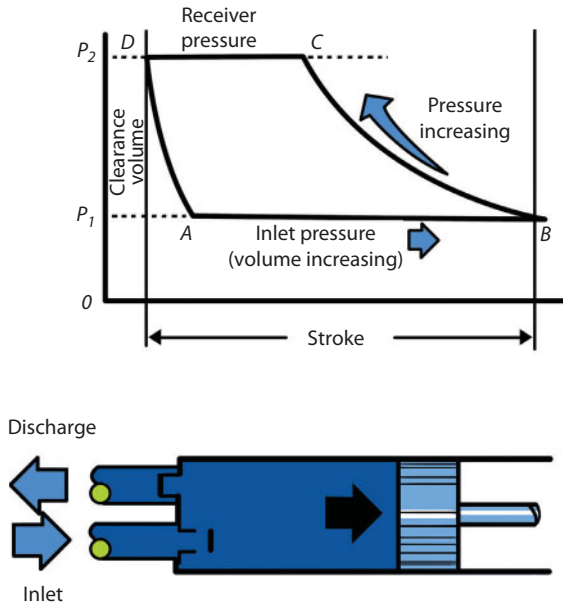


Figure 3.3 Pressure versus stroke diagram for a reciprocating compressor.

We can put this into words by saying that pressure (p) times the volume (V) raised to the power of k , which is the ratio of specific heats is equal to a constant, C . (Ratio of specific heats is defined as the specific heat at constant pressure divided by the specific heat at constant volume, which is sometimes referred to as the adiabatic index or the heat capacity ratio.)

To get a clearer idea what adiabatic compression is, let's consider a simple example: Consider a piston cylinder in a reciprocating compressor. As we push the piston from point B to C, the volume of the cylinder decreases, causing the pressure to increase. This is illustrated in pressure vs. volume graph below (Figure 3.3). Since the piston is moving at a high rate of speed, there is likely very little heat added or removed from the compressor cylinder during the compression process. In this example, we can conclude this situation approximates an adiabatic process. For similar reasons, we usually assume positive displacing compressors,

such as reciprocating and screw compressors, can be reasonably described as an adiabatic process.

One useful calculation for predicting the discharge temperature of a reciprocating compressor is the theoretical discharge temperature equation for adiabatic processes. The equation is written as shown here:

$$T_f = T_i \times \left(\frac{P_f}{P_i} \right)^{\left(\frac{k-1}{k} \right)} \quad (3.2)$$

Where:

T_f is the final absolute temperature (T in degrees Rankine = T in degrees Fahrenheit + 459.67)

T_i is the initial temperature (T in degrees Rankine = T in degrees Fahrenheit + 459.67)

P_f is the final absolute pressure (Psia = Psig + 14.7)

P_i is the initial absolute pressure (Psia = Psig + 14.7)

k is the ratio of specific heats

See the table below for a list of common ratios of specific heats (k).

Before the reader gets a math anxiety attack, let me explain how this equation can be analyzed in steps using a common scientific calculator:

Step 1: Calculate the compression ratio ($P_{\text{discharge}}/P_{\text{suction}}$) using the absolute pressure values. Remember that absolute pressure is the pressure in psi plus 14.7.

Step 2: Look up the k value for your gas (see Table 3.1) and calculate the exponent (k-1)/k. Write the value down.

Step 3: Using a scientific calculator, raise the compression ratio ($P_{\text{discharge}}/P_{\text{suction}}$) to the (k-1)/k power. Use the xy button on your calculator to perform this mathematical operation.

Step 4: Multiply the number obtained in Step three with your absolute inlet temperature, i.e., T_{suction} in degrees Fahrenheit + 459.67.

Step 5: Subtract 459.67 from the number obtained in Step 4.

The Ratio of Specific Heats

The ratio of specific heats is a thermodynamic parameter used in many compressor performance calculations, usually denoted by k . The ratio of specific heats is defined as $k = c_p / c_v$, where c_p is the specific heat for a gas in a constant pressure process and c_v is the specific heat for a gas in a constant volume process.

Table 3.1 Below contains K values for some common gases.

Gas	Ratio of specific heats
Carbon Dioxide	1.3
Helium	1.66
Hydrogen	1.41
Methane or Natural Gas	1.31
Nitrogen	1.4
Oxygen	1.4
Standard Air	1.4

The final number represents the final, or discharge temperature in degrees Fahrenheit.

Adiabatic Example 1:

Let’s assume we are compressing methane at 60 °F ($k = 1.31$ @ 60 °F) from 100 psig to 250 psig. First, we calculate the compression ratio: P_2 (absolute)/ P_1 (absolute) = (250 psi + 14.7 psi) / (100 psi + 14.7 psi) = 2.308. Next, we plug all the values in to the theoretical discharge temperature formula:

$$\begin{aligned} T_f &= T_i \times \left(\frac{P_f}{P_i} \right)^{\left(\frac{(k-1)}{k} \right)} \\ &= (60^\circ F + 459.67) \times (2.308)^{\left(\frac{(1.31-1)}{1.31} \right)} = 663.37^\circ R = 173.74^\circ F \end{aligned}$$

This result suggests that we can expect a discharge temperature of 173.74 °F. In reality, the actual temperature may vary from this value depending on the actual condition and construction of the compressor. As a reciprocating compressor wears and internal leakage becomes more and more significant, the difference between the actual discharge temperature and the theoretical adiabatic discharge temperature will increase. Keep basic equation in mind when you want to assess the condition of a reciprocating compressor under field conditions.

Adiabatic Example 2:

Now, let's assume we are compressing hydrogen at 60 °F (1.41 @ 60 °F) from 100 psig to 250 psig. The compression ratio is the same: P_2 (absolute)/ P_1 (absolute) = (250 psi + 14.7 psi) / (100 psi + 14.7 psi) = 2.308. Next, we plug all the values in to the theoretical discharge temperature formula:

$$\begin{aligned} T_f &= T_i \times \left(\frac{P_f}{P_i} \right)^{\left(\frac{k-1}{k} \right)} \\ &= (60 F + 459.67) \times (2.308)^{\left(\frac{(1.41-1)}{1.41} \right)} = 667.75 ^\circ R = 202.65 ^\circ F \end{aligned}$$

Notice by simply changing the type of gas being compressed that the theoretical discharge temperature increased from 173.74 °F to 202.65 °F. This simple example illustrates why it's important when troubleshooting to know if your gas conditions have changed from your original design conditions. Changes in the composition of the compressed gas can affect your discharge temperature as well as the horsepower required for compression.

3.2 Polytropic Compression

A polytropic process is a thermodynamic process that obeys the relation:

$$pV^n = C = \text{constant} \quad (3.3)$$

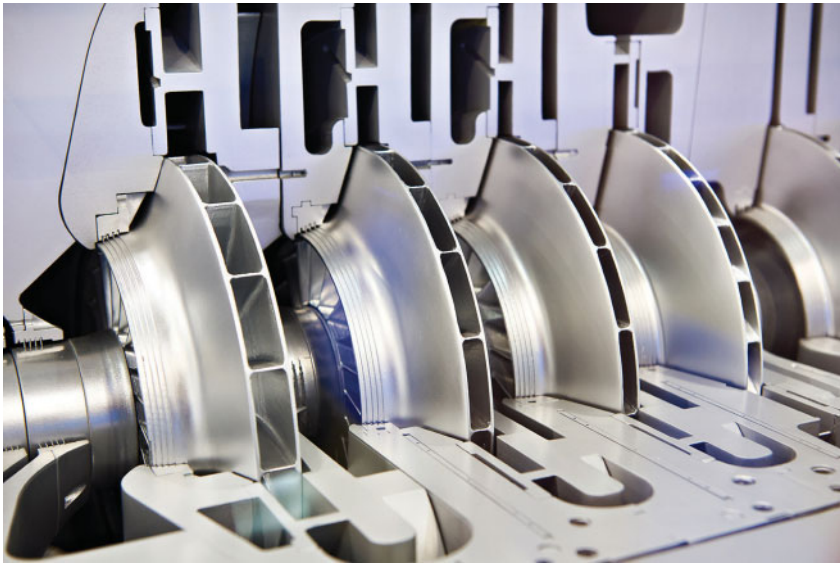


Figure 3.4 Centrifugal compressor performance can be approximated using polytropic formula.

We can put this into words by saying that pressure (p) times the volume (V) raised to the power of n , which is the polytropic index (defined below) is equal to a constant, C . The polytropic process equation can be described as multiple compression and expansion steps which include heat transfer. Equations based on the polytropic compression process closely predict the performance of real-world centrifugal compressors and are commonly used as the basis for evaluating centrifugal compressor field performance.

An ideal adiabatic process must occur very rapidly without any flow of energy in or out of the system. In practice, most expansion and compression processes are somewhere in between or said to be polytropic. Centrifugal compression tends to behave like a polytropic process.

One useful calculation for predicting the discharge temperature of centrifugal compressors is the theoretical discharge temperature equation for polytropic processes. The equation is written as shown here:

$$T_f = T_i \times \left(\frac{P_f}{P_i} \right)^{\left(\frac{n-1}{n} \right)} \quad (3.4)$$

Where:

T_f is the final absolute temperature (T in degrees Rankine = T in degrees Fahrenheit + 459.67)

T_i is the initial temperature (T in degrees Rankine = T in degrees Fahrenheit + 459.67)

P_f is the final absolute pressure (Psia = Psig + 14.7)

P_i is the initial absolute pressure (Psia = Psig + 14.7)

n is the polytropic index

$$\frac{n-1}{n} = \frac{k-1}{k} \times \frac{1}{\epsilon_p} \quad (3.5)$$

Where:

n is the polytropic index

k is the ratio of specific heats

ϵ_p is the polytropic compressor efficiency. (Typical polytropic efficiencies range from 70% to 85% percent.)

As I did before, let me explain how this equation can be analyzed in steps using a common scientific calculator:

Step 1: Calculate the compression ratio (P_f/P_i) using the absolute pressure values. Remember that absolute pressure is the pressure in psi plus 14.7.

Step 2: Look up the k value for your gas (see Table 3.1) and calculate the exponent $(k-1)/k$. Write the value down.

Step 3: Calculate the polytropic index exponent by dividing your $(k-1)/k$ by the polytropic efficiency, ϵ_p . (polytropic index = $(k-1)/(k \cdot \epsilon_p)$). Write the value down.

Step 4: Using a scientific calculator, raise the compression ratio (P_f/P_i) to the $(k-1)/(k \cdot \epsilon_p)$ power. Use the x^y button on your calculator to perform this mathematical operation.

Step 5: Multiply the number obtained in Step four with your absolute inlet temperature, i.e., T_i in degrees Fahrenheit +459.67.

Step 6: Subtract 459.67 from the number obtained in Step 5. The final number represents the final, or discharge temperature in degrees Fahrenheit.

3.2.1 Polytropic Example #1

Let's assume we are compressing methane at 60 °F ($k = 1.31$ @ 60 °F) from 100 psig to 250 psig with a centrifugal compressor that has a polytropic efficiency of 80%. First, we calculate the compression ratio: $(250 \text{ psi} + 14.7 \text{ psi}) / (100 \text{ psi} + 14.7 \text{ psi}) = 2.308$. Next, we calculate the $(n-1)/n$: $(k-1)/k \cdot \epsilon_p = (1.31 - 1)/1.31 \cdot 0.80 = 0.2958$. Now we can plug all the values in to the theoretical discharge temperature formula:

$$T_f = T_i \times \left(\frac{P_f}{P_i} \right)^{\left(\frac{n-1}{n} \right)}$$

$$= (60^\circ F + 459.67) \times (2.308)^{(0.2958)} = 665.53^\circ R = 205.86^\circ F$$

This result suggests that we can expect a discharge temperature of 205.86 °F. Remember that the actual temperature will vary from this value depending on the actual condition and construction of the compressor. As a centrifugal compressor wears and internal leakage becomes more and more significant, the difference between the actual discharge temperature and the theoretical polytropic discharge temperature will increase. Keep basic equation in mind when you want to assess the condition of a centrifugal compressor under field conditions.

3.2.2 Polytropic Example 2

Now, let's assume we are compressing hydrogen at 60 °F ($k = 1.41$ @ 60 °F) from 100 psig to 250 psig with a centrifugal compressor that has a polytropic efficiency of 80%. The compression ratio is the same: $(250 \text{ psi} + 14.7 \text{ psi}) / (100 \text{ psi} + 14.7 \text{ psi}) = 2.308$. Next, we calculate the $(n-1)/n$: $(k-1)/k \cdot \epsilon_p = (1.41 - 1)/1.41 \cdot 0.80 = 0.3635$. Now we can plug all the values in to the theoretical discharge temperature formula:

Next, we plug all the values in to the theoretical discharge temperature formula:

$$\begin{aligned} T_f &= T_i \times \left(\frac{P_f}{P_i} \right)^{\left(\frac{n-1}{n} \right)} \\ &= (60^\circ F + 459.67) \times (2.308)^{(0.3635)} = 704.31^\circ R = 244.64^\circ F \end{aligned}$$

Notice by simply changing the type of gas being compressed that the theoretical discharge temperature increased from 205.86 °F to 244.64 °F. This simple example illustrates why it's important when troubleshooting to know if your gas conditions have changed from your original design conditions. Changes in the composition of the compressed gas can affect your discharge temperature as well as the horsepower required for compression.

4

What Role the Compression Ratio Plays in Compressor Design and Selection

One of the critical parameters in compressor design and selection is the compression ratio, often denoted as r , required for each stage of compression. The compression ratio (r) is simply the ratio of the absolute stage discharge pressure to the absolute stage suction pressure, i.e., P_2 (absolute)/ P_1 (absolute). Because most gases increase in temperature when they are compressed, the final compressor outlet temperature is always a concern. A high discharge temperature can lead to the failure of internal components due to material degradation or excessive thermal expansion. Compression ratio is also important in the determination of the required horsepower, i.e., the higher the ratio the greater the required horsepower for that stage.



Figure 4.1 Skid mounted reciprocating compressor.

4.1 Compression Ratio versus Discharge Temperature

Here is a simple example of how to calculate your compression ratio. Let's say you are compressing a gas with a ratio of specific heats of 1.3 (see Table 4.1 for common *ratio of specific heats* values) from a suction pressure -0.5 psig to a discharge pressure of 35 psig. To calculate the compression ratio, you first must convert both pressures to absolute pressures by adding 14.7 to them and then divide the absolute discharge pressure by the absolute suction pressure as shown here:

$$r = P_d/P_s = (35 + 14.7)/14.7 = 3.5 \quad (4.1)$$

Once we know the compression ratio, we can determine the theoretical discharge temperature, which assumes there are no internal losses and adiabatic compression, using the equation 2.

Common ratio of specific heats values

The ratio of specific heats is a thermodynamic parameter used in many compressor performance calculations, usually denoted by *k*. The ratio of specific heats is defined as $k = c_p/c_v$, where c_p is the specific heat for a gas in a constant pressure process and c_v is the specific heat for a gas in a constant volume process.

Table 4.1 Below contains *K* values for some common gases.

Gas	Ratio of specific heats
Carbon Dioxide	1.3
Helium	1.66
Hydrogen	1.41
Methane or Natural Gas	1.31
Nitrogen	1.4
Oxygen	1.4
Standard Air	1.4

$$T_d = T_s \times r^{\frac{k-1}{k}} \tag{4.2}$$

In this equation, *T* is in degrees Rankin (*R* = *F*+460), *k* is the ratio of specific heat, and *r* is the compression ratio calculated by equation 1. Assuming a suction temperature of 60 °F, *r* = 3.5, and *k* = 1.3, we arrive at a theoretical discharge temperature (*T_d*) of 234 °F.

Let's take this exercise a step further by increasing the compressor discharge pressure in 5 psi increments to see what happens to the discharge temperature. Table 4.2 summarizes the results. You can clearly see that as the discharge pressure increases, the compression ratio rises and correspondingly the discharge temperature (*T_d*) increases. In this example, *T_d*

Table 4.2 The Effect of discharge pressure on the theoretical discharge temperature.

Pd (psig)	Compression ratio	Theoretical discharge temperature	Design limit
35	3.50	234.3	275
40	3.85	249.8	275
45	4.20	264.3	275
50	4.56	277.9	275
55	4.91	290.7	275
60	5.26	302.8	275
65	5.61	314.3	275
70	5.96	325.2	275
75	6.32	335.7	275
Assumptions	Ps	−0.5	
	k=	1.3	
	Ts (°F)=	60	
	(k-1)/k=	0.231	
	Efficiency	100%	

increases from 234.3 F for a compression ratio of 3.5 to 335.7 F for a compressionratio of 5.61.

4.2 Design Temperature Margin

If we compare our results to a hypothetical design limit of 275 °F, we see that at a compressor discharge pressure of 50 psig,

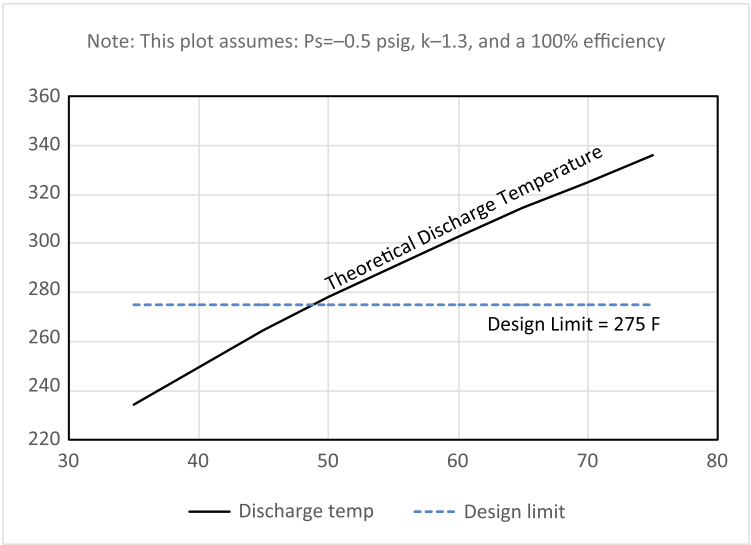


Figure 4.2 How the theoretical discharge temperature is affected by the compressor discharge pressure.

we begin to exceed our design limit temperature. This relationship between the theoretical discharge temperature and design limit temperature can clearly be seen in Figure 4.2. I recommend that you select a conservative design temperature limit during the selection phase of a project to ensure you will have a safe operating margin to handle any unexpected process conditions or unexpected internal cylinder losses due to wear or fouling.

Let’s assume the compressor you are considering has a recommended discharge temperature alarm limit of 325 °F and an automatic shutdown at 350 °F. If your actual discharge pressure is 60 psig, you should expect a minimum T_d of about 303 °F, as listed in Table 4.1. (Remember that the discharge temperature values in Table 4.1 are theoretical values. In reality, your discharge temperature will be higher due to internal losses as your compressor experiences normal degradation.) If your actual T_d is more like 318 °F, you will only have a margin of 7 °F!!

Note:

The examples shown here are based on theoretical adiabatic compression calculations. Adiabatic compression formulas are used to approximate the performance of positive displacement compressors, i.e., reciprocating and screw compressors. Readers should work with compressor manufacturers to determine actual compression ratio and discharge temperature limits for the specific compressor design. The theoretical calculations used in this column were used to demonstrate the general trend expected from high and higher compression ratios.

For centrifugal compressors, the following theoretical temperature rise equation, based on polytropic compression, is commonly used:

$$T_d = T_s \times r^{\frac{n-1}{n}} \quad (4.3)$$

Here $(n-1)/n$ is defined by the relationship:

$$\frac{(n-1)}{n} = \frac{(k-1)}{k} \times \frac{1}{\eta_p}$$

where η_p is the compressor's polytropic efficiency.

Equation 4.3 should only be used to estimate the theoretical discharge temperature in dynamic compressors.

This is going to lead to countless alarms and midnight phone calls. The way to avoid this situation is to use a conservative design discharge temperature specification and use more compression stages to ensure smaller compression ratios per stage. Inspection of Table 4.1 will lead you to the conclusion that for this example design compression ratios should not exceed 4.5

per stage if you wish to maintain a healthy margin between the operating temperature and alarm limit.

4.2.1 Design Trade-Offs

The reader can see there is always a trade-off between number of compression stages and the compressor discharge temperature at each stage. The more compression stages you have the costlier the compressor installation will be due to its overall complexity, i.e., more piping and cooling required. However, the more compression stages installed the smaller the temperature rise will be across each stage, allowing the compressor components to operate cooler and more reliably. This is the trade-off all compressor professionals must make as they begin their selection process.

The overall compression ratio is the product of all the individual compression ratios, i.e., $r_o = r_1 \times r_2 \times r_3 \times \text{etc.}$ Assuming all stages have the same compression ratio, we can write:

$$r_o = r_s^n \quad (4.4)$$

In this formula, n is the number of stages and r_s is the compression ratio per stage. From this relationship, we can conclude that the number of stages required to achieve the required overall ratio is:

$$n = \ln(r_o) / \ln(r_s) \quad (4.5)$$

For example, if your required r_o is 20 and our recommended maximum r_s is 4.5, then we would need two stages of compression, since:

$$n = \ln(20) / \ln(4.5) = 1.99$$

In general, we can conclude that the smaller the allowable compression ratio (r_s) is, the more stages (n), required.

My recommendation is to give your compressors a fighting chance. Try to use the widest economical margin between the calculated discharge temperature and the compressor design temperature limit. In the example above, I recommended a design margin of 50 °F (325–275 °F) to provide a buffer for

unforeseen internal compressor degradation. If you expect factors such as dirty gas you may want to increase your design margin even further.

Remember, your best friend during the compressor selection process will be the compressor manufacturer. They will keep you out of trouble by helping you select the right compressor for your application. For them to be successful you must provide them the most accurate compression ratio, gas composition, and suction temperature specification available. Together, you can collaborate to select an efficient and trouble-free compressor.

Quick Review of Chapters 1 Through 4



Gases are typically composed of a mixture of molecules, that are widely spaced as they fill a volume of space. For a gas at room temperature and pressure, the gas particles themselves occupy about 0.1% of the total volume. The other 99.9% of the total volume is empty space, compared to liquids and solids, which occupy about 70% of their total volume.

Scientists simplified the model of gases by imagining the behavior of an ideal gas. An ideal gas differs from a real gas in that the particles are assumed to be point masses and have no

attractive or repulsive forces at all between the particles. The ideal gas model is used to predict changes in four related gas properties: volume, number of particles, temperature, and pressure. Keep in mind the ideal gas laws only apply during isothermal compression, i.e., compression occurring at a constant temperature, inside lab devices that move slowly and are poorly insulated.

Most industrial gases are a mixture of gases. The exact composition of the gas mixtures has a significant effect on how it behaves inside of a compressor as it is compressed, expanded, accelerated, decelerated. The higher a gas's average molecular weight the more horsepower it takes for compression and the more pressure is generated in a centrifugal compressor at a given speed.

There are two common types of compression cycles found in the compression processes: the adiabatic compression process and polytropic compression process. To properly analyze field performance, you must know the type of compressor that's being used. Positive displacement compressors, such as reciprocating and screw compressors, tend to behave like an adiabatic process, and centrifugal compressors tend to behave like a polytropic process.

The takeaway: It is important to know a gas's composition, pressure, temperature and the type of compressor being used to understand how it's performing under field conditions. Some critical operating conditions are:

1. Suction pressure
2. Discharge pressure
3. Compression ratio
4. Suction temperature
5. Discharge temperature
6. MW
7. Speed
8. Flow

Some rules to always remember when trying to understand compressor performance:

1. Pressure rises as the volume is decreased; conversely a gas's volume decreases as its pressure is increased.
2. Pressure drops as the volume is increased; conversely a gas's volume increases as its pressure is reduced.
3. The temperature of a gas rises as it is compressed.
4. The temperature of a gas drops as it is expanded.
5. To determine gas properties, you need to know its composition.
6. The two most common types are the adiabatic compression process and polytropic compression process.
7. Positive displacement compression processes are best defined by adiabatic equations.
8. Dynamic compression processes are best defined by adiabatic equations.

Note: If you are interested in improving your knowledge of basic compressor theory, you can work through the practice problems related to the material in Appendix A of this book.

5

An Introduction to Compressor Operations

5.1 Compression Basics

Gases are fluids that take the shape of their container and are highly compressible when compared to liquids. Gases can be either pure or a composition of different gases with varying molecular weights. Given two equal volumes of gas at identical pressures and temperatures, the volume containing gases with a higher average molecular weight (MW) will have a higher density, which means it will have a greater mass per unit volume. We must know the volume, pressure, temperature, and composition of a gas to fully define its nature and condition.

A compressor is a fluid handling machine that takes in gas at a lower density and pressure via a suction nozzle and compresses it, resulting in the gas having a higher density and pressure in the discharge nozzle. Figure 5.1 demonstrates how

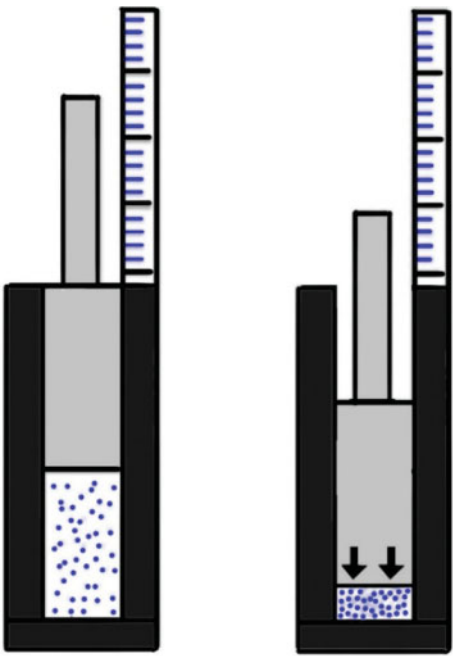


Figure 5.1 During the gas compression process, a volume of gas is decreased in order to increase its pressure, as shown above on the right.

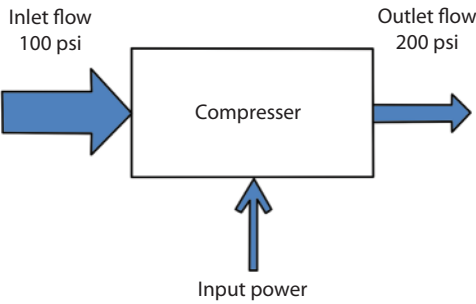


Figure 5.2 Compressor schematic.

compression works: It begins with a given starting volume of gas, as shown on the left in Figure 5.2. The gas is squeezed into a smaller volume, as shown on the right. Notice that the starting volume of the gas has been reduced, while the number of molecules inside the container remains the same. A smaller volume with the same number of molecules means that density

has increased. Most centrifugal compressors, like centrifugal pumps, have a larger inlet nozzle than the discharge nozzle making it easy to identify the general flow path through the compressor.

The act of compression requires power, provided by drivers such as motors, steam turbines, and gas turbines, to push the molecules into a smaller volume. This, in turn, forces them closer together and pressure to increase. It is the higher pressure that pushes gas flow out of the compressor's discharge nozzle into the process. In addition to an increase in pressure, the act of compression causes the gas to heat up. The greater the level of compression, the higher the temperature increase you can expect.

5.2 Defining Gas Flow

One way that compressors are different from pumps is the way flow rates are expressed. Because liquids are essentially incompressible, there is no need to be concerned about the effect of pressure on the flow volume. 10 ft³/minute of a liquid at pressure of 10 psi is essentially volumetric flow (ft³/minute) at 500 psi. The same cannot be said about gases. Gases are compressible, and the density of a gas is highly dependent on its pressure and temperature. Gas flow is expressed in one of two ways: 1) standard cubic feet per unit time or 2) actual cubic feet per unit time.

Standard cubic feet per minute (SCFM) is the flow rate of a gas corrected to standard temperature and pressure. Expressing gas flow in terms of standard conditions is handy because the value is independent of temperature and pressure. Standard conditions are 14.7 psi (absolute) and 60° F. If we say we have 100 standard cubic feet of a gas it means that at 14.7 psia and 60° F we would expect to have 100 cubic feet of gas. In contrast, the term “actual cubic feet per minute” (ACFM) is used to express the volume of gas flowing anywhere in a system. If a

system were moving a gas at exactly standard conditions, then ACFM would equal SCFM.

Figure 5.2 shows a gas compressor changing a gas stream from 100 psi to 200 psi. The larger input arrow on the left is meant to show that the actual cubic feet per minute is larger than the actual cubic feet per minute at the outlet conditions shown by the smaller arrow on the right. Even though the actual cubic feet per minute value is smaller on the right, the standard cubic feet per minute is the same at the inlet and outlet conditions. The lesson to remember here is that you need to understand which measurement units of the flow you are dealing with when examining compressor performance.

5.3 Compressor Types

Compressors come in many different designs in order to handle a wide variety of process applications. The two most common compressor designs are *positive displacement* and *dynamic* (see Figure 5.3). Positive displacement compressors include reciprocating (see Figure 5.4) and rotary units (see Figure 5.5). Positive displacement compressors work by continually forcing gas into a smaller and smaller volume, using either a piston or tight-fitting rotors, and then expelling the reduced volume of gas into a discharge passageway.

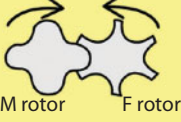
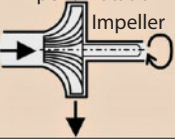
Compressor type	Capacity type		Centrifugal type
	Screw	Reciprocating	Turbo
Compression principle	Compresses by rotating screw rotor  M rotor F rotor	Compresses through the reciprocating motion of the piston  Piston	Compresses through centrifugal force using impeller rotation  Impeller

Figure 5.3 Basic compressor designs.

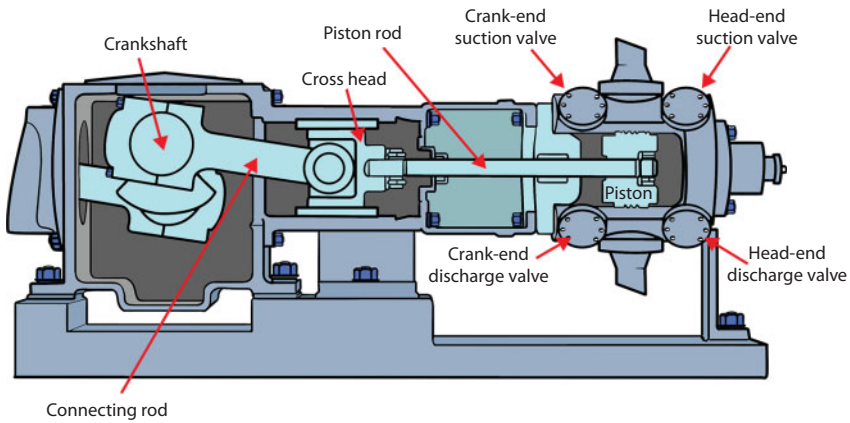


Figure 5.4 Cross section of a reciprocating compressor.

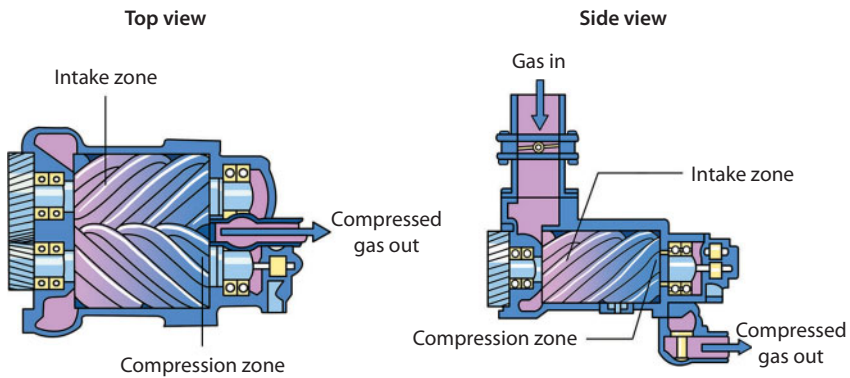


Figure 5.5 Rotary screw compressor.

Figure 5.5 shows a cutaway view of a rotary screw compressor composed of two tightly fitting rotors inside a pressure containing housing. Positive displacement compressors generally deliver smaller gas volumes and much higher differential pressures than centrifugal or axial flow compressors. Piston compressors have a potential drawback of discharging the gas flow in pulses, which some processes may not be able to tolerate.

Centrifugal compressors (see Figure 5.6) and *axial* compressors fall in the category of dynamic compressors. This type works by accelerating gas with a rotating impeller or set of rotating blades and then converting the exiting high velocity

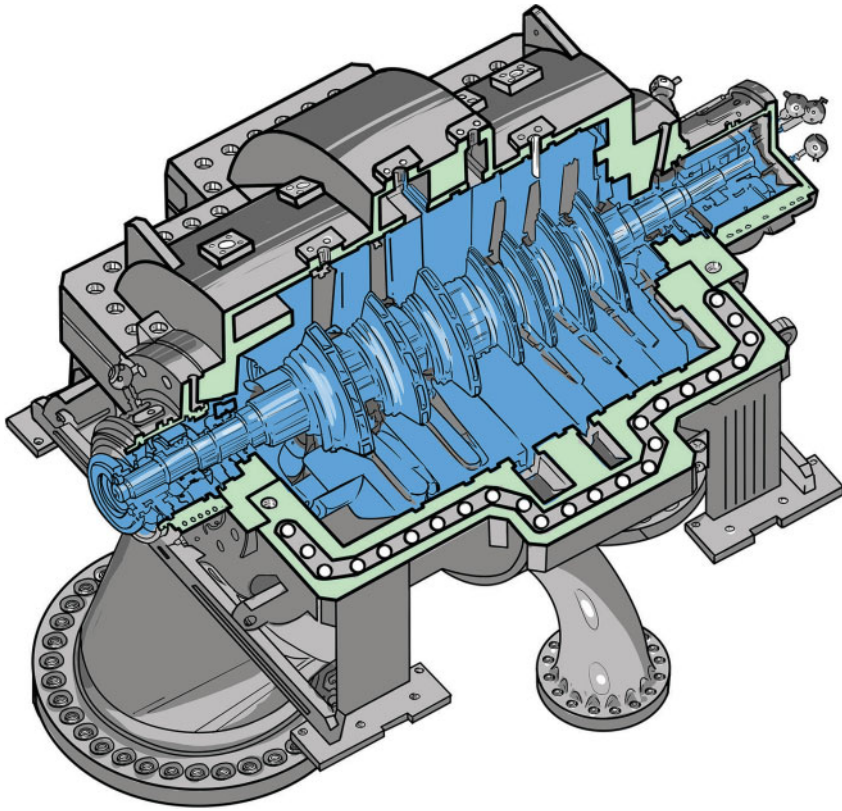


Figure 5.6 Multistage centrifugal compressor.

gas stream into a higher pressure by decelerating the gas stream for the next compression stage. Typically, centrifugal compressor inlets are larger than the outlet making it easy to identify the inlet and outline nozzles when tracing down the piping. The output of these compressors tends to be continuous, with very low-pressure pulsations. However, because dynamic compressors create pressure due to aerodynamic effects, they are susceptible to a flow condition called surging, which is characterized by violent, periodic flow reversals followed by the reestablishment of flow from suction to discharge. This is a serious operating condition that should be avoided. Surging can be detected either by a characteristic sound or, if equipped with proximity thrust probes, by the periodic back and forth axial

Table 5.1 Compression ratio ranges.

Compressor type	Typical range of design compression ratios
Lobe blower	1 to 2.0
Screw-dry helical compressor	2 to 5.0
Screw-flooded compressor	2 to 5.0
Single-stage centrifugal compressor	Up to 1.2
Multistage centrifugal compressor	Up to 3
Non-lubrication reciprocating compressor	Up to 3.5 per stage
Lubricated reciprocating compressor	Up to 3.5 per stage

movement of the rotor. (More information on surge and surge control can be found in Chapter 9 of this book.)

Centrifugal and reciprocating compressors are the type most often found in most petrochemical processes. Centrifugal compressors are used whenever high gas flows (up to 20,000 inlet cubic feet per minute) must be delivered at low to moderate differential pressures (see Table 5.1). Reciprocating compressors are used whenever low to medium gas flows (up to 3500 inlet standard cubic feet per minute) must be delivered at high differential pressures (see Table 5.1). For extremely high delivery pressure (up to 50,000 psi) applications, special reciprocating compressors, called hyper-compressors, are utilized. This book will cover only conventional reciprocating compressors.

5.4 Multistaging

Because the act of compression heats up the gas, there is a limit to how much the gas pressure can be raised across any given compressor stage. Higher and higher pressures can be achieved by placing multiple stages of compression in series, that is, having

the discharge of one stage of compression feed the intake of the next stage and so on (see Figure 8.6). “Multistaging” is the term used whenever multiple stages are used to meet the pressure requirements of a given application. Both reciprocating compressors and centrifugal compressors can be configured in multistage arrangements. If the temperature from one compression stage is predicted to get excessive, then interstage cooling is required to drop the gas temperature to a safe level before entering the next stage. Problems with interstage cooling systems can result in loss of flow and pressure capability.

5.5 Key Reliability Indicators

Some *increase in the gas temperature* is normal and expected. Many compressors have intercoolers or aftercoolers to handle this expected increase in temperature. However, whenever internal compressor issues, such as internal recycling, fouling, etc., are present, gas temperatures can rise well above what is expected. For this reason, compressor temperatures should constantly be closely monitored as a means of detecting internal problems. If high temperature conditions are not detected early, they can lead to the degradation of internal non-metallic elements, which in turn can escalate into a major failure. It is best to set a temperature alarm on discharge of every compressor stage that is somewhat below the shutdown level. If the alarm sounds, you will have time to investigate the problem before the compressor either shuts down or you have to shut it down. Early intervention may prevent loss of run time productivity. This is another case where knowing what “normal” is and knowing what the temperature was yesterday, last week, and last month is critical to detecting problems early.

Vibration is another major concern with and excellent indicator of mechanical condition of compressors. Vibration can be caused by imbalance, flow issues, foundation problems, internal looseness, etc. Make sure to monitor piping and skid

vibration around reciprocating compressors as well the working components. When reporting a vibration problem, note any special conditions present when the abnormality occurs, such as the rotational speed or process conditions. These notations can be very helpful during troubleshooting later.

6

Centrifugal Compressors

Centrifugal compressors fall into a category of compressors called dynamic displacement compressors (see Figure 6.1). Rather than physically reducing the volume of a captured pocket of gas, akin to how positive displacement compressors create pressure, dynamic displacement compressors speed up the gas to a high velocity and then restrict the flow so that the reduction in gas velocity results in an increase in pressure.

Typically, multiple stages of rotating impellers with corresponding stationary diffusers are required to generate the required differential pressure (Figure 6.2) across a centrifugal compressor body. The compressor rotor, the rotating impeller/shaft assembly, spins inside a pressure containing casing that supports and houses the diffusers, which convert high velocity gas flow off each impeller into higher and higher pressures until gas flow reaches the discharge nozzle. The key components

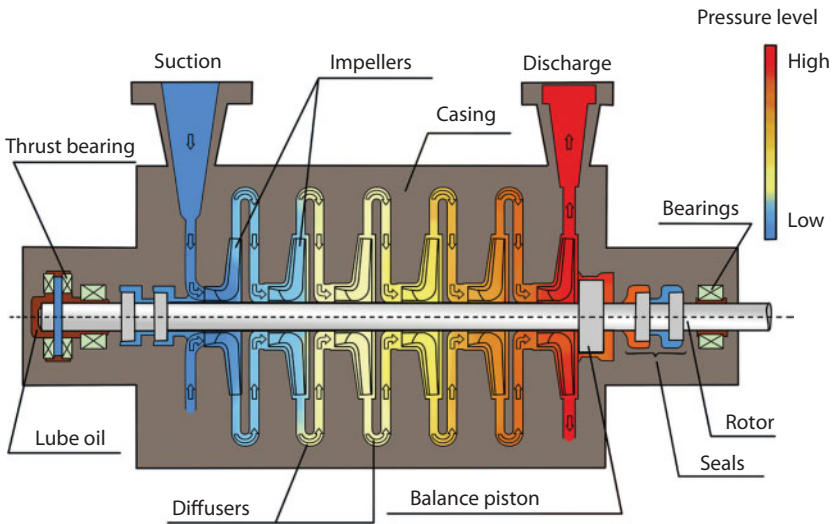


Figure 6.1 Centrifugal compressor cutaway.



Figure 6.2 Compressor rotor inside of a split casing. Notice that every impeller exit vanes line up with a corresponding diaphragm.

that make up centrifugal compressors are the rotor, rotor support bearings, end seals, diffusers between impeller stages, and a pressure casing (Figure 6.1). Labyrinth seals are incorporated throughout the compressor casing to minimize the leakage between stages (see Figure 6.3).

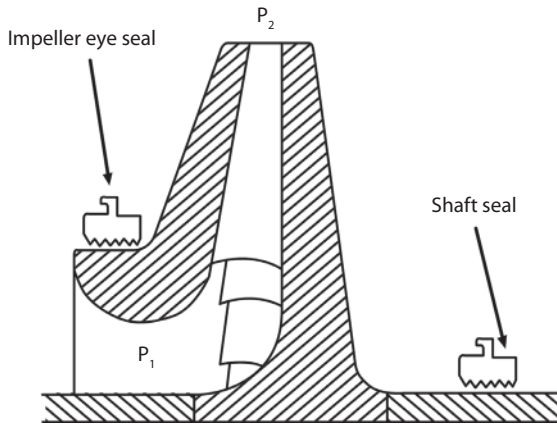


Figure 6.3 Labyrinth seals are used to minimize gas leakage between stages.

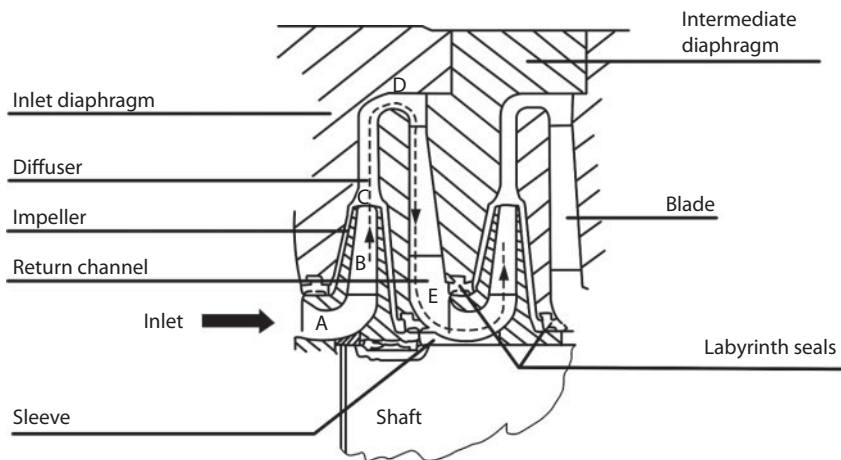


Figure 6.4 Cross section of centrifugal compressor.

Here is a brief description of what occurs inside each centrifugal compressor stage: Gas enters the eye of the impeller (A) and turns 90°, resulting in a radial flow toward the impeller's outer diameter (Figure 6.4). On its path through the impeller, the gas encounters rotating blades (B) that push the gas in a circular motion, which results in a static pressure rise due to the centrifugal force of rotation. For impellers with the typical backward-leaning blades, approximately two-thirds of

total static pressure increase of the stage is obtained within the impeller.

Upon leaving the impeller, the high velocity gas then enters the diffuser (C). The diffuser is a stationary component designed to convert gas velocity into pressure. The diffuser's radial inlet passage is about the same width as the impeller's exit area; however, the diffuser's radial area gradually expands in order to provide the desired pressure recovery effect. Most of the static pressure rise for the centrifugal stage has occurred after the gas exits the diffuser and enters the return bend (D). The return bend is a 180° passageway that is usually integral with the upstream stage's diffuser and the downstream stage's guide vanes. The purpose of stationary return bend is to redirect the gas into the return channel which contains the next stage's guide vanes (E). Each impeller stage adds more and more pressure to the gas flow until it reaches the discharge nozzle.

Centrifugal compressor performance curves are similar to those of centrifugal pumps. They show that the pressure rise across a compressor increases as speed increases and drops as the flow through the compressor increases (see Figure 6.5).

6.1 Centrifugal Compressor Piping Arrangements

Before attempting to troubleshoot a centrifugal compressor, the troubleshooters must understand the nuances of their specific compressor. They must ask: 1) How is the piping designed? 2) Are there any flow bypasses? 3) Are there any coolers? 4) Are the controls automatic or manual? Remember that the devil is in the details.

A typical piping arrangement for centrifugal compressor (shown in Figure 6.6) includes:

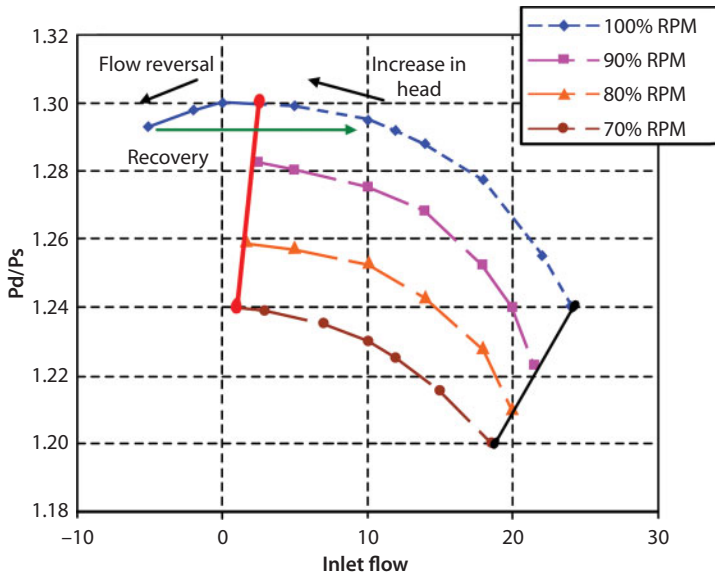


Figure 6.5 Centrifugal Compressor Performance Curve. There is a performance curve for operation at 100%, 90%, 80%, and 70% of the rated speed. At 90% speed with a flow of 20 units, the expected pressure ratio is 1.24.

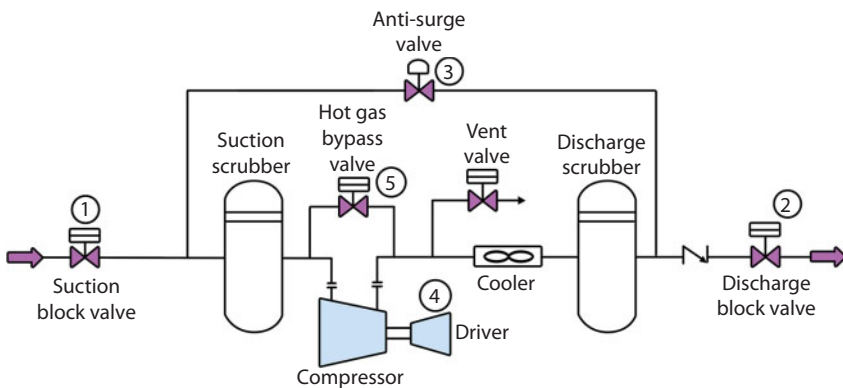


Figure 6.6 Typical single-stage compressor piping arrangement.

1. A suction scrubber to remove liquids and solids from the gas before it enters the compressor suction nozzle.
2. An anti-surge valve to protect the compressor from surge.

3. A hot gas bypass (HGB) valve that may be required for surge control systems that require quicker response times to transient conditions.
4. A discharge check valve to prevent backflow.
5. Gas coolers to cool discharge gas.
6. A vent valve to allow depressurization of the compressor system.

6.2 Start-Up Configuration

When starting up a centrifugal compressor, you need to know how to put the compressor into a bypass mode to unload the compressor. Additionally, you may need to throttle the compressor suction to reduce horsepower requirements for start-ups. Throttling is accomplished by pinching down on the suction valve in Figure 6.6.

Throttling should only be required during start-up. After start-up, ensure that the suction valve is fully open once you have attained normal operations. Keep in mind that, if you have trouble developing flow, you might need to check if the bypass line is open and robbing you of system flow.

6.3 Centrifugal Compressor Horsepower

A compressor's horsepower rating is defined as the amount of power that must be transmitted to the compressor to perform the desired function of compression. Typically, horsepower is transmitted from a driver, such as an electric motor, steam turbine, etc., to the compressor via a coupling. In its most basic form if the horsepower formula is written as the transmitted speed times the transmitted torque or $HP = \text{torque} \times \text{speed}$.

On the compressor end of things, process compressors are rated in terms of pressures and flow capacity. For example, a

compressor may be rated for 10 mmscfd of gas flow at a suction pressure of 100 psig and a discharge pressure of 250 psig. To better understand how driver horsepower and compressor horsepower are related, let's review a simplified version of a compressor horsepower equation. The equation below is for a special compressor, a blower. A blower is an air compressor that takes suction at atmospheric pressure and compresses it to a low (less than 1 psi) discharge pressure. Here is the equation for a fan (or blower).

$$HP = \frac{CFM \times PSI}{229 \times Fan\ Efficiency} \quad (6.1)$$

Where,

CFM = Cubic Feet per Minute (At suction conditions)

PSI = Pounds per Square Inch

Notice that there are two major design factors in this equation: 1) the air flow in cubic feet per minute and 2) the discharge pressure in psi. The equation seems to make sense since we know intuitively that 1) as the gas flow increases, the horsepower requirement increases and that 2) as the blower discharge pressure increases, the horsepower requirement must also increase. Note: These relationships only apply to radial flow centrifugal compressors. (Radial flow is flow that is oriented 90 degrees from the axis of rotation.)

For a compressor train that is operating at a constant speed, the required compressor horsepower must be exactly equal to the horsepower supplied by the drive, minus any mechanical losses. If there is inadequate horsepower supplied for compression, the train will slow down, and if there is ample horsepower provided for compression, the train will speed up. When variable speed drivers are used, some type of speed control is required to account for changes in load conditions. With a properly designed speed control system, the compressor train can maintain a constant speed during steady state operation, i.e., normal operations.

6.4 Troubleshooting Tips

Here are some useful relationships to remember when dealing with centrifugal compressors:

- If the pressure on the discharge pressure gauge increases, the flow is likely decreasing. Conversely, if the pressure on the discharge pressure gauge decreases, the flow is likely increasing if the suction pressure remains the same.
- If the compressor speed increases, the flow and discharge pressure will likely increase. The horsepower required by a compressor increases at higher speeds, which should be reflected by increasing amps or kilowatts if the compressor is motor driven. Conversely, if the compressor speed decreases, the flow and discharge pressure will likely decrease as with a turbine driven compressor. The horsepower required by the compressor at lower speed will decrease, which should be reflected by decreasing amps or kilowatts on motor driven compressors or decreasing steam input to the turbine.
- During start-up and shutdown or upset conditions, the gas composition can change. If the molecular weight of the gas being compressed increases, the discharge pressure will increase and the required horsepower will also increase. Conversely, if the molecular weight of the gas being compressed decreases, the discharge pressure will decrease and the required horsepower will decrease.
- If the suction pressure increases, the discharge pressure will increase and the required horsepower will also increase. Conversely, if the suction pressure decreases, the discharge pressure will decrease and the required horsepower will decrease. All else

being equal, a compressor is a differential pressure machine. Whatever goes in at one pressure will come out at a designed multiple pressure increase. If you raise the inlet pressure, the discharge pressure will also increase by the multiplier of the increase in the inlet pressure.

- Monitor the compressor balance line for temperature differences over time. A significant change can mean a balance piston, seal problems, which can result in damaged thrust bearings.

These relationships will help to troubleshoot a problematic centrifugal compressor. When reporting a problem with a compressor, always list the symptoms that caught your attention, such as a low flow, high amps, leaking seals, vibration, not enough pressure, etc.

6.5 Centrifugal Compressor Start-Ups

Every compressor is unique in its design and construction and therefore should have its own customized procedure to ensure that all necessary steps are followed during start-ups. Only approved procedures that follow the manufacturer's recommendations combined with operating experience on *this* compressor in *this* service should be used—attention to detail is paramount. If the compressor is motor driven, care must be taken to avoid multiple start-up attempts. Generally, if the compressor does not start by the second attempt there is no point in making a third attempt because large motors will generally lock out and there will be a significant cool down time before it can be attempted again. A check should be made of all the permissives after the first attempt to see why the compressor did not start. (A permissive is a particular type of safety interlock used to prevent actions from taking place until predefined criteria have been satisfied. For example, some permissives prevent

machine start-ups whenever there is a condition of low oil pressure, high suction drum level, high winding temperatures, etc. If these permissives are not cleared before more attempts to start, the motor will likely be locked out for hours before you can try to start the compressor again.)

Once full load conditions (i.e., speed, temperature, pressure, and flow) are reached, use the Centrifugal Compressor Checklist below to monitor machine condition and performance. Continue to monitor compressor conditions for at least an hour to ensure everything is normal. If unresolved issues are encountered during this start-up phase, either call for help in order to get a second opinion or shut down until additional technical assistance arrives to help you understand what is going on. If a problem is detected, be sure to capture actual data while the problem is present.

6.6 Centrifugal Compressor Checklist

- **Net flow through the compressor.** (Net flow is the total flow exiting the compressor.) Is the flow too low or too high? If the flow is low, check for open bypass or spill back valves. If the flow is too high, check the compressor speed.
- **Suction and discharge pressure.** Are the pressures normal? If not, begin to troubleshoot the system to find out what has changed. Look for a plugged strainer, fouled cooler, or incorrect valving.
- **Suction and discharge temperatures.** Are the temperatures normal? If the suction temperature is normal, but the discharge temperature is high, you may be looking at an indication of internal wear. A higher than normal discharge pressure may be caused by a higher than normal differential pressure.

- **High discharge temperatures or surging conditions.** These may indicate fouling. Check temperatures on any product coolers in the compressed gas stream. Fouling can be checked by looking at the color of the back flush water coming out of the exchanger or by a lack of differential temperature on water and gas flows into and out of the cooler.
- **Oil pressures and levels.** Be sure to check all lubrication levels and pressures, including seal oil pressures and flows if applicable.
- **Gas Seal Panel.** If the compressor is using dry-gas seals, check seal instrumentation panels to ensure flows and pressures are normal.
- **Bearing Temperatures.** If bearing temperatures are higher than normal first check over the lubrication system to make sure flows and supply oil temperatures are normal. Determine if it is the thrust bearing or the journal temperature that is running unusually high.
- **Vibration.** Are vibration levels normal? If not, have someone from the vibration department analyze the situation to determine what may be wrong and how long the compressor can continue to be run. Be sure to monitor the thrust position monitors. Remember that a failure in the thrust bearing will severely damage the machine beyond replacing the damaged bearing.
- **Power loading.** High power levels may be due to a higher than normal speed or heavier than normal gas. For example, a compressor that compresses hydrogen normally might experience driver overloading if it was started with nitrogen still in the system.
- **Signs of surging.** If surging is detected, and it is not too severe, try and find the source. It may be

eliminated by increasing flow by opening a spill-back line or by removing any possible restrictions. If the surging is very severe and there are no other in plant instructions, unload the compressor or shut down immediately and investigate the cause of the surging. Surging will wreck the compressor severely should it continue.

Common problems to watch for:

- Low or high rotational speed (unless motor driven).
- Low or high suction pressure.
- Low or high discharge pressure.
- Gas that is too light or too heavy (especially during start-up or upset conditions).
- Internal wear due to loss of interstage seals.
- Internal fouling.
- Plugged suction strainer.
- Discharge system restriction due to a partially closed valve, fouled piping or coolers. Watch for surging under these conditions.
- Power loading. High power levels may be due to a higher than normal speed, heavier than normal gas, higher than normal differential pressure or high suction pressure.
- Higher than normal balance line temperature.

7

How Process Changes Affect Centrifugal Compressor Performance

Key point: Compressor performance is greatly affected by density of the gas being compressed

7.1 Baseball Pitcher Analogy

To explain how a centrifugal compressor generates gas pressure, I will use a simple analogy we are all familiar with: A pitcher throwing a baseball. Let's assume the pitcher releases every ball he throws at 95 mph and that he throws two different types of balls: a standard baseball and a whiffle ball. We know that a baseball is about seven times heavier than a whiffle



Figure 7.1 Pitcher throws a baseball towards the batter's box.

ball. (Note a baseball weighs about 5.25 ounces and a whiffle ball weighs about .75 ounces, as found in Table 7.1 below.) By comparison a tennis ball weighs about two ounces.) Which ball represents the most energy after leaving the pitcher's hand at 95 mph? Obviously, the baseball will have about seven times more kinetic energy than the whiffle ball, since kinetic is proportional to mass (equation 7.1).

A dynamic compressor can be considered an analog for a baseball pitcher, due to the fact that it imparts a fixed speed to the gas stream for a given rotational speed. For example, a 20" centrifugal compressor impeller spinning at 6000 rpm will

Table 7.1 The Weights and weight ratios (ball weight/foam ball weight) of various types of balls.

Type of ball	Weight in ounces	W/W _{fb} ratio
Baseball	5.25	40.4
Rubber ball	4.0	30.7
Tennis ball	2.0	15.4
Whiffle ball	0.75	5.8
Foam ball	0.13	1

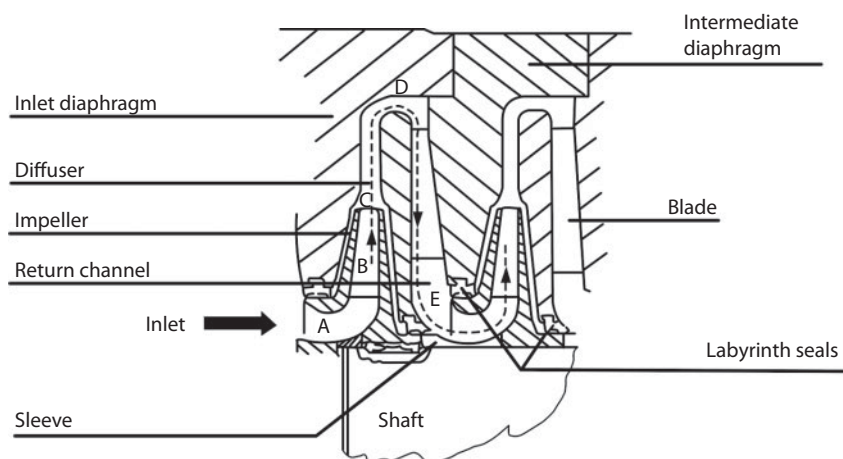


Figure 7.2 Cross section of a centrifugal compressor showing the impeller and diffusers.

have a blade tip speed of 524 ft/second, or 357 mph, regardless of the gas being compressed. As long as the compressor is supplied with sufficient horsepower, this 20" impeller will deliver gas at 524 ft/second at its exit.

Like the baseball example, a heavier gas density entering the inlet of a centrifugal compressor will generate more pressure, i.e., energy, at the outlet of the compressor. Inside a centrifugal compressor, the gas enters the impeller eye (Figure 7.2) where it

is accelerated and then thrown out radially toward the diffuser. In this analogy, the impeller is the pitcher who throws gas radially towards the casing and the diffuser acts like a gas catcher. The kinetic energy generated inside the impeller is converted into pressure energy as the gas decelerates within the diffuser and then gets directed either to the next stage of compression stages or out the discharge nozzle. This pitch-and-catch action converts shaft horsepower into gas horsepower. As a rule, more horsepower is required to compress heavier gases and in turn generate higher differential pressures.

So, at a set speed, a centrifugal compressor expels the gas at a given speed and with a given amount of kinetic energy per unit volume (see equation 7.1). It logically follows that as the gas density increases the pressure energy per unit volume generated by a centrifugal compressor goes up as the gas density increases.

$$KE = \frac{1}{2} mv^2 \quad (7.1)$$

Equation 7.1 is the equation for kinetic energy. If the impeller's exit speed (v) is constant, like to a baseball pitcher's toss, the kinetic energy will be proportional to the unit mass of the gas. Therefore, we can conclude that KE of the gas flow is directly proportional to density of the gas. For the sake of simplicity, let's also say the kinetic energy (KE) imposed on a gas stream is analogous to the differential pressure compressor generates. With this analogy in mind, we can infer that at a given impeller exit speed as the inlet gas density increases, the kinetic energy and the associated differential pressure across a centrifugal compressor will increase.

7.2 How Gas Density Affects Horsepower

To continue our pitcher analogy, let's return to our simple equation for blower horsepower. (A blower is an air compressor that

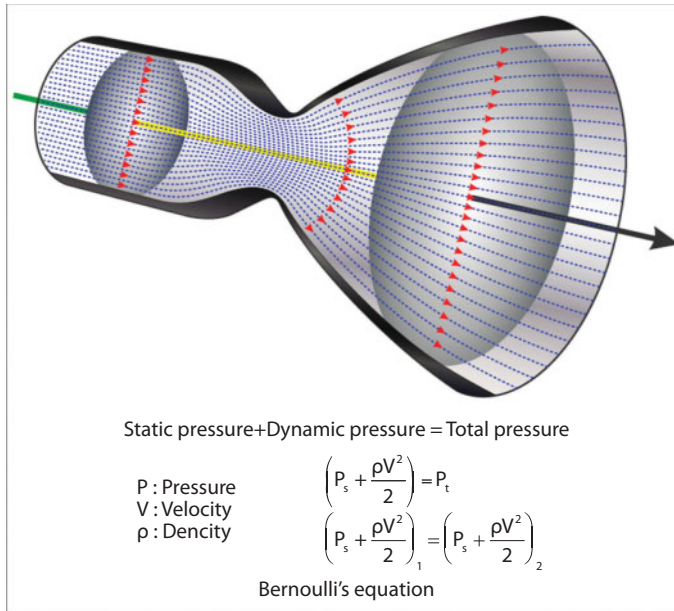


Figure 7.3 Technically, the creation of gas pressure inside a compressor is best explained by Bernoulli's equation, shown above, which mathematically describes what happens as the fluid velocity of a flow stream decelerates along its flow path. In basic terms, as the fluid decelerates the pressure in the flow stream increases. The reader should keep in mind that the pitching analogy used in this chapter is only a crude representation of what occurs inside a compressor, but it does relate the influence of velocity and gas density on pressure generation.

takes suction at atmospheric pressure and compresses it to a low discharge pressure.) Here is the equation for a fan (or blower).

$$HP = \frac{CFM \times PSI}{229 \times Fan Efficiency} \quad (7.2)$$

Where:

CFM = Cubic Feet per Minute (at suction conditions)

PSI = Pounds per Square Inch

Remember that before we said that kinetic energy in a gas is equivalent to the differential pressure that a compressor generates. Therefore, at a fixed inlet flow, the required horsepower (for a simple radial flow compressor) varies with the inlet gas density.

Assuming a single-stage gas blower operating at a constant volumetric flow and speed, we can make the following predictions:

1. As the inlet pressure increases, the gas density increases so the differential pressure and power requirements will increase.
2. As the inlet pressure decreases, the gas density decreases so the differential pressure and power requirements will decrease.
3. As the gas molecular weight increases, the gas density increases so the differential pressure and power requirements will increase.
4. As the gas molecular weight decreases, the gas density decreases so the differential pressure and power requirements will decrease.
5. As the inlet gas temperature increases, the gas density decreases so the differential pressure and power requirements will decrease.
6. As the inlet gas temperature decreases, the gas density increases so the differential pressure and power requirements will increase.

Table 7.2 summarizes the approximate relationships between changes in gas conditions and kinetic energy, horsepower, and pressure head. For example, an increase in molecular weight leads to an increase in horsepower requirements and head. An increase in suction pressure will result in a decrease in horsepower requirements and a decrease in pressure head.

7.3 Theory versus Practice

One day, a process engineer asked me, "What will happen to the compressor motor amps if we increase the interstage cooling?" The question involved a centrifugal compressor with two

Table 7.2 How deviations from design conditions affect centrifugal compressor performance.

Gas density = $\rho = \frac{P \times M}{T \times K}$	Kinetic energy	Gas horsepower requirements	Pressure head
Increase in suction pressure	Increases	Increases	Increases
Increase in molecular weight	Increases	Increases	Increases
Increase in suction temperature	Decreases	Decreases	Decreases
Decrease in suction pressure	Decreases	Decreases	Decreases
Decrease in molecular weight	Decreases	Decreases	Decreases
Decrease in suction temperature	Increases	Increases	Increases

compression sections. Between the first and second compression sections was a bank of fin-fan coolers used to drop the gas temperature before the gas entered the second section. I thought this was an easy question. I looked at my trusty cheat sheet and saw that if you decrease the inlet temperature of the gas flow that directionally the compressor power should increase, so that's what I told the process engineer.

I expected him to respond with kudos, but instead my theory was quickly rejected! He responded by saying that in fact he had conducted the experiment before. Whenever he increased the interstage cooling by starting up more cooling fans, the compressor amps would drop. His response didn't make sense to me until he explained that the additional cooling tended to condense the heavier ends in the gas, which would drop out as liquids and eventually dropped out in the second-stage suction drum.

There were two effects fighting each other in this situation. Intercooling was making the gas denser, which should have increased the horsepower requirements, but intercooling was also causing the heavier ends in the gas to drop out of the vapor phase. In our specific situation the controlling effect was the loss of the heavier ends from the gas stream. The net effect was that horsepower requirements were dropping because the effect of the average molecular weight of the gas was dropping exceeded the effect of the drop in the gas density due to cooling.

The moral of the story is that the tables presented here only apply if the average molecular weight of the gas is unchanging across the compressor. Multicomponent gas has a tendency of complicating matters. You need to clearly understand your gas properties when analyzing your compressors and the entire piping system.

8

How to Read a Centrifugal Compressor Performance Map

Users of centrifugal compressors need to learn how their compressors will perform under actual process conditions. They may want to know: How much pressure can a given compressor generate at a certain speed or flow or what is the maximum or minimum flow your compressor can operate safely? Manufacturers' compressor performance maps were created to help users answers these and many more performance-related questions. However, users must also be mindful that compressor maps have limitations and can be misleading if actual process conditions vary significantly from the original design conditions.

A *compressor performance map* is a diagram created to depict the expected performance of a process compressor and assist users in understanding how their compressor will function under certain conditions (see Figure 8.2). Detail performance



Figure 8.1 A cutaway of a centrifugal compressor showing the rotor and diffusers.

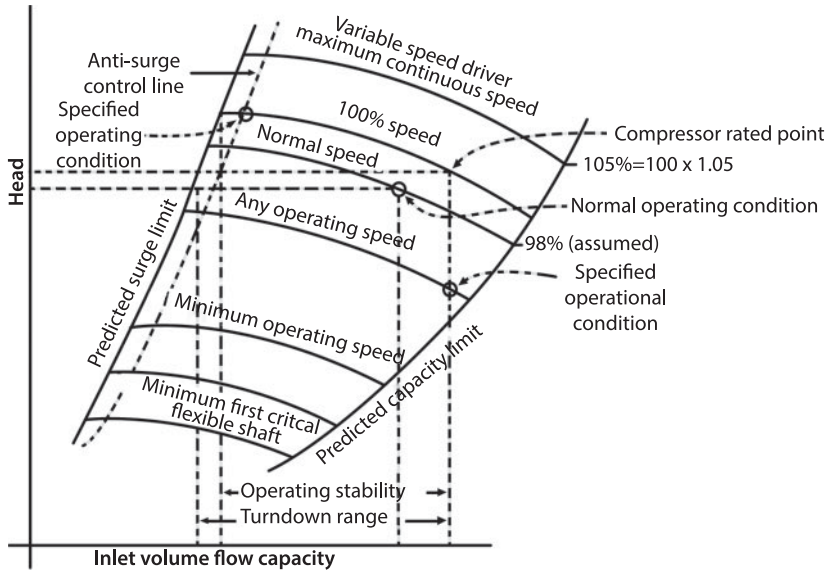


Figure 8.2 Generalized high-pressure compressor map.

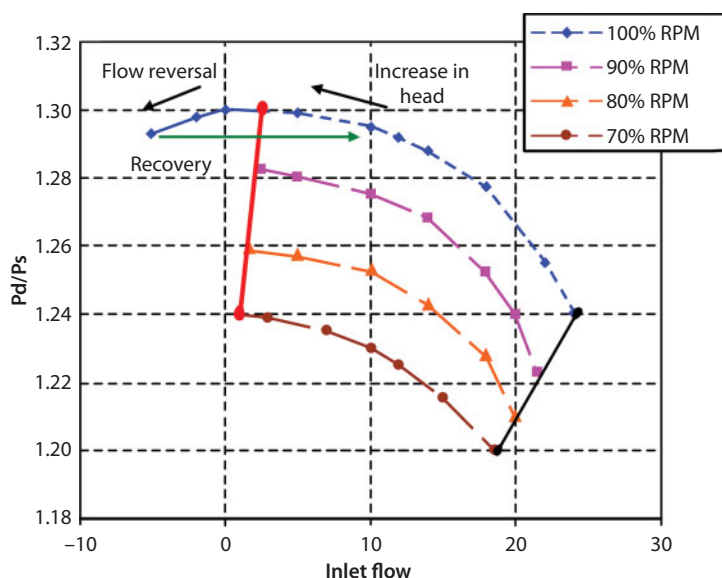


Figure 8.3 Typical high-pressure compressor map.

maps are based on compressor rig test results or predicted by a special computer program developed by compressor manufacturers. Compressor maps are necessary to predict gas compressor performance when operating at design and off-design conditions.

8.1 The Anatomy of a Compressor Map

While compressor maps may appear different from one another, they typically contain the similar features. Let's review the most noteworthy features:

8.1.1 Flow Axis (See Figures 8.2 and 8.3)

The x-axis (horizontal axis) represents the gas flow into the compressor's suction nozzle. The inlet flow can either be shown in standard cubic feet per minute, hour, or day or can be shown in actual cubic feet per minute, hour, or days.

8.1.2 Head or Pressure Ratio Axis (See Figures 8.2 and 8.3)

The y-axis (vertical axis) can either be shown in units of head rise or as the pressure ratio ($P_{\text{discharge}}/P_{\text{suction}}$) across the compressor. For a given set of suction conditions, i.e., gas molecular weight, suction pressure, suction temperature, the pressure ratio can be predicted if the flow is known. For example, in Figure 8.3, by studying the 80% speed line, we can predict a pressure ratio of about 1.25 at a flow of 10 units of flow and a pressure ratio of about 1.21 at 20 units of flow.

8.1.3 Predicted Surge Line (See Figures 8.2 and 8.3)

The line labeled “Predicted surge limit” on the far left of the map (Figure 8.2) defines the flow where compressor surge occurs. In Figure 8.3 the predicted surge line is depicted by a solid nearly vertical line at the left side of the compressor map. Centrifugal compressors surge is defined as the operating point where the centrifugal compressor peak head capability and minimum flow limits are reached. Surge can be a detrimental condition in compression systems because it causes the compressor to vibrate violently due to rapid internal flow reversals (see “flow reversal and recovery” in Figure 8.3), potentially damaging internal compressor parts. Surge control systems are specifically designed to prevent compressors from operating below the surge flow point or line. Figure 8.4 depicts a typical surge control system, which is composed of a flow measurement element, a controller with surge control algorithm program, and a recycle valve. If, at any time, the inlet flow approaches the surge control line flow programmed into the surge controller, the recycle valve will open to increase the total compressor flow and prevent surge from occurring.

8.1.4 Predicted Capacity Limit (Figures 8.2 and 8.3)

At the far right of the compressor maps in Figures 8.2 and 8.3 are lines that predict the capacity limits. These limits are called

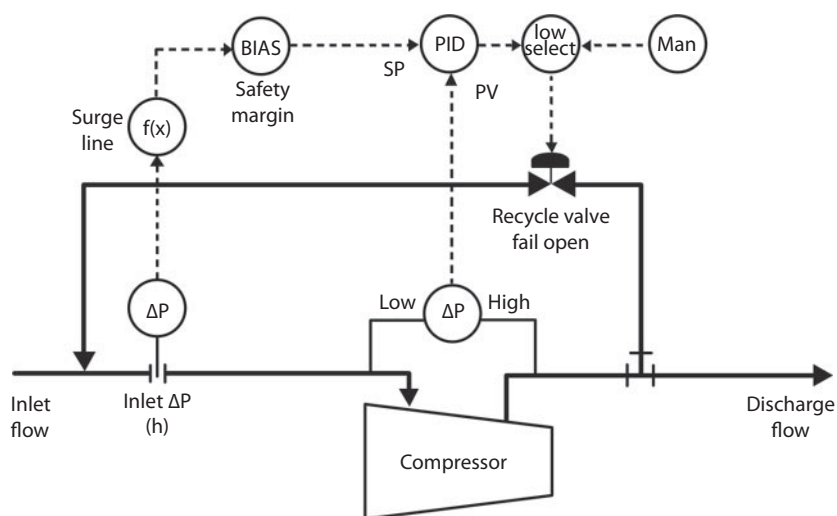


Figure 8.4 Typical centrifugal compressor surge control system.

stonewall or choked flow conditions. Centrifugal compressor stonewall or choked is a flow condition that occurs when the sonic velocity of gas is reached at the exit of a compressor wheel. When stonewall flow is reached, the discharge pressure and flow drop precipitously. Flow through the compressor cannot be increased any further without an increase in the suction pressure. At the stonewall (or choke) flow the pressure vs. volume curve drops precipitously and is not able to develop any significant head or pressure at higher flows. When the required operating flow exceeds the stonewall limit, the only remedy is to reconfigure the compressor with impellers (and matched stationary hardware) designed for larger flow rates.

8.1.5 Surge Margin (See Figure 8.2)

Most compressor manufacturers recommend a surge control margin to provide a safety operating margin in the event of a process upset. In Figure 8.2 the dotted line just to the right of the “predicted surge limit” is labeled the “anti-surge control line”. A surge margin of 10% to the right of the “predicted surge

line” or more is commonly programmed into the surge controller to prevent operating to the left of the surge line.

8.1.6 Speed Lines (See Figures 8.2 and 8.3)

Centrifugal compressor performance is highly affected by the rotational speed. At high rotational speeds, they generate more compressor head and therefore more flow. Notice in Figure 8.2, there are lines labeled as “minimum operating speed”, “normal speed,” and “maximum operating speed.” It is important that compressors are not operated beyond their rated maximum speed, where mechanical damage can occur.

8.2 Design Conditions

Most compressor maps provided to users, such as the one shown in Figure 8.3, are based on the design conditions provided to the compressor designers. Sometimes there is a single set of conditions provided and sometimes there are multiple conditions provided to take into account possible changes in future operating conditions. Users should keep in mind that it is vital to know your operating conditions when using your compressor map because centrifugal compressor performance is highly dependent on gas properties at the compressor suction nozzle.

Let's assume the compressor map in Figure 8.3 is based on the following inlet conditions:

1. Suction pressure of 100 psig
2. Suction temperature of 100 °F
3. Gas molecular weight of 28

These inlet conditions set the density of the gas entering the compressor and determine how much pressure the compressor can generate. If any of these inlet conditions vary more than

5% to 10%, there will be a significant effect on the compressor performance. Note: if you frequently find yourself operating far away from the original design speed or flow you might want to talk to the compressor manufacturer about a possible rerate.

Keep in mind that compressor performance maps estimate the performance of your compressors and are not 100% accurate under all field conditions. Changes in gas properties, suction conditions, speed errors, internal fouling, unexpected piping losses, can all affect the actual performance of your compressor.

Always, be aware of your actual operating conditions. If any of the inlet conditions change more than 10%, you should consider having the compressor manufacturer generate a new performance map. You may discover that the surge line, choke line, etc., may have all moved dramatically.

Note: If compressor internal components foul due to adverse process conditions, the compressor performance map can change due to a combination of reduced flow areas along the flow path and an increase in surface roughness. These effects can reduce a compressor's performance and overall efficiency.

9

Keeping Your Centrifugal Compressor Out of Harm's Way

Centrifugal compressors are simple machines. They have only one moving part, called the rotor. A centrifugal compressor operates by taking gas that enters the suction nozzle (see Figure 9.1) and accelerates it within an impeller and then it decelerates the gas inside a diffuser to create a positive differential pressure across the stage. Every successive impeller stage generates more and more pressure until it exits the compressor casing via the discharge nozzle and enters the process. Their design simplicity makes centrifugal compressors one of the most reliable process machine types found in process facilities. API designs, for example, are so reliable that they can easily operate over 10 years between overhauls or maintenance event.

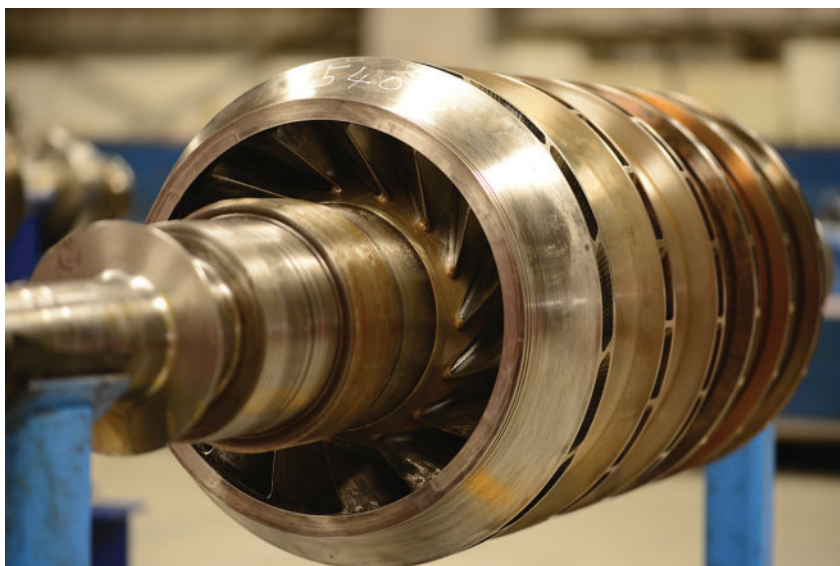


Figure 9.1 Multistage centrifugal compressor rotor.

You may ask: If centrifugal compressors are so simple why do they require so much associated piping and instrumentation? The reason is that there are certain operating conditions that can rapidly damage a centrifugal compressor, which is why it is wise to set clear operating limits for your compressors and install reliability safeguards to prevent operation outside manufacturer's prescribed limits. Centrifugal compressors are designed for a specific set of operating conditions, called the design conditions, that define expected flows, pressures, temperatures, and the gas properties. Significant changes from this design conditions can put your compressor in harm's way.

I will briefly discuss the various unsafe conditions compressor operators should be aware of and how safeguards can be instituted to prevent damage during the normal course of the compressor's useful lifetime. Safeguards can be either procedural in nature, automatic controls, alarms, or trips.

9.1 Compressor Operating Limits

An operating limit is any condition that has the potential to harm a compressor. For example, all compressors have a maximum speed limit. Above the maximum speed limit there is the potential to permanently damage the compressor rotor due to overstress. Every operating limit typically has a related adverse consequence, i.e., if a certain condition occurs, then something bad could happen. Some examples of operating limits are:

1. Critical speeds
2. Minimum speed
3. Maximum speed
4. Surge flow
5. Stonewall flow (sonic flow conditions)
6. Maximum discharge temperature
7. Maximum vibration (% of bearing clearance)
8. Radial and thrust bearing temperatures
9. Horsepower limit

9.2 Compressor Flow Limits

Figure 9.2 defines some of the operating conditions typically defined by centrifugal compressor manufacturers. I will briefly touch on surge and stonewall conditions here.

Centrifugal compressor surge is defined as the operating point where the centrifugal compressor peak head capability and minimum flow limits are reached. Surge can be a detrimental condition in compression systems because it causes the compressor to vibrate violently due to rapid internal flow reversals, potentially damaging internal compressor parts. Surge control systems are specifically designed to prevent compressors from operating below the surge flow point or line. Figure 9.3 depicts a typical surge control system, which

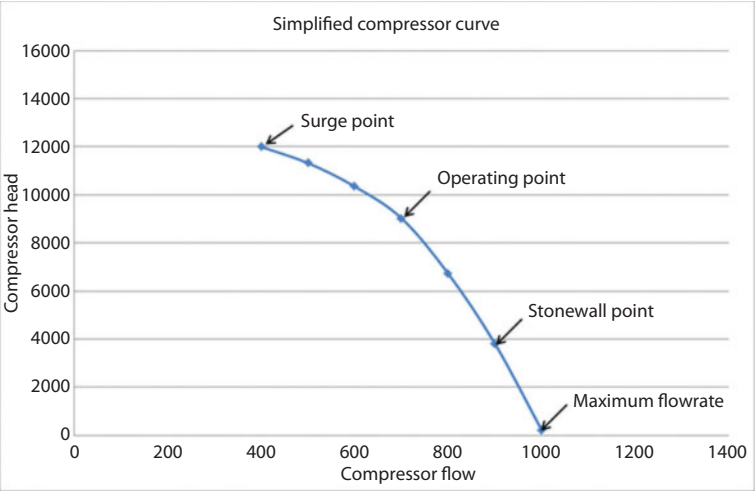


Figure 9.2 Centrifugal compressor operating limits. This compressor curve represents performance at a single speed.

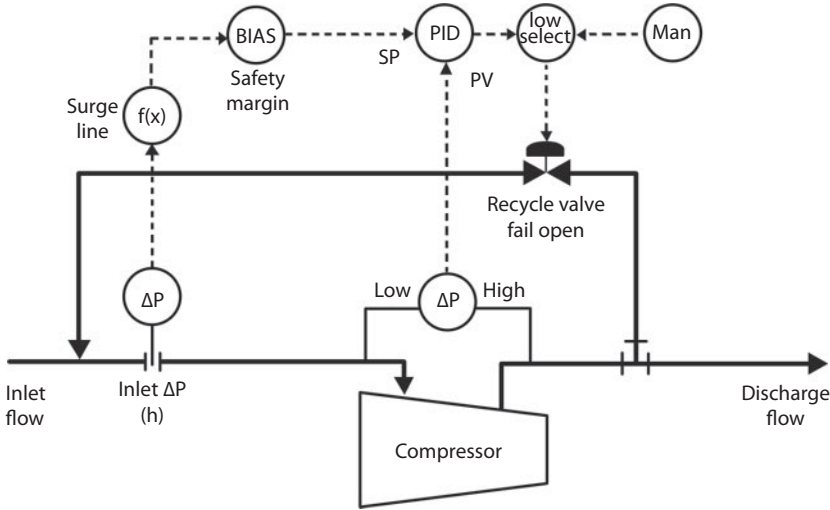


Figure 9.3 Typical Centrifugal Compressor Surge Control System.

is composed of a flow measurement element, a controller with surge control algorithm program, and a recycle valve. If, at any time, the inlet flow approaches the surge control line flow programmed into the surge controller, the recycle valve will open

to increase the total compressor flow and prevent surge from occurring.

Centrifugal compressor stonewall or choke flow is an operating condition that occurs when the sonic velocity of gas is reached at the exit of a compressor wheel. When stonewall flow is reached, the discharge pressure and flow drop precipitously. Flow through the compressor cannot be increased any further without an increase in the suction pressure. At the stonewall (or choke) flow the pressure vs. volume curve becomes essentially vertical, and it is not possible to develop head or pressure at any greater flow. When the required operating flow exceeds the stonewall limit, the only remedy is to reconfigure the compressor with impellers (and matched stationary hardware) designed for larger flow rates.

Take some time to perform a thorough technical audit of your field compressors. Ensure they have proper written and hardware safeguards to prevent surge or stonewall conditions from occurring. Also check to see that adequate overspeed prevention safeguards are in place.

9.3 Critical Speeds

A critical speed is defined as any operating rotational speed that coincides with one of the rotor's natural frequencies. Operation at the critical speed for extended periods of time can result in internal damage due to excessive vibration levels. It is important for operators to know their compressor critical speeds so they can be avoided. The original equipment manufacturer usually provides a list of critical speeds for users to avoid. Force response plots such as the one shown in Figure 9.4 can be used to locate critical speeds once the compressor has been installed in the field. The reader can see by studying the forced response plots in Figure 9.4 that rotor vibration increases significantly at about 1350 rpms and 2385 rpms due to the existence of rotor critical speeds.

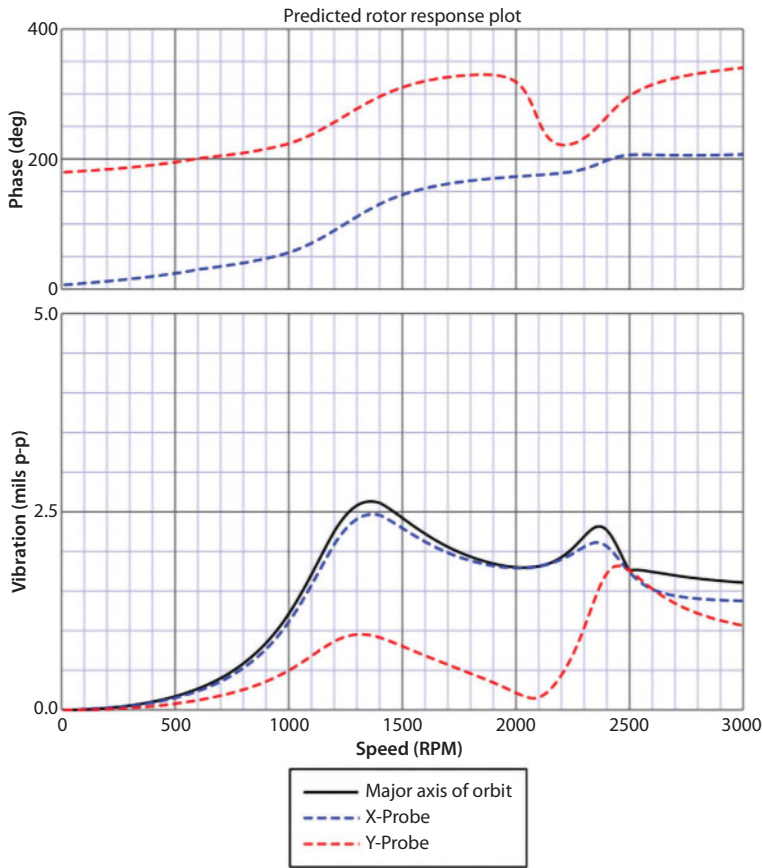


Figure 9.4 Here is a typical predicted forced response plot. The upper plot is the 1x phase versus the rotor speed and the lower plot is lateral vibration versus rotor speed. This analysis was run with mid-span imbalance. Notice: 1) There seems to be a rotor critical at about 1350 rpm where both the X and Y probes will see vibration peaks, which appears to be the rotor first critical. 2) There seems to be a second rotor critical at about 2375 rpm. 3) Both critical speeds have major phase shifts associated with them.

9.4 Horsepower Limits

The horsepower load on a centrifugal compressor is governed by the flow, gas density, the rotor geometry, and the rotational speed. For a given set of impeller diameters and gas conditions, the fan affinity laws define how flow, pressure, and the

required horsepower vary as the compressor's rotational speed changes:

1. The flow rate varies directly with the speed
2. The differential pressure varies with the square of the speed
3. The power use varies with the cube of the speed

Small operating changes can have significant effects on how a compressor performs, which is why operators should carefully track the compressor's overall operating characteristics, including the horsepower load, and look for trends. An overload condition may be telling you something is wrong. Overloading may be manifested by high amp loads for compressors with electric motors drivers or a loss of speed for compressors with gas or steam turbine drivers. If amp loads are seen to be rising or dropping, try to determine the cause of the change. Rising amp loads may be caused by changing process conditions or machine degradation, depending on the shape of the compressor performance curves. Falling amp loads may be caused by a decrease in flow due to a flow restriction or a decrease in the suction pressure.

9.5 Temperatures

As most process gases get compressed, they heat up in a predictable way. For a given suction temperature, we should expect a predictable temperature rise due to compression at the discharge nozzle. However, deviations from predicted temperature values can occur, which is why operators need to keep an eye on compressor gas operating temperatures. Temperature levels provide vital clues to the condition of a compressor. High compressor discharge temperature or interstage temperatures can be a sign of compressor degradation or a higher than normal compression ratio, due to system fouling or a restriction.

Takeaway: Gas temperature trending is a simple way of assessing the overall aerodynamic health of a centrifugal compressor.

Bearing temperatures can also provide vital clues to the condition of a centrifugal compressor. An upward trending bearing temperature is a sign that something is changing, such as the bearing's condition or the oil supply temperature. The most critical bearing temperatures are the thrust bearing pad temperatures. A failure of the thrust bearing pad will result in a catastrophic rotor failure as excessive axial movement occurs, which is why the axial position of the rotor should always be closely monitored and trended.

For an extra level of thrust bearing failure detection, a dual voting (2 out of 2, i.e., 2oo2 means that both sensors must be in a danger condition for a trip to occur). arrangement composed of two proximity probes are normally installed on critical compressors. (Note: The proximity probes must be installed axially so that they observe the thrust collar or other integral portion of the shaft surface. Probe targets, such as shaft sleeves, that are not integral to the shaft are not recommended.) Each machine train thrust monitor should be configured in a dual voting configuration, 2 out of 2 (2oo2), which is considered best practice by API 670 to ensure a high level of machinery protection. In some highly critical applications, an OEM or end user may even request a triple modular redundant (TMR) configuration, 2 out of 3 voting (2oo3), to achieve the required reliability level.

We must be ever vigilant if we wish to keep centrifugal compressors out of harm's way and operating reliably by avoiding unwanted operating conditions. Reliable operation requires:

- 1) proper physical safeguards, such as surge control, temperatures and vibration monitors, load monitors, speed sensors, etc., to prevent and detect unwanted operating conditions, and
- 2) well-written operating procedures. Before writing compressor operating and monitoring procedures, you need to clearly understand your operating limits. If in doubt, talk to the

original equipment manufacturer about any concerns you may have. Always try to be conservative when setting your limits. If you faithfully protect your compressor, you will be rewarded with many years of safe and reliable service.

10

Troubleshooting Centrifugal Compressors in Process Services

What is the sound of one hand clapping? This thought-provoking koan is used to challenge students engaged in the practice of Zen. (Note: A koan is a paradox used to train Zen Buddhist monks to abandon ultimate dependence on reason and to force them into gaining sudden intuitive enlightenment.) While this question seems out of place here, it applies to process machinery very well.

I ask the reader to answer this question: Can a centrifugal compressor operate within expected parameters without a complete system? The answer is clearly no. Without a continuous suction gas supply, some type of control system, and piping system, a centrifugal compressor could never function properly. Just as a single hand cannot clap, the components in a gas compression system cannot function independently. In

other words, the compressor and its subsystems must all function together to meet the process needs.

If one element of this compression is malfunctioning, the whole system will suffer. Here are some possible system issues that can lead to serious consequences:

- Suction strainer is partially plugged
- Discharge piping has some unexpected blockage
- Suction or discharge piping is undersized or oversized
- Spillback valve is leaking back to the compressor suction
- The suction pressure is significantly lower than the design value
- Compressor driver is undersized
- Foundation is too flimsy
- Piping resonance is present due to lack of supports
- Oil coolers are undersized
- Relief valve is leaking
- Vibration sensors are out of calibration

Even though any of these non-machinery issues will usually lead to major problems, it would be no surprise if one of the resulting symptoms was reported as a “compressor problem.”

Similarly, here are some issues with subsystems that could lead to undesirable symptoms in other machine classes:

- Low suction pressure on a centrifugal compressor will increase the overall compression ratio and lead to high discharge temperatures.
- Low gas density could cause centrifugal compressor flow and differential pressure to both run below design levels.
- Scaling or a flow blockage in an oil cooler would cause rolling bearing temperatures to rise.

- Process changes could cause gas properties to vary significantly from the original design condition. A change in gas properties could result in a number of problems, such as driver overload, low of flow, surging, etc.

To successfully assess and troubleshoot process machinery, owner/operators must take a holistic view of machine trains and the systems they are associated with.

A holistic view of a machine system means understanding the following:

- Actual process conditions: such as process pressures, temperatures, fluid densities, fluid compositions, and flow rates.
- Control schemes: level control, flow control, pressure control, temperature, or other?
- Machine construction details: bearing design, lubrication system details, seal design, etc.
- Machine history: Installation date, last repair or overhaul, and last upgrade.
- Current overall mechanical condition of the machine: Are there any vibration or bearing temperature levels in alarm? Is the compressor performing as expected?
- Known issues: Have any leaks, alarms, noise, etc., been reported? Have there been any recent unexplained trips? Are there open works orders?

A holistic view allows machinery professionals to fully understand how machinery systems are designed to function and where potential problems could possibly be lurking.

In the “olden days,” a machinery professional would take a little bit of data and then make the call. The “new paradigm” requires that the machine evaluator:

Table 10.1 Key troubleshooting questions.

Question	Description
Is it the machine?	Design flaw or assembly issue
Is it the overall system?	Actual operation condition compared to design specifications
Is it the process?	Any variation in the process conditions, such as fluid properties, pressure, or temperatures
Is it a subsystem?	Lubrication systems malfunction or control issue
Is it the installation?	Foundation, grout, or misalignment problem
Is it a secondary element to the machine?	Loose or improperly installed non-functional connector, such as piping or bolt

- View the installed machine as a system
- Talk to the operators, mechanics, control specialist, etc.
- Collect the right data
- Understand what the data is saying
- Know the risks involved

The second item on the list, talking with people, especially those people involved with the day to day upkeep of the machine is important in that you want people to be comfortable talking with you when a situation does arise. Ultimately, the troubleshooter is trying to evaluate the basic questions in Table 10.1:

Try to keep an open mind during the investigation phase of your analysis so that you obtain a “30,000 ft” view of the problem before focusing in on the likely cause of the problem.

10.1 The Field Troubleshooting Process—Step by Step

The basic troubleshooting steps that should be taken are: 1) Define the problem, 2) Collect data, 3) Analyze, and 4) Act and Confirm. Following this methodology, you can improve your chances of quickly finding the root of a machinery problem in order to implement the proper solution. The four troubleshooting steps are explained below.

10.1.1 Step 1: Define the Problem

First you need to develop a preliminary problem definition. This step requires that you talk to the machine operator (or owner) about the machinery issue at hand. Ask:

- a. What seems to be the problem?
- b. Where does the problem manifest itself?
- c. When was the problem first noticed?

At this time, you are trying to get a basic idea of the nature of the problem.

10.1.2 Step 2: Collect All Pertinent Data

Now that you have decided to proceed with the analysis, you must first decide who knows the most about the problem. Some potential interviewees are:

- a. Process engineers
- b. OEM
- c. Control engineers
- d. Mechanics

Compile a list of all the interviewees along with date and time you plan to perform the interviews. Make sure you take detailed notes during the interviews.

10.1.3 Step 3: Analyze the Body of Data as a Whole

1. Look through all the process data, vibration data, temperature data, performance data, etc., and look for changes, trends, and correlations that may provide clues as to what is going on. Carefully analyze the body of data as a whole.
2. If it is still not clear what the problem is, generate a list of plausible theories that can explain all the symptoms. (At this point, the more possible causes you can postulate the better your chances are of finding the root cause.) The following techniques can be helpful in paring down the possible causes to determine the most likely cause:
 - a. Field testing or trials to support or refute some of the theories.
 - b. Analytical modeling, such as rotordynamic or computational fluid dynamic modeling to validate certain theories.
 - c. "Cause mapping" to evaluate all potential causes

10.1.4 Step 4: Act and Confirm

1. Once you have decided to act, here are your choices:
 - a. Make no immediate operating changes or maintenance adjustments but continue to monitor the machine's condition.
 - b. Test and assess: 1) Vary operating condition speed, flow, pressures, suction level, etc., and observe how the machine responds. 2) Check for off-design machine operation due to off-design process conditions, open bypasses, restrictions, control issues, etc. Correct any simple problems such as open bypass lines, low suction levels, and off-design conditions as you find them. If

the problem can be resolved with simple field adjustments, then you are done.

- c. If you determine that you are dealing with a system issue that cannot be corrected on-line, you will need to shut down. Here are few examples of issues that can only be corrected:
 - i. Clearing line blockages
 - ii. Repairing leaking valves
 - iii. Replacing or adjusting troublesome control valves
 - iv. Cleaning fouled coolers
- d. If you determine that you are dealing with a machine issue, here are your options:
 - i. Switch to spare, if available.
 - ii. Plan a shutdown for balancing, alignment, oil change out, piping modifications, regrouting, etc.
 - iii. Plan a shutdown for repair (i.e., full machine overhaul if needed). Reduce stresses if possible while waiting for repair by reducing process induced loads.
 - iv. Shut down immediately for balancing, alignment, oil change out, piping modifications, regrouting, etc.
 - v. Shut down immediately for repair.
2. You should always follow up with a field audit in order to confirm that the root cause of the problem has been addressed and that machine conditions are back to normal. If the problem has been solved, you are finished; if not, go back to the “Collect all pertinent data” step and continue the investigation.
3. If the problem is solved, take some time to document the final results. Place a copy of the final report in the equipment file and send copies to coworkers, team members, and management.



Figure 10.1 Hourglass approach. Funnel—pertinent data collection. Focus—analysis, decide, act and confirm. Think globally and then act by applying investigation findings to similar machines.

10.2 The “Hourglass” Approach to Troubleshooting

When approaching a new field troubleshooting problem, it is always helpful to visualize an approach with the shape of an hourglass (see Figure 10.1). During the initial phase, you should

maintain a broad view of the problem, similar to the top of an hourglass. Avoid zeroing in on any one cause initially and consider all possibilities. Focusing in on only one possible cause too early can derail the investigation and waste time and money.

Next, focus in on the root cause by gradually eliminating possible causes. Once the most likely cause is determined, it's time to act and confirm. During this time, you are in the neck of the hourglass. Once you have confirmed that the problem is solved, you are ready to move to the bottom of the hourglass.

World-class organizations take advantage of the learnings garnered from troubleshooting efforts by applying the findings across their organizations. The key question to ask yourself at the end of an investigation is: Can this learning (or learnings) be applied elsewhere in the organization? By thinking globally, the organization can avoid a similar issue, by acting preemptively.

10.3 Thinking and Acting Globally

Every machine problem, failure, or issue an organization experiences is an opportunity to learn something new about their machines and processes. We can capitalize on these experiences by thinking globally in order to apply our learnings across the organization. Here are a few ways to get the word out:

- Company-wide bulletins
- Modification of existing operating or maintenance procedures
- Modification of purchasing specifications
- Information sharing at in-house group meetings

Sharing your learnings throughout the company maximizes everyone's time and effort, while minimizing valuable

resources. It is very rewarding to uncover and correct previously hidden issues that may be present in your plant before they can lead to dire consequences. Your management will greatly appreciate all the positive effects realized across the site due to your troubleshooting efforts.

10.4 Troubleshooting Matrix and Table

The purpose of the centrifugal compressor troubleshooting matrix and table is to assist field troubleshooters in the identification and resolution of the following categories of process machinery problems. Keep in mind that the following troubleshooting matrix and table contains only the more common symptoms and causes found in the field. These matrixes should be considered a starting point for your analysis. You may need to contact the manufacture for assistance in solving rarely encountered or subtle problems.

10.5 Centrifugal Compressor Troubleshooting Example

Here's an example illustrating how the centrifugal compressor matrix can be used. Let's assume you are experiencing a high-power load on an electric motor driver. You first select the column titled "Power too high" and then write down all the "Common Root Causes" for further review (see Table 10.1a). The matrix indicates that 1) an incorrect impeller diameter, 2) the incorrect impeller speed, 3) suction pressure too low, 4) back pressure too high, 5) plugged suction strainer or piping, and so on. There are 11 possible causes in all. Next, you systematically pare down the list by determining which potential causes are unlikely based on the available information and data collected. Eventually you will be left with the most likely cause, or root cause, of your problem.

Table 10.1a Centrifugal compressor troubleshooting matrix example.

		Common Symptoms									
Instructions: First find the “Common Symptom” column that best describes your field condition and then follow the column down to find all the possible causes (denoted by the symbol “X”). The “Common Root Causes” identified with an “X” are all the possible causes that should be investigated to determine the actual root cause of the problem. Additional troubleshooting advice can be found in Table 10.1c.		Flow too low	Flow too high	Unstable or erratic flow	High discharge temperature	Discharge pressure too high	Discharge pressure too low	High pressure pulsations	Power too high (1)	Power too low (1)	High vibrations or noisy (2)
Common Root Causes	a. Incorrect impeller diameter(s)	X	X		X	X	X	X	X	X	
	b. Incorrect rotor speed	X	X		X	X	X	X	X	X	X
	c. Impeller(s) installed backwards or rotor turning backwards	X					X			X	
	d. Back pressure too high	X		X	X	X		X	X	X	X
	e. Suction pressure too low	X			X		X		X	X	X
	f. Plugged suction strainer or piping	X		X	X		X		X	X	
	g. Gas density lighter than design	X					X	X		X	
	h. Gas density higher than design		X		X	X		X	X		
	i. Flow much less than design			X	X	X		X	X	X	X
	j. Flow much higher than design			X			X	X	X	X	X
	k. Open or leaking spillback valve	X					X		X	X	
	l. Rotor imbalance										X
	m. Compressor to driver misalignment										X
	n. Suction temperature higher than design	X			X		X			X	
	p. Excessive internal compressor wear	X			X		X		X	X	X
	q. Driver issue or malfunction. Go to “Electric Motor” or “Steam Turbine” troubleshooting tables for more guidance	X	X	X	X	X	X				
	r. Surge control system malfunction	X	X	X	X	X	X	X	X	X	X

(1) The potential root causes for these two columns only apply if horsepower increases with flow. These column headings must be reserved if horsepower decreases with flow. Refer to the “How Horsepower Varies with Flow” at the end of this section for more information on this topic.

Table 10.1b Centrifugal compressor troubleshooting matrix

		Common Symptoms									
Instructions: First find the “Common Symptom” column that best describes your field condition and then follow the column down to find all the possible causes (denoted by the symbol “X”). The “Common Root Causes” identified with an “X” are all the possible causes that should be investigated to determine the actual root cause of the problem. Additional troubleshooting advice can be found in Table 10.1c.		Flow too low	Flow too high	Unstable or erratic flow	High discharge temperature	Discharge pressure too high	Discharge pressure too low	High pressure pulsations	Power too high (1)	Power too low (1)	High vibrations or noisy (2)
Common Root Causes	a. Incorrect impeller diameter(s)	X	X		X	X	X	X	X	X	
	b. Incorrect rotor speed	X	X		X	X	X	X	X	X	X
	c. Impeller(s) installed backwards or rotor turning backwards	X					X			X	
	d. Back pressure too high	X		X	X	X		X	X	X	X
	e. Suction pressure too low	X			X		X		X	X	X
	f. Plugged suction strainer or piping	X		X	X		X		X	X	
	g. Gas density lighter than design	X					X	X		X	
	h. Gas density higher than design		X		X	X		X	X		
	i. Flow much less than design			X	X	X		X	X	X	X
	j. Flow much higher than design			X			X	X	X	X	X
	k. Open or leaking spillback valve	X					X		X	X	
	l. Rotor imbalance										X
	m. Compressor to driver misalignment										X
	n. Suction temperature higher than design	X			X		X			X	
	p. Excessive internal compressor wear	X			X		X		X	X	X
	q. Driver issue or malfunction. Go to “Electric Motor” or “Steam Turbine” troubleshooting tables for more guidance	X	X	X	X	X	X				
	r. Surge control system malfunction	X	X	X	X	X	X	X	X	X	X

(1) The potential root causes for these two columns only apply if horsepower increases with flow. These column headings must be reserved if horsepower decreases with flow. Refer to the “How Horsepower Varies with Flow” at the end of this section for more information on this topic.

Table 10.1c Centrifugal compressor troubleshooting tips.

Instructions: First find the "Common Symptom" column that best describes your field condition and then follow the column down to find all the possible causes (denoted by the symbol "X"). The "Common Root Causes" identified with an "X" are all the possible causes that should be investigated to determine the actual root cause of the problem. Additional troubleshooting advice can be found in Table 10.1c.		Centrifugal compressor field troubleshooting tips 
Common root causes	a. Incorrect impeller diameter(s)	1. Check repair records to see if there is any written record of the installed impeller diameter(s) and then compare it to the design diameter(s). 2. Check the speed to make sure you are at the rated speed before pulling the rotor to check impeller diameter(s).
	b. Incorrect rotor speed	Check the rotor speed indication if installed or by using a tachometer and compare with design speed. A speed issue can be the result of an erroneous input speed signal to a VFD.
	c. Impeller(s) installed backwards or rotor turning backwards	1. First check that you have the correct driver rotation. 2. If the driver rotation is correct, the speed is correct, and the pump differential is only a fraction of the rated differential, then you should consider pulling the rotor to see if the impeller(s) are installed backwards.
	d. Back pressure too high	1. Check for a flow obstruction, such as a partially closed valve or a plugged strainer, in the discharge line. If there is a strainer in the discharge line, pull the strainer to see if it is clean. Check with operator to see if there has been a history of line pluggage. 2. Check the downstream system pressure to see if it is higher than normal due to a change in the process or an upset condition.

(Continued)

Table 10.1c Cont.

e.Suction pressure too low	1. Check the upstream system pressure to see if it is lower than normal due to a change in the process or an upset condition. 2. Check for flow obstruction in suction piping.
f.Plugged suction strainer or piping	1. Check for flow obstruction in the piping. 2. Check any strainers in the suction and discharge line to see if they are plugged. 3. Ask the operator if the compressor piping has a history of pluggage.
g.Gas density lighter than design	Pull a representative process gas sample in order to determine its molecular weight. The compressor differential pressure and amp draw are directly proportional to the molecular weight of the gas. Watch out for significant deviations from the normal molecular weight of the gas during start-ups.
h.Gas density higher than design	Pull a representative process gas sample in order to determine its molecular weight. The compressor differential pressure and amp draw are directly proportional to the molecular weight of the gas. Watch out for significant deviations from the normal molecular weight of the gas during start-ups.
i.Flow much less than design	1. Check the flow and determine if you are significantly below (<80%) of the design flow. 2. If the compressor has a surge control system, check to see it is functioning properly. 3. If the compressor doesn't have a surge control system, perform a test by temporarily increasing the flow through the compressor. If symptoms disappear, you may need to review the compressor sizing and piping system to understand the source of the problem.
j.Flow much higher than design	1. Check the flow and determine if you are significantly above (>120%) of the design flow. 2. Perform a test by temporarily decreasing the flow through the compressor. If symptoms disappear, you may need to review the compressor sizing and piping system to understand the source of the problem.

(Continued)

Table 10.1c Cont.

	k.Open or leaking spillback valve	1. Check all potential bypass valves to ensure they are closed or are not leaking. 2. If the compressor has a surge control system, make sure it is working properly. 3. Check relief valves with a ultrasonic gun for any indications of leakage.
	l.Rotor imbalance	1. The predominate vibrational component will be 1x running speed if imbalance is truly the problem. 2. The horizontal 1x component should be approximately equal to the vertical 1x component.
	m.Compressor to driver misalignment	1. Usually this type of misalignment will show up as 1x or 2x vibrational components in the spectrum. High axial vibration is another indication of compressor to driven misalignment. 2. Start by checking the driver to compressor alignment if vibration analysis suggests misalignment is present. 3. Misalignment can be the result of piping strain. If driver to machine alignment looks okay, then check piping fit-up for possible pipe strain issues.
	n.Suction temperature higher than design	Check to see if the compressor suction temperature is deviating from the design value by more than 5%. A deviation greater than 5% will affect horsepower and performance.
	p.Excessive internal compressor wear	1. Begin by checking compressor's performance in the field to determine if degradation has occurred. Plot a few performance points of the performance map to see if the compressor is on its curve. 2. Compare the actual discharge temperature to the design discharge temperature. The discharge temperature will begin to rise above the design discharge temperature as internal compressor wear occur.
	q.Driver issue or malfunction. Go to "Electric Motor" or "Steam Turbine" troubleshooting tables for more guidance	1. Failing to reach rated speed is an indication of a driver issue. 2. Erratic speed is another indication of driver control issue. 3. It is a good idea to have the OEM on-site whenever steam turbines or electric motors with VFDs are first commissioned.

(Continued)

Table 10.1c Cont.

	r.Surge control system malfunction	The combination of low flow, low pressure differential, and high horsepower indications usually point to an open spillback valve. If a surge control system is installed, it is possible that the surge valve is leaking or open. First check if the valve is open. If it is, determine if it's due to a malfunctioning surge control system or to a malfunctioning valve. If in doubt, have the surge control system designer double check the surge algorithms to ensure they are correct.
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11

Reciprocating Compressors

Reciprocating compressors (see Figure 11.1) represent a class of gas compressors that raises the pressure of a gas by means of a tightly fitting reciprocating piston inside a cylindrical chamber, reducing volume from intake to discharge, in order to generate flow. Check valves within the cylinder are used to control the flow of gas within the cylinder.

Reciprocating compressor components include:

1. A **piston** with rings and rider bands, cross head, and piston rod, which moves back and forth with each rotation of the crankshaft (see Figure 11.2). The rings act like seals to keep gas from bypassing the piston. Rider bands are wide ring-shaped devices that support the piston in the cylinder and prevent the two from touching. Some reciprocating compressors are “single acting” and some are

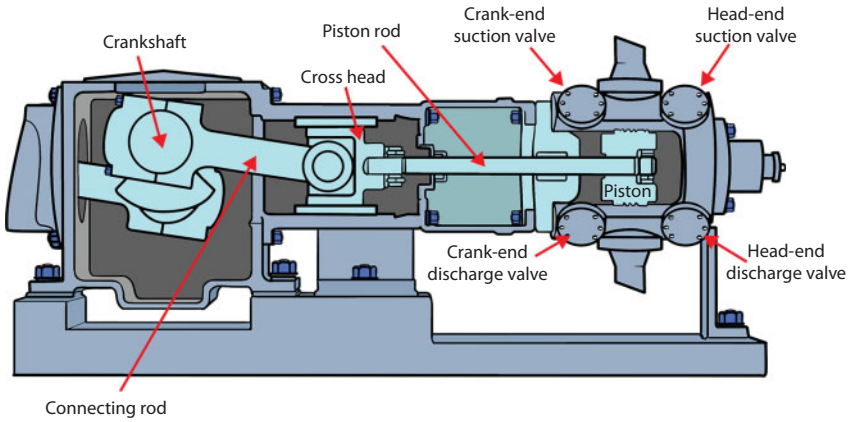


Figure 11.1 Reciprocating compressor cylinder cross section.



Figure 11.2 A crankshaft is a subcomponent of a reciprocating compressor that converts rotary motion into reciprocating linear piston motion. (Also known as a *Crank*).

“double acting”. Single-acting reciprocating compressors compress gas only when the piston is moving in one direction. When the piston moves toward the cylinder head the gas is being compressed but when the piston is moving away from the head, suction is pulling gas into the cylinder. (Read more about the single action compression cycle in Figure 11.3.)

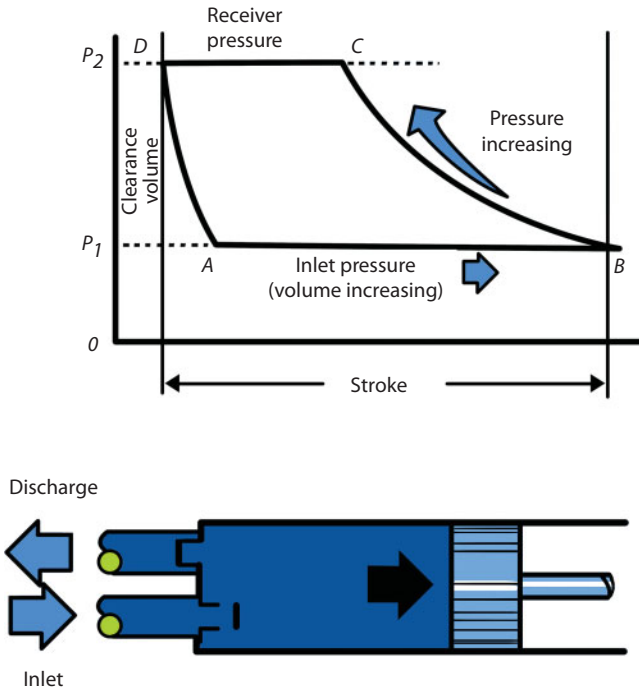


Figure 11.3 The upper plot, comprised of points A, B, C, D, shows how the pressure changes inside a single-acting compressor cylinder as the piston moves through one compression cycle. The lower image shows an idealized single-acting compressor cylinder. Here is a brief description of what's happening during one compression cycle: 1) From point A to point B, the piston moves to the right until the inlet valve opens, allowing gas to fill the gradually increasing internal volume (as shown in the lower image). 2) From point B to point C, the piston has completed a full suction stroke and begins to move to the left until enough pressure is produced to fully close the inlet valve. As the piston continues moving to the left, pressure steadily builds inside the cylinder until it exceeds the discharge pressure, forcing the discharge valve to open. 3) From point C to point D, the discharge valve is fully open, permitting gas to flow through it while the piston completes the discharge stroke. 4) From point D to point A, the piston begins moving to the right again, which drops the pressure inside the cylinder to the point the discharge valve closes. With the discharge valve fully closed, the cylinder pressure gradually drops to the point the inlet valve is open, drawing fresh gas into the cylinder. At this point, a single compression cycle is complete. Note: In double-acting compressor cylinders, this same single-acting compression cycle described here occurs at both ends of a compressor cylinder.

Double-acting compressor cylinders compress gas on both movements of the piston. When the piston is moving toward the head of the cylinder it is compressing gas as well as putting it into the process. At the same time the piston is on the suction stroke of the crankcase half of the cylinder. When the piston begins to move toward the crankshaft the crankshaft half of the cylinder is on compression and the head end is on suction. The effect is that there is almost twice the output from a double-acting cylinder as from a single-acting one.

2. A compressor **cylinder** contains the internal pressure and supports the reciprocating piston. The cylinder also has a head on each end with water passages to remove some of the heat of compression. The outer head may also include something called a clearance pocket for adjusting compressor output. The piston rod head has high-pressure packing in it to keep the pumped gas inside the cylinder.
3. A set of **suction and discharge valves**. Reciprocating compressor valves are essentially check valves (see Figure 11.4). The suction valves permit flow into the cylinder but not back out, and discharge valves permit flow to exit the cylinder but not return back in. Suction and discharge valve failures are the most common component failure in a reciprocating compressor. Unloaders may be part of the suction valve assembly.

There are three ways of controlling the output of a reciprocating compressor that is motor driven (constant speed):

- i. reducing suction pressure;
- ii. adding a clearance pocket to the cylinder by opening and closing the valve that separates the

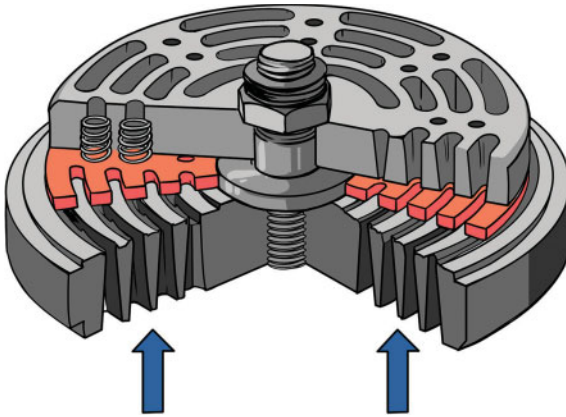


Figure 11.4 There are many different types of reciprocating compressor valves. The one shown above is plate type compressor valve. Flow can only move through the valve upward, i.e., in the direction of the arrows. Like a check valve, if flow tries to reverse, the plate will seat and prevent reverse flow.

cylinder from the additional volume in the clearance pocket;

- iii. activating unloaders on the intake valves, which will hold them open.

4. A **crank shaft, crank case, and connecting rods.**

Some reciprocating compressors have two lube systems, one for the crank shaft and cross head and another for the cylinder and packing. The lubricant may be the same in both lubricating systems, but in some cases is different. Be careful not to mix or switch the two oils if they are different.

If all the compression components are working properly, the flow through a compressor cylinder is controlled by 1) the compressor speed, 2) the displacement of the piston within the cylinder, and 3) the suction pressure. Increasing either the compressor speed, cylinder displacement, or suction pressure will increase the net flow through the cylinder and the horsepower required from the driver. However, the design flow through a compressor

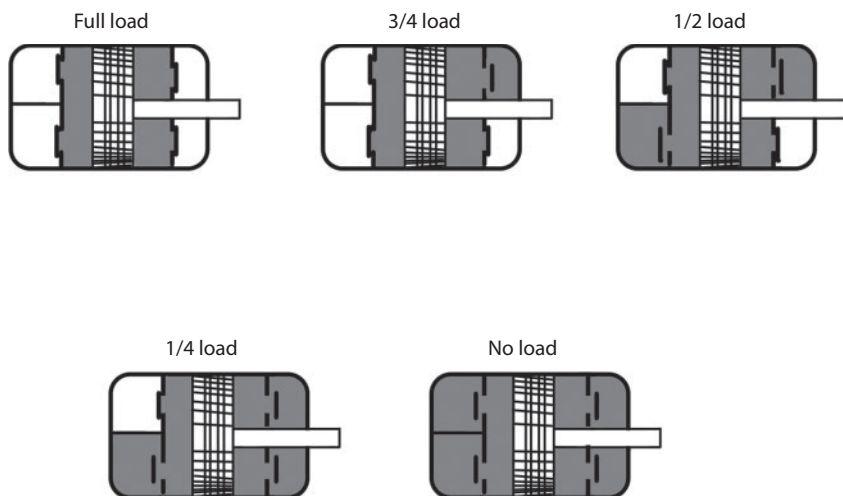


Figure 11.5 Valve unloaders can be used to partially unload or fully unload a reciprocating compressor cylinder. At the upper left, all cylinder valves are fully functional, so the cylinder has a 100% flow capability. The cylinder in the center of the upper row has a single valve (the upper, crank end valve) held open by an unloader, which reduces the cylinder's capacity to 75% of full load. As more and more valves are held open, the cylinder capacity drops accordingly. If all the valves are held open with an unloader, as seen in the lower right, the cylinder's capacity drops to zero.

cylinder can be reduced by numerous factors, such as internal leakage at the piston rings or valves, preheating of the inlet flow because of high ambient temperatures, a fouled exchanger on the suction gas, a lower suction pressure, etc.

Reciprocating compressors are often equipped with valve unloaders (Figure 11.5) and clearance pockets (Figure 11.6) as a means of controlling compressor throughput. If these flow control components malfunction, compressor flow may decrease dramatically or even drop to zero. Unloaders are control elements designed to hold intake cylinder valves open (Figure 11.7) when they are activated, so that no gas can be trapped or compressed inside the cylinder during the compression cycle. Therefore, any compressor cylinders with activated unloaders will not able

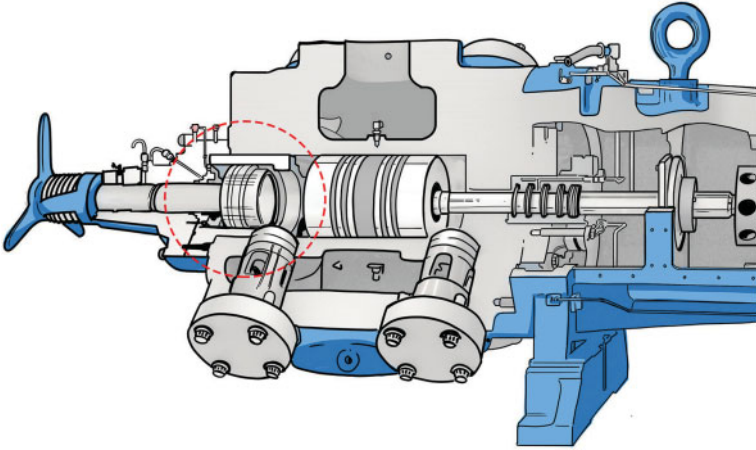


Figure 11.6 A volume unloader can either increase the volumetric efficiency (VE) of a cylinder by decreasing the head-end volume or decrease the VE by increasing the head-end volume.

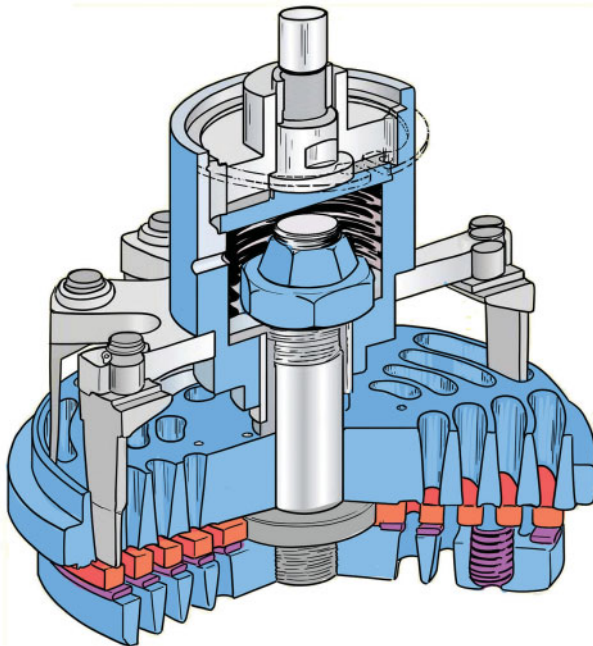


Figure 11.7 Finger type unloaders hold a valve open to prevent the valve from working normally.

to contribute any flow to the compressor's total flow. The more unloaders that are activated the lower the total flow will be. Unloaders may be operated independently or in groups to produce the desired number of compressor "load steps." In contrast, clearance pockets are used to add or remove internal volume within a compressor cylinder, which in turn adjusts when the intake valves open and varies throughput. Generally, as flow is decreased with the use of unloaders and/or clearance pockets, the required horsepower decreases. Conversely, as flow is increased with the use of unloaders and/or clearance pockets, the required horsepower increases.

11.1 Reciprocating Compressor Installations

Before attempting to troubleshoot your reciprocating compressor, you must understand its design details. You must ask: 1) How is the piping designed? 2) Are there any bypasses? 3) Are there any coolers? 4) Are the controls automatic or manual. Remember that the devil is in the details.

Most reciprocating compressors have one or more coolers to extract heat generated by the heat of compression. It's important to know where the coolers are located and how they are arranged. Most coolers are configured so that the discharge gas from each staged is cooled before preceding to the next stage. If there are spillback lines, their takeoffs are normally downstream of a cooler or coolers, which enables operators to spillback indefinitely without fear of shutting down due to a high gas temperature trip.

A typical piping arrangement for multistage reciprocating compressor (shown in Figure 11.8) includes:

1. Suction scrubbers on each stage to remove liquids and solids from the gas before entering any of the compressor cylinders

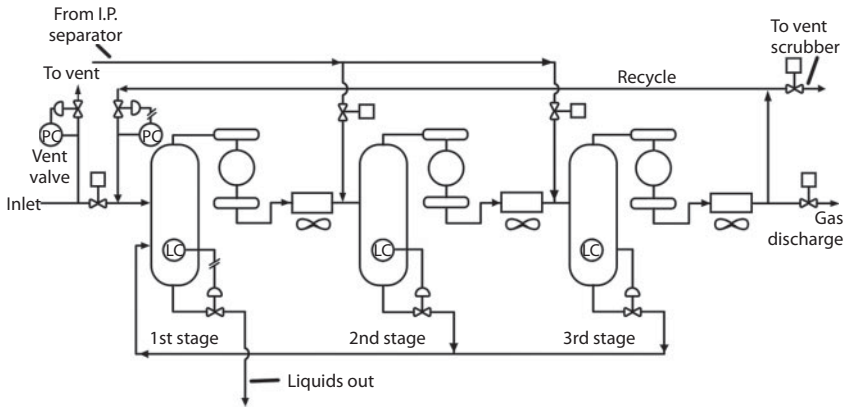


Figure 11.8 Multistage reciprocating compressor piping arrangement.

2. Coolers on the discharge of each stage to provide cool gas to next stage of compression and to allow recycling during start-up or upset conditions
3. A recycle line with valve to allow recycling during start-ups or provide pressure control

Normally you need to start up a reciprocating compressor with the spillback fully open and the cylinders fully unloaded. Depending on your time constraints and loading step options, you should gradually move from a fully unloaded conditions to a partial cylinder load condition. Once you have established flow through the spillback line, wait until everything looks normal before closing the spillback line. On closing of the spillback line, you should obtain forward flow and begin to load the compressor.

When starting up, you need to know how to put the compressor into a bypass mode to unload the compressor. Remember that if you have trouble developing flow after start-up, you might need to check to see if the bypass line is open and robbing you of system flow.

11.1.1 How Process Conditions Affect Reciprocating Compressor Performance

Changing the operating conditions from their original design values can significantly impact a compressor's actual capacity, horsepower, and discharge temperature. Some factors affecting performance changes are the compressor speed, suction and discharge pressure and temperature, gas composition, suction and discharge piping design, and compressor valve design. The table below summarizes how different process variables can affect field performance. Keep these relationships in mind when evaluating your reciprocating compressors in the field.

Troubleshooting tips:

- If the differential pressure (i.e., the difference between the discharge pressure and the suction pressure) across any compression stage increases, the discharge temperature will increase on that stage. Conversely, if the differential pressure across any compression stage decreases, the discharge temperature will decrease on that stage.
- If the compressor speed increases, the net flow will increase. The horsepower required by a compressor increases at higher speeds, which should be reflected by increasing amps or kilowatts. Conversely, if the compressor speed decreases, the net flow will likely decrease. The horsepower required by a compressor at lower speed will decrease, which should be reflected by decreasing amps or kilowatts. This relationship between compressor speed and horsepower is not seen on motor driven compressors unless the motor is equipped with a variable speed electrical drive.
- If the molecular weight of the gas being compressed increases, the required horsepower will also increase. Conversely, if the molecular weight

Table 11.1 How process conditions affect reciprocating compressor performance. Note: The results in this table assume that only one variable is changing at a time.

Process modification	Flow (at standard conditions)	Discharge temperature	Gas horsepower
Speed increase	Increases	Increases	Increases
Speed decrease	Decreases	Decreases	Decreases
Increase in suction pressure	Increases	Decreases	Increases
Increase in suction temperature	Decreases	Increases	Decreases
Decrease in suction temperature	Increases	Increases	Increases
Decrease in suction pressure	Decreases	Increases	Decreases
Decrease in the ratio of specific heats (k)	Decreases	Decreases	Decreases
Increase in the ratio of specific heats (k)	Increases	Increases	Increases

of the gas being compressed decreases, the required horsepower will decrease.

- If the suction pressure increases, the mass flow and required horsepower will also increase due to an increase in the gas density. Conversely, if the suction pressure decreases, the mass flow and required horsepower will decrease. If a higher suction pressure puts more gas into the compressor,

more horsepower will be required to achieve the same pressure.

- If a multistage compressor begins to lift a relief valve between stages, the stage downstream of the relief device needs to be checked as it is not “keeping up” with the previous stage.
- If throughput falls quickly, check to insure all unloaders on intake valves are functioning correctly and not allowing the intake valves to close. If clearance pockets are used to control compressor throughput normally, insure the valve that controls their addition or removal from the cylinder volume is closed to separate them from the cylinder.

These relationships will help to troubleshoot reciprocating compressors that are not performing as expected. When reporting a problem with a compressor, always list the symptoms that caught your attention, such as a low flow, high amps, knocking sound, vibration, not enough pressure, etc.

11.2 Reciprocating Compressor Start-Ups

Most large reciprocating compressors have a jacking or barring gear to allow the compressor to be turned over to insure that there is no liquid in any of the cylinders prior to starting. The compressor should be barred over at least twice before attempting to start a compressor. Reciprocating compressors are normally started with an open spillback line to minimize the process load on the driver. This method allows the circulation of some gas, while rotating the compressor and driver under a light load. Keep in mind that every compressor is unique in its design and construction, and therefore should have its own customized procedure to ensure all necessary steps are followed during start-ups. Only approved procedures that follow

the manufacturer's recommendations and experience with this compressor in this application should be used—attention to detail is paramount.

Once full load conditions (i.e., speed, temperature, pressure, and flow) are reached, use the Reciprocating Compressor Checklist below to monitor machine condition and performance. Continue to monitor compressor conditions for at least an hour to make sure that everything is normal. If problems are encountered during this start-up phase, either call in help for a second opinion or shut down until additional technical assistance is available.

11.3 Reciprocating Compressor Checklist

- Net flow through the compressor. If the flow is low, check for open bypass valves and, if unloaders are used, if they are functioning correctly. If the flow is too high, check the compressor speed and suction pressure.
- Suction and discharge pressure. Are the pressures normal? Remember, when troubleshooting, check the easy things such as valve positions, compressor speed, etc., first. Ideally there are pressure gauges on the suction and discharge pressure gauges and in-between stages.
- Suction and discharge temperatures. Are the temperatures normal? If not, begin to troubleshoot the system to find out what has changed. If the suction temperature is normal but the discharge temperature is high, you may be looking at an indication of internal wear or fouling valves. Use an infrared gun, contact pyrometer, or camera to check for hot valves.

- Oil pressures and levels. Check both crankcase oil and cylinder lube in case they are different. Watch for increasing crankcase levels which normally means water is entering the lubricant.
- Bearing temperatures. Are the bearing temperatures higher than normal? If temperatures are found to be elevated, first check over the lubrication system to ensure flows are normal and that supply oil temperatures are normal.
- Vibration of piping, compressor casing, or skid vibration. Are vibration levels normal? If not, have someone from the vibration department analyze the situation to determine what may be wrong.
- Packing leakage. Check for high pressure and ensure that distance pieces are not filling with liquid. Make sure liquid traps are working properly.
- Drainage traps. If there are “knock outs” between stages with traps for draining them, notice how much liquid is draining. If it is possible that liquid is entering a reciprocating compressor cylinder it can wreck the compressor completely in a very short time.

Common causes of problems to watch out for:

- **Abnormal rotational speed** (This is generally not a problem if the compressor is motor driven because motors are constant speed devices.)
- **Abnormal suction pressure.** Low pressure can be caused by a partially opened suction valve or by blockages upstream of the first stage. High suction pressure can be caused by a control valve problem that regulates the pressure to the compressors first stage.

- **Abnormal discharge pressure.** A high discharge pressure can be caused by a blockage downstream. A low discharge pressure can be caused by one of the preceding stages not compressing properly or low inlet pressure to the first stage.
- **Abnormal gas density**—especially if process upsets that can change the gas composition have been known to occur during previous start-ups.
- **Internal gas slippage** due to leaking piston rings
- **Plugged suction strainer** that can result in a drop in flow or a loss of pressure.
- **Discharge restriction** due to downstream plugging or a partially closed valve.
- **Broken valves** will be evident by being much hotter than other valves. This condition can be caused by a bad valve with broken internals, a broken valve gasket, or a bad seat.
- **Fouled compressor valves** due to dirty gas will generally affect the output of the cylinder with the fouled valves.
- **Unusual gas temperatures** due to a loss of cooling somewhere in the process, usually caused by fouling or obstruction. Check intercoolers and after-coolers on both the process side and water side.

11.4 Criticality

Compressors of all classes and designs should be monitored closely because they tend to be high-energy, high-horsepower machines. In many cases, they are unspared, which makes them highly critical to the processes they support. When they fail, management takes notice.

Because compressors tend to represent the most critical machines at most sites, it makes sense to watch them closely

and keep them maintained and operating efficiently. Here are just a few ways to ensure your compressors perform safely and reliably day in and day out:

1. Perform daily inspections.
2. Check compressor performance on a regular basis. Keep process logs in order to know what is normal. Look at the logs; historical process logs will help you know how your compressors were performing yesterday, last week, and last month.
3. Analyze the lubricating oil regularly by checking levels and color of oil. Increasing oil level, in the crankcase, can mean water building in the crankcase.
4. Watch rod drop indicators and other indicators of potential problems on a daily basis.
5. If a divider block is used for lubrication, check for any indicator pins that may be popping out, indicating blocked lubrication to a cylinder or packing.
6. If there is a panel for high-pressure packing leakage, monitor it for increased flow. Check flow on sweeps of distance pieces.
7. Operators should help perform regular maintenance as prescribed by the manufacturer when appropriate or during turnarounds in order to learn more about a compressor's internal construction. This type of hands-on knowledge is invaluable when troubleshooting and can make you feel more comfortable when inspecting and operating these machines.

12

Troubleshooting Reciprocating Compressors in Process Services

Here is a question to ponder: Can a reciprocating compressor operate within expected operating parameters without a complete system? The answer is clearly no. Without a continuous suction gas supply, a driver, such as an electric motor, engine, etc., some type of control system, and a piping system, a reciprocating compressor could never function as it was designed. The various components in a gas compression system cannot function independently; all its subsystems must function together to meet the process needs.

If one element of this compression is malfunctioning, the whole system will suffer. Here are some possible system issues that can lead to serious consequences:

- Suction strainer is partially plugged
- Discharge piping has some unexpected blockage
- Suction or discharge piping is undersized or oversized
- Spillback valve is leaking back to the compressor suction
- The suction pressure is significantly lower than the design value
- Compressor driver is undersized
- Foundation is too flimsy
- Piping resonance is present due to lack of supports
- Oil coolers are undersized
- Relief valve is leaking
- Vibration sensors are out of calibration

Even though any of these non-machinery issues will usually lead to major problems, it would be no surprise if one of the resulting symptoms was reported as a “compressor problem.”

Similarly, here are some issues with subsystems that could lead to undesirable symptoms in other machine classes:

- Low suction pressure on a reciprocating compressor will increase the overall compression ratio and lead to high discharge temperatures.
- Low gas density could cause reciprocating compressor flow and differential pressure to both run below design levels.
- Scaling or a flow blockage in an oil cooler would cause bearing temperatures to rise due to a higher than normal oil supply temperature.

- Process changes could cause gas properties to vary significantly from the original design condition. A change in gas properties could result in a number of problems, such as driver overload, low of flow, surging, etc.

To successfully assess and troubleshoot process machinery, owner/operators must take a holistic view of machine trains and the systems they are associated with.

A holistic view of a machine system means understanding the following:

- Actual process conditions: such as process pressures, temperatures, fluid densities, fluid compositions, and flow rates.
- Control schemes: level control, flow control, pressure control, temperature, or other?
- Machine construction details: the number of throws, number of stages, valve loading schemes, lubrication system details, interstage cooling design, etc.
- Machine history: Installation date, last repair or overhaul, and last upgrade.
- Current overall mechanical condition of the machine: Are there any vibration or bearing temperature levels in alarm? Is the compressor performing as expected?
- Known issues: Have any leaks, alarms, noise, etc., been reported? Have there been any recent unexplained trips? Are there open works orders?

A holistic view allows machinery professionals to fully understand how machinery systems are designed to function and where potential problems could possibly be lurking.

Table 12.1 Basic troubleshooting questions.

Question	Description
Is it the machine?	Design flaw or assembly issue
Is it the overall system?	Actual operation condition compared to design specifications
Is it the process?	Any variation in the process conditions, such as fluid properties, pressure, or temperatures
Is it a subsystem?	Lubrication systems malfunction or control issue
Is it the installation?	Foundation, grout, or misalignment problem
Is it a secondary element to the machine?	Loose or improperly installed non-functional connector, such as piping or bolt

In the past, a machinery professional would take a little bit of data and then make the call. The “new paradigm” requires that the machine evaluator:

- View the installed machine as a system
- Talk to the operators, mechanics, control specialist, etc.
- Collect the right data
- Understand what the data is saying
- Know the risks involved

The second item on the list, talking with people, especially those people involved with the day to day upkeep of the machine is important in that you want people to be comfortable talking with you when a situation does arise. Ultimately, as a troubleshooter you are trying to evaluate the basic questions in Table 12.1.

Try to keep an open mind during the investigation phase of your analysis so that you obtain a “30,000 ft” view of the problem before focusing in on the likely cause of the problem.

12.1 The Field Troubleshooting Process—Step by Step

The basic troubleshooting steps that should be taken are: 1) Define the problem, 2) Collect data, 3) Analyze, and 4) Act and Confirm. Following this methodology, you can improve your chances of quickly finding the root of a machinery problem in order to implement the proper solution. The four troubleshooting steps are explained below.

12.1.1 Step 1: Define the Problem

First you need to develop a preliminary problem definition. This step requires that you talk to the machine operator (or owner) about machinery issue at hand. Ask:

- a. What seems to be the problem?
- b. Where does the problem manifest itself?
- c. When was the problem first noticed?

At this time, you are trying to get a basic idea of the nature of the problem.

12.1.2 Step 2: Collect All Pertinent Data

Now that you have decided to proceed with the analysis, you must first decide who knows the most about the problem. Some potential interviewees are:

- a. Process engineers
- b. OEM
- c. Control engineers
- d. Mechanics

Compile a list of all the interviewees along with date and time you plan to perform the interviews. Make sure you take detailed notes during the interviews.

12.1.3 Step 3: Analyze the Body of Data as a Whole

1. Look through all the process data, vibration data, temperature data, performance data, etc., and look for changes, trends, and correlations that may provide clues to what is going on. Carefully analyze the body of data as a whole.
2. If it is still not clear what the problem is, generate a list of plausible theories that can explain all the symptoms. (At this point, the more possible causes you can postulate the better your chances are of finding the root cause.) The following techniques can be helpful in paring down the possible causes to determine the most likely cause:
 - a. Field testing or trials to support or refute some of the promising theories.
 - b. Analytical modeling to validate postulated theories, such as performing an acoustic analysis of the piping system, a dynamic analysis of a piping system or structure, or a rod load analysis using actual field data.
 - c. "Cause mapping" to evaluate all potential causes

12.1.4 Step 4: Act and Confirm

1. Once you have decided to act, here are your choices:
 - a. Make no immediate operating changes or maintenance adjustments but continue to monitor the machine's condition.
 - b. Test and assess: 1) Vary operating condition speed, flow, pressures, suction level, etc. and observe how machine responds. 2) Check for off-design machine operation due to off-design process conditions, open bypasses, restrictions, control issues, etc. Correct any simple problems such as open bypass lines, low suction levels, and off-design conditions as you find them. If

the problem can be resolved with simple field adjustments, then you are done.

- c. If you determine that you are dealing with a system issue that cannot be corrected on-line, you will need to shut down. Here are a few examples of issues that can only be corrected:
 - i. Clearing line blockages
 - ii. Repairing leaking valves
 - iii. Replacing or adjusting troublesome control valves
 - iv. Cleaning fouled coolers
 - v. Adding orifices to dampen an acoustic resonance
 - vi. Supporting a resonant piping span
 - d. If you determine that you are dealing with a machine issue, here are your options:
 - i. Switch to spare, if available
 - ii. Plan a shutdown for balancing, alignment, oil change out, piping modifications, regrouting, etc.
 - iii. Plan a shutdown for repair (i.e., full machine overhaul if needed). Reduce stresses if possible while waiting for repair by reducing process induced loads
 - iv. Shut down immediately for balancing, alignment, oil change out, piping modifications, regrouting, etc.
 - v. Shut down immediately for repair
2. You should always follow up with a field audit in order to confirm that the root cause of the problem has been addressed and that machine conditions are back to normal. If the problem has been solved, you are finished; if not, go back to the “Collect all pertinent data” step and continue the investigation.
 3. If the problem is solved, take some time to document the results. Place a copy of the final report in

the equipment file and send copies to coworkers, team members, and management.

12.1.5 Troubleshooting Matrix and Table



The purpose of the reciprocating compressor troubleshooting matrix and table is to assist field troubleshooters in the identification and resolution of the following categories of process machinery problems. Keep in mind that the following troubleshooting matrix and table contains only the more common symptoms and causes found in the field. Theses matrixes should be considered a starting point for your analysis. You may need to contact the manufacture for assistance in solving rarely encountered or subtle problems.

12.1.6 Reciprocating Compressor Troubleshooting Example

Here's an example illustrating how the reciprocating compressor matrix can be used. Let's assume you are experiencing a high discharge temperature issue. You first select the column titled "High Discharge Temperature" and then write down all the "Common Root Causes" for further review (see Table 12.2a). The matrix indicates that 1) Intercooler ineffective, 2) Suction pressure too low, 3) Plugged suction strainer, 4) Leaking cylinder valve, 5) Suction temperature higher than design, and so on. There are

Table 12.2c Reciprocating compressor troubleshooting tips.

Instructions: This table is intended to complement Table 12.3b by providing additional troubleshooting advice. Additional field inspections or tests may be required to identify the hidden cause of a given machinery problem. For example: If you suspect that “wrong compressor speed” is a likely rootcause, go to the “Field Troubleshooting Tips” column and read the advice provided. For this potential root cause, it is recommended that you check the compressor speed indication if installed or by using a tachometer and compare the indicated speed with the design speed.		Reciprocating compressor field troubleshooting tips 
Common root causes	a. Incorrect cylinder clearance volume	1. If volume unloaders are installed, first check to make sure they are all in the correct positions. Leaking volume unloaders can significantly affect compressor flow. 2. Check the repair records to determine if the clearance volumes were measured and recorded. 3. Conduct a compressor performance (PV, i.e., Pressure versus Volume) analysis and see if cylinder performance appears to be normal.
	b. Wrong compressor speed	Check the compressor speed indication if installed or by using a tachometer and compare with design speed.
	c. Intercooler ineffective (multistage units)	Check the inlet and outlet temperatures on your coolers and compare them to the design values. A high inlet temperature is an indication of a compressor issues. A normal inlet temperature with a high outlet temperature is an indication of a cooler issue.
	d. Pulsation dampner dampener or plugged	If the problem has suddenly appeared, you may have a problem internal to the pulsation dampener. When possible, use a boroscope to inspect the pulsation dampener internals. Look for pluggage, or broken internal components.
	e. Pulsation dampener ineffective	If the problem is present immediately after commissioning, it may be a pulsation dampner design problem. Have someone verify the design.

(Continued)

Table 12.2c Cont.

	f.Suction pressure too low	1. Check the upstream system pressure to see if it is lower than normal due to a change in the process or an upset condition.. 2. Check for flow obstruction in suction piping.
	g.Plugged suction strainer, valve, or piping restriction	1. Check for flow obstruction in the piping. 2. Check any strainers in the suction and discharge line to see if they are plugged. 3. Ask the operator if the compressor piping has a history of pluggage.
	h.Open or leaking spillback valve	1. Check all potential bypass valves to ensure they are closed or are not leaking. 2. Check relief valves for any indications of leakage.
	i.Mechanical or acoustic resonance	Check vibration and pulsations as a function of speed. Resonances will appear as defined spikes in the data. Refer to Chapter 18 in "Is My Machine OK" for ways to identify resonances.
	j.Inadequate compressor support	Start your investigation by checking compressor vibration levels close to the skid or subbase. If vibration levels are high near hold down bolts, then you skid or subbase is not providing adequate support.
	k.Leaking cylinder valve	Use an infrared temperature gun or contact thermometer to check for hot valve caps. An isolated hot valve cap is an indication of a leaking cylinder valve. Then, replace all valves with hot valve caps and see if the symptoms disappear.
	l.Unbalanced reciprocating weights	Imbalanced reciprocating weights should show up as high 1x vibrational components in the spectra.
	m.Broken hold down bolt or support member	If the vertical vibration levels are more than two times higher than the horizontal vibrating level, you should suspect that you have a broken or loose hold down bolt or support member.

(Continued)

Table 12.2c Cont.

	n.Suction temperature higher than design	Check to see if the compressor suction temperature is deviating from the design value by more than 5%. A deviation greater than 5% will affect horsepower and performance.
	o. Gas ratio of specific heats (K) is lower than the design value.	Pull a representative process gas sample in order to determine its ratio of specific heats (K). Watch out for significant deviations from the normal molecular weight of the gas during start-ups.
	p.Gas ratio of specific heats (K) is higher than the design value.	Pull a representative process gas sample in order to determine its ratio of specific heats (K). Watch out for significant deviations from the normal molecular weight of the gas during start-ups.
	q.Leaking piston rings	Conduct a compressor performance (PV) analysis to determine if excessive leakage is present.
	r.Excessive cylinder valve losses	Perform a compressor performance analysis (PV) to evaluate the magnitude of the cylinder valve losses. Compare the valve losses with the manufacturer's estimate.
	s.Driver issue or malfunction.	1. Failing to reach rated speed is an indication of a driver issue. 2. Erratic speed is another indication of driver control issue.

13

Screw Compressors

Screw compressors are positive displacement type compressors used for medium flow and medium pressure applications when compared with other types of industrial compressors (see Figure 13.2). They contain two rotating screws; one of them is driven by a prime mover (motor or steam turbine) and the other is driven off of the driven screw. The compression process occurs in three distinct phases: suction inflow, compression, and discharge flow. Two counter-rotating helical screws (See Figure 13.1) are arranged in a compressor casing with gas inlet and discharge nozzles that are located at opposite ends of the casing. 1) First, the gas is inducted into the inlet nozzle as the counter-rotating screws move apart from one another. 2) Next, as the gas moves from inlet port into the compressor casing (see Figure 13.3), gas is gradually compressed until it arrives at the discharge end of the



Figure 13.1 Typical screw compressor rotors.

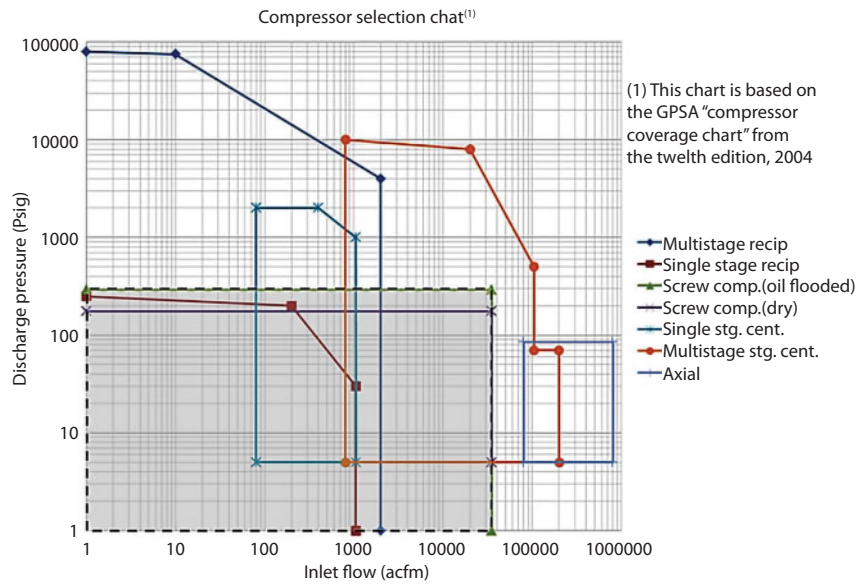


Figure 13.2 Flow and pressure range comparison of screw, reciprocating, and centrifugal compressors. The gray region in this chart represents the ideal operating range for a screw compressor.

casing. 3) Finally, at the discharge port the gas is expelled at the discharge pressure, as the counter-rotating screws move together, thereby decreasing the volume and raising the pressure of the trapped gas.

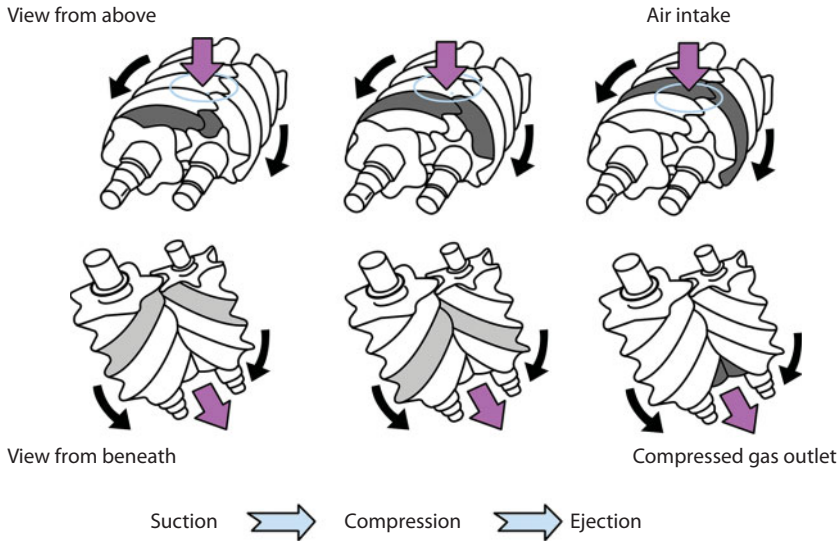


Figure 13.3 Screw compressor compression process.

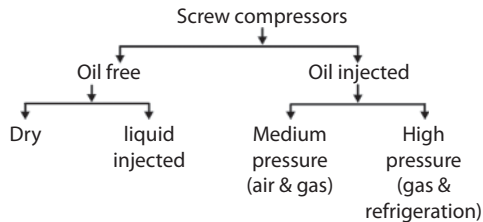


Figure 13.4 Types of screw compressors.

Screw compressors (see Figure 13.4) are typically used for either oil-free or oil-injected compression of air in mining, construction, industrial refrigeration, or a host of other applications where their relative simplicity, general reliability, and high availability is required. Rotary screw compressors are equally suited to compress process gases. Industrial screw compressors provide:

- High levels of reliability due to their simplicity
- Power savings due to an integral slide valve design
- Gas flows with lower pressure pulsations levels

They are also popular because they:

- Require less maintenance than reciprocating compressors, due to the fact they contain fewer internal parts. Five to six years of continuous operation may be possible between major overhauls.
- Tend to operate at lower pulsation and vibration levels than reciprocating compressors.

13.1 Oil Injected Screw Compressors

In an oil-injected rotary screw compressor, the oil serves five key purposes: it cleans, cools, lubricates, seals, and protects metal surfaces. They are comprised of two rotors inside the casing and they contact each other at lobe surface via an oil film. (A cutaway drawing of a typical oil-injected type screw compressor is shown in Figure 13.5.) The oil is separated from the discharge stream, then cooled, filtered and recycled. For optimum efficiency, internal clearances within the machine should be kept as small as possible. The presence of a large quantity of oil during the compression process lessens the chance of contact between the screw rotors.

Oil is supplied not only to the bearings and seal, but also to the rotor chamber directly. The oil acts as lubricant, coolant and sealant in the rotor chamber. Since oil is injected into the rotor chamber, the seal area between the rotor lobe and bearing is no longer necessary. There is only a single mechanical seal located at the drive shaft end. There are typically sleeve-type journal bearings on both ends of the rotor lobes. The thrust bearings are the typically tilting pad type, which are located outside of the journal bearings.

After leaving the compressor discharge flange, gas and oil mixture flows into an oil separation system located downstream of the compressor (see Figure 13.6). The bulk of the oil is separated in the primary oil separator, but a secondary coalescing

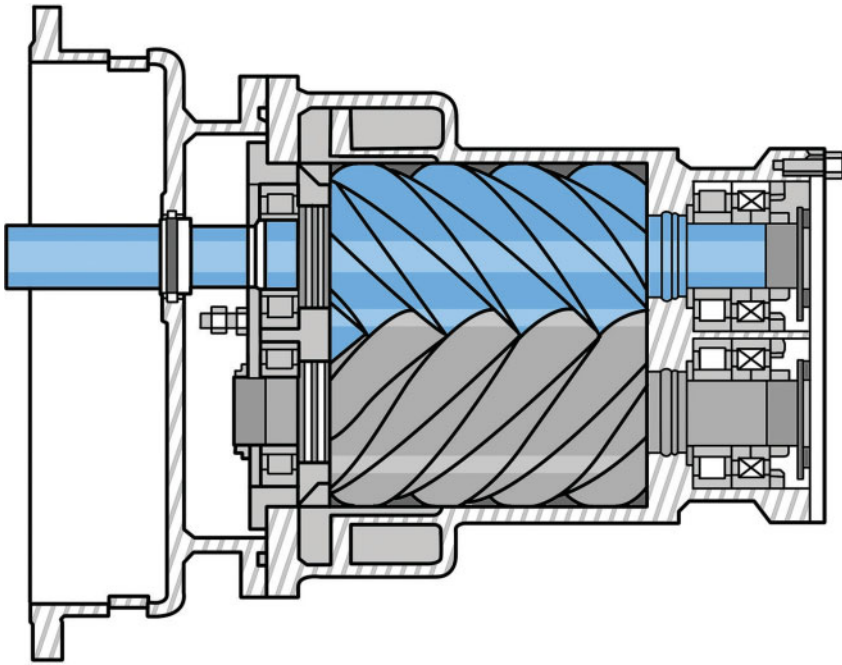


Figure 13.5 Oil-flooded screw compressor.

oil separator may be used to provide additional separation. The oil that is removed in the oil separation system is circulated back into the compressor lube system. Oil separation is one of the important factors for oil-injected screw compressors.

13.2 Screw Compressor Modulation

A slide loading valve, which is located inside the compressor casing beneath the twin rotors, is used to adjust the inlet flow volume. The slide valve can be actuated with either a hydraulic or air cylinder. As the slide moves toward the discharge end of the compressor, a larger and larger area is opened allowing more and more gas to spill back gas back to suction (see Figure 13.7). This control method provides flow control and allows for horsepower savings. The typical control range with

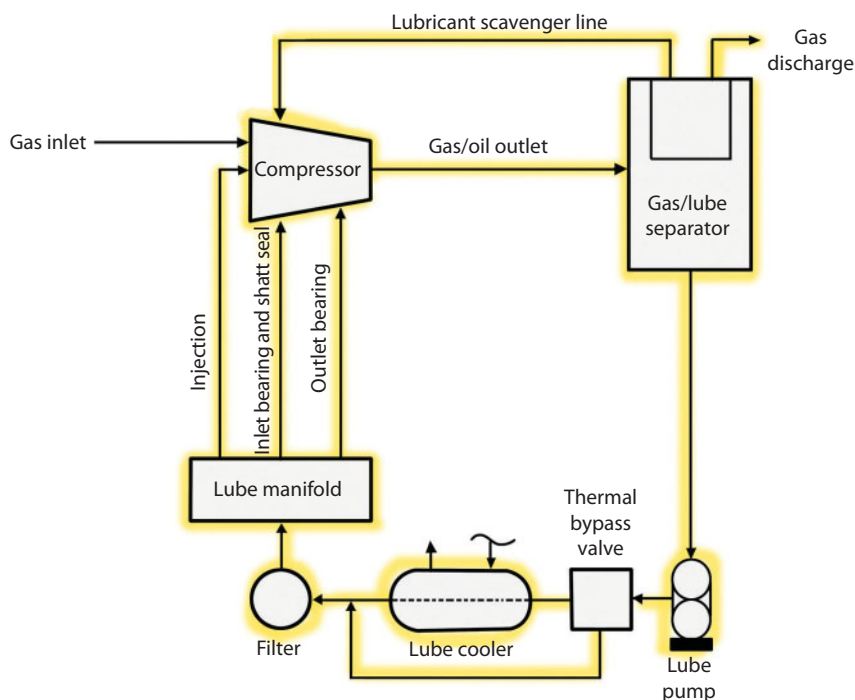


Figure 13.6 Oil-flooded screw compressor package.

slide valve is from 100% to 15%, in a continuous and stepless manner.

For constant and speed variable machines, a bypass control line may be needed if zero flow is required for the process. In multistage applications a gas cooler between the compressor stages is necessary to reduce the second-stage suction temperature. It may be necessary to install individual bypass control lines around each compression stage if precise interstage pressure control is required.

13.3 Pressure Pulsation Issues

Screw compressors are examples of positive displacement machines that generate pulsation energy at multiples of the lobe, or vane passing frequency. Vibration problems can occur when

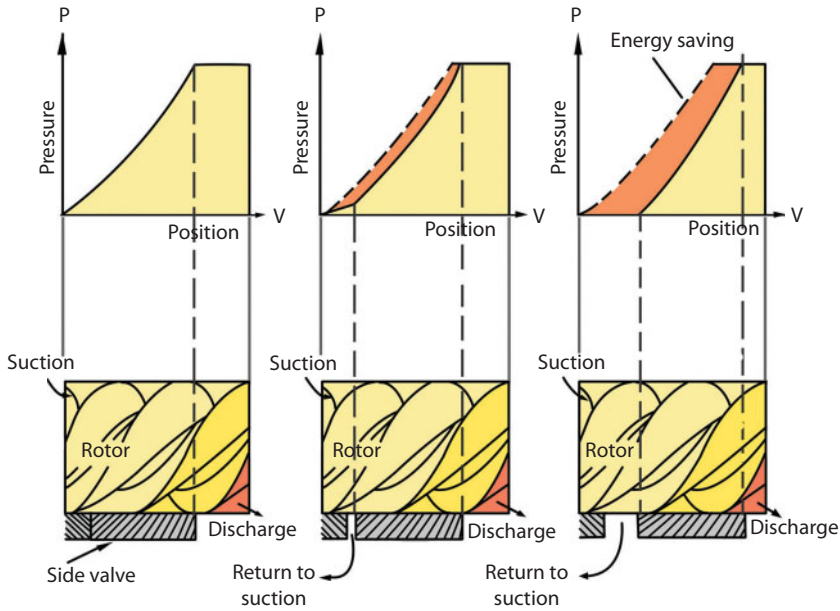


Figure 13.7 An internal slide valve is used to modulate screw compressor flow.

these pulsations create acoustical resonances within the piping or resonate with the mechanical natural frequencies (MNFs) of the vessel shell walls, which can in turn lead to severe noise and vibration issues, such as excessive instrumentation and small-bore piping vibration. To prevent these types of issues, silencers, or pulsation dampeners, are employed. To avoid vibration problems, a pulsation and mechanical analysis is used to evaluate the compressor and piping system. These studies are typically performed during the design stage and are also used to help identify and resolve problems that occur in the field.

The possible outcomes of a pulsation and mechanics study are 1) do nothing, 2) install orifices, or 3) install a silencer.

Orifice plates are often effective in attenuating pulsations generated by the compressor. They are effective in attenuating the pulsation levels at certain acoustical natural frequencies; however, the addition of an orifice plate will not attenuate the pulsations at all the multiples of the pocket passing frequency (PPF). For the orifice plates to be effective they must be installed

at a location which corresponds to the point of maximum particle velocity for that particular acoustical natural frequency.

Another way to address pulsations is to use a silencer, which is actually a pulsation attenuation device or dampener, in the downstream piping. There are three types of silencer designs – absorptive, reactive, and combination (reactive/absorptive):

13.3.1 Absorptive Type Dampeners

These types of silencers are also referred to as dissipative silencers. Their design depends on sound absorbing material to dissipate the sound energy. The sound waves pass through the spaces between the tightly packed, small diameter fibers of the absorptive material and the resulting viscous friction dissipates the sound energy as small amounts of heat. The absorptive silencers are like “glass-pack” mufflers used on cars. These designs provide good attenuation for high-frequency pulsation but less attenuation at the low and middle frequencies.

13.3.2 Reactive Type Dampeners

These silencers, also referred to as Helmholtz filters, are multi-chamber designs which are effective in reducing low-frequency pulsation energy; however, they are generally not effective in attenuating high-frequency energy at the higher multiples of the PPF. These types of filters are commonly used on reciprocating compressors to attenuate low-frequency pulsation.

13.3.3 Combination Type (Reactive and Absorptive)

This silencer type is a multichamber reactive silencer with absorptive material. There are many versions of this design with some configurations having additional absorptive material along the sides of the chambers (see Figure 13.8).

Keep in mind, silencers must be designed and built to withstand the high-frequency energy without experiencing fatigue failures of the internal parts and the vessel itself.

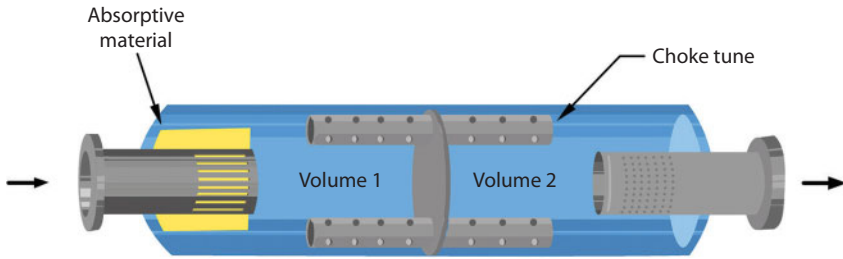


Figure 13.8 Combination silencer.

13.3.4 Oil Contamination

In a wet screw compressor, the oil is flooding the compression chamber, so any gas being compressed tends to mix with the oil. Therefore, it is critical to ensure that your oil can handle the gas and continue to maintain its lubricating properties. To ensure your compressor lubricant possesses the right properties for reliable operation, oil samples must be taken and then checked periodically. Among the oil analysis tests that should be performed are an elemental analysis, a viscosity analysis, and a wear debris analysis. (Note: The original equipment manufacturer may recommend other lubrication test. Based on their experiences.)

Users should frequently monitor oil viscosity to ensure that refrigerant dilution is not causing a decrease in viscosity, which will surely lead to an increase in machine wear. In some cases, oil samples must be degassed before they can be shipped to the laboratory or analyzed. With proper care and attention, the compressors in your refrigeration systems can provide years of trouble-free service. When changing the oil in these systems, keep in mind that it must be compatible with the refrigerant and the fluid previously used in the system.

Recovery of oil from the discharge gas is an important consideration in the design, operation, and maintenance of wet screw refrigeration compressors. The oil separator usually uses coalescing filter technology. If vapor-phase oil carryover is a significant concern, put in two oil separators and

install an aftercooler (to condense vaporized oil) between the primary and secondary oil-separator vessels. Usually, oil content (including vapor, aerosols, etc.) ranges from 1 to 5 ppm. Services requiring lower residual oil content (for example, 0.1–0.5 ppm or even less) may need a three-level oil-separator system.

13.3.5 How Process Conditions Affect Screw Compressor Performance

Changing the operating conditions from their original design values can significantly impact a compressor's actual capacity, discharge temperature, and horsepower demands. Some operating factors affecting performance changes are the compressor speed, suction and discharge pressure and temperature, gas composition, and suction and discharge piping design. The table below summarizes how different process variables can affect field performance. Keep these relationships in mind when evaluating your screw compressors in the field.

13.4 Troubleshooting Screw Compressors

Here are some common screw compressor issues and some advice on how to address them:

1. Reasons why a compressor won't start:
 - a. The compressor has tripped due to a safety device
 - b. There is some type of driver issue. Check the driver.
2. Reasons for low flow
 - a. A pluggage in the suction line
 - b. The suction pressure is significantly lower than the design pressure

Table 13.1 How process conditions affect screw compressor performance. Note: The results in this table assume that only one variable is changing at a time.

Process modification	Flow (at standard conditions)	Discharge temperature	Gas horsepower
Speed increase	Increases	Increases	Increases
Speed decrease	Decreases	Decreases	Decreases
Increase in suction pressure	Increases	Decreases	Increases
Increase in suction temperature	Decreases	Increases	Decreases
Decrease in suction temperature	Increase	Increases	Increase
Decrease in suction pressure	Decreases	Increases	Decreases

- c. The suction temperature is significantly higher than the design pressure
- d. The compressor input speed is lower than the normal operating speed. A low input speed will result in a proportionally lower flow.
3. Reasons for high discharge temperature. (Note: Rotary screw compressor failure due to high discharge temperature caused by an excessive differential pressure.)
 - a. A discharge line blockage or restriction
 - b. The suction pressure is lower than the design pressure
 - c. The discharge pressure is higher than the design pressure
 - d. The suction temperature is higher the design value

- e. The compression ratio (P_d/P_s) is higher than that recommended by the manufacturer
 - f. The input compressor speed is higher than normal. A high input speed will result in a proportionally higher flow and a resulting high discharge pressure.
4. Reasons for driver overload or trip
- a. The suction pressure is higher than the design pressure
 - b. The suction temperature is lower than the design value
 - c. The gas composition is heavier than expected
 - d. The compression ratio is higher than that recommended by the manufacturer
5. Reasons for high pressure pulsations or vibration
- a. Screw compressors generate pulsation energy at multiples of the pocket, lobe, or vane passing frequency. Vibration problems can occur when these pulsations cause acoustical resonance within the piping or resonate with the mechanical natural frequencies (MNFs) of the vessel shell walls or other piping. Check to see if a pulsation analysis was performed on your compressors and that all the recommendations were implemented.
 - b. Vibration problems can also occur due to driver critical speed problems. Perform vibration analysis during runup and shutdowns to identify critical speeds. Implement the required recommendations to eliminate critical speed in the normal operating speed range.

14

Compressor Start-Up Procedures

We all know that starting up a machine incorrectly can seriously damage a costly machine. In this chapter, we will cover generic start-up procedures for compressor trains for common operational situations. Operating procedures provide a consistent basis for safely operating your equipment.

Every compressor is unique due to its construction and design conditions; therefore, every compressor train should have its own customized procedure to ensure that all necessary steps are followed during start-ups. Only approved procedures that follow the manufacturer's recommendations, combined with operating experience on this compressor in this service should be used.

Remember that attention to detail is paramount when writing start-up procedures and that every compressor train should have provisions for common situations, such as:

1. Starting up a compressor after a major repair
2. Starting up a compressor after a minor repair, such as a seal or bearing change
3. Starting up a compressor with a start-up gas
4. A normal start-up after an unexpected shutdown.

For all these situations, operations must have a set of procedures that guarantee these shutdowns are uneventful and safe.

14.1 Compressor Start-Up Risks

By following predefined start-up procedures, in the same sequence, every time, operators can reduce the risks related to start-ups. Just like piloting an airplane, where takeoffs and landings represent the highest risk events, start-ups represent the times of greatest risk to a compressor. High-risk times are periods when there is a realistic probability that an undesirable event can occur, due to unforeseen circumstances. An undesirable event is any situation fraught with risk. For example, driving over a pothole is an undesirable event because it can lead to anything from an irritating jolt to a bent wheel or front-end alignment issue. We cannot say for sure what will happen in all cases. The extent of the damage depends on the size and depth of the pothole and the speed of the car driving over it. We instinctively know we should avoid potholes because something bad can potentially result from failing to do so. Similarly, we know there are undesirable events we should always try to avoid around compressors, such as liquids carryover, surging, oil contamination, a lack of lubrication, and overspeed events. To truly appreciate the importance of avoiding undesirable

events, I will describe some of the undesirable events we want to avoid around compressors:

1. Liquid slugs: Compressors are not designed to handle liquids. Liquid in a reciprocating compressor will lead to a catastrophic failure. Liquids in centrifugal and screw compressors can also result in damage and trips due to excessive loading.
2. Overspeed: Overspeeding a compressor train will usually result in a trip. If a trip does not occur, any of the rotors in the train could suffer significant damage. Any overspeed trip event has the possibility of leading to a full-blown rotor failure.
3. Lack of lubrication: Lubrication issues will lead to bearing failure and in some cases can result in major shaft damage.
4. Oil contamination: Water or solids in the lubricant will normally result in premature bearing failures. In the worst cases, shaft damage and even rotor damage can occur.
5. Surging: Surge can be a detrimental condition in compression systems because it causes the compressor to vibrate violently due to rapid internal flow reversals, potentially damaging internal compressor parts. Surge control systems are specifically designed to prevent compressors from operating below the surge flow point or line. But to work properly, the surge piping and system must be set up properly.
6. Operating past stonewall: Centrifugal compressor stonewall or choked is a flow condition that occurs when the sonic velocity of gas is reached at the exit of a compressor wheel. When stonewall flow is reached, the discharge pressure and flow drop precipitously. Flow through the compressor cannot

be increased any further without an increase in the suction pressure. At the stonewall (or choke) flow the pressure vs. volume curve becomes essentially vertical, and it is not possible to develop head or pressure at any greater flow. When the required operating flow exceeds the stonewall limit, the only remedy is to reconfigure the compressor with impellers (and matched stationary hardware) designed for larger flow rates. Note: Axial compressors have been known to fail catastrophically at high flows due to blade vibration and high temperatures generated when operated beyond the stonewall line.

To maximize compressor reliability, we must avoid the undesirable events mentioned here. The good news is that all these undesirable events are avoidable by design, vigilance, and by following detailed operating procedures.

14.2 Generic Start-Up Procedure

Every compressor is unique in its design and construction and therefore should have its own customized procedure to ensure that all necessary steps are followed during start-ups. Only approved procedures that follow the manufacturer's recommendations combined with operating experience on this compressor in this service should be used—attention to detail is paramount.

General sequence:

1. Purge compressor and piping system of air
2. Start up ancillary equipment
3. Start and run unloaded
4. Run at the minimum speed
5. Warm up for a set period of time
6. Gradually load the compressors

7. Monitor the performance and condition of the compressor for a minimum of 4 hr

Start-up procedures vary considerably from machine to machine. However, the following generic start-up procedure can be applied to most compressors:

1. Assure that the machine is completely assembled and ready for start-up, and that the process is ready for the compressor to come on-line. Inform all affected personnel that the compressor is going to be started up.
2. Place ancillary lubricating and seal oil systems, housing vent lines, and other components into operation.
3. Use an inert gas, such as carbon dioxide or nitrogen, to purge the compressor case or cylinders of air, if required to prevent air from entering the process.
4. Once the machine is pressurized, completely drain the compressor's case or cylinders to remove any liquid, and then check for leaks.
5. Adjust the controls to ensure a minimum driver load at start-up (e.g., place centrifugal compressors on bypass or unload reciprocating compressor cylinders).
6. Prepare the driver for start-up (e.g., prepare steam systems and superheaters and/or remove tagout devices from electrical switches).
7. Start the machine under a low or no-load condition and bring the train up to a minimum speed as allowed by the driver.
8. Closely monitor operation as the machine is idling. After about 1 to 2 hr and, when you are satisfied that there are no issues, gradually accelerate the train up to normal operating speed. During

Table 14.1 Recommended Compressor Inspection.

Recommended compressor field inspections		
Look	Listen	Feel
Check oil levels to ensure they are satisfactory	Listen for abnormal noises	Feel for excessive vibration. Call for assistance if you spot a problems area.
Check seals and flanges to ensure there are no leaks		Feel for excessive heat. Use an infrared gun to investigate, if you find any questionable locations.
Check vibration monitors to ensure all readings are satisfactory		
Verify that suction drum levels are within acceptable limits		

the acceleration process, avoid running the equipment in its critical speed ranges for any extended periods of time.

9. Once at full speed conditions, closely monitor operation as the machine runs in and comes to normal operating conditions. (Table 14.1 is a listing of recommended field inspections that apply to all compressor types.)
10. Make final adjustments to properly load the compressor train for the process requirements. For reciprocating compressors, adjust the cylinder loadings for efficient operation once you

are satisfied that the machine is running smoothly.

14.3 Centrifugal Compressor Start-Ups

Every compressor is unique in its design and construction and therefore they should all have their own customized procedure to ensure that all necessary steps are followed during start-ups. Only approved procedures that follow the manufacturer's recommendations combined with operating experience on this compressor in this service should be used—attention to detail is paramount.

If the compressor is motor driven, care must be taken to avoid multiple start-up attempts. Generally, if the compressor does not start by the second attempt there is no point in making a third attempt because large motors will generally lock out and there will be a significant cool down time before it can be attempted again. A check should be made of all the permissives after the first attempt to see why the compressor did not start. (A permissive is a particular type of safety interlock used to prevent actions from taking place until predefined criteria have been satisfied. For example, some permissives prevent machine start-ups whenever there is a condition of low oil pressure, high suction drum level, high winding temperatures, etc. If these permissives are not cleared before more attempts to start, the motor will likely be locked out for hours before you can try to start the compressor again.)

After start-up, continue to monitor compressor conditions for at least an hour to ensure everything is normal. If unresolved issues are encountered during this start-up phase, either call for help to get a second opinion or shut down until additional technical assistance arrives to help you understand what is going on. If a problem is detected, be sure to capture the process data present when the problem occurred.

Here's a starting procedure based on our generic compressor start-up procedure:

1. Assure that the machine is completely assembled and ready for start-up, and that the process is ready for the compressor to come on-line.
2. Inform all affected personnel that the compressor is going to be started up.
3. If not already done, use an inert gas, such as carbon dioxide or nitrogen, to purge the compressor case of air, if required to prevent air from entering the process.
4. Place ancillary lubricating and seal oil systems, housing vent lines, and other components into operation.
5. Once the machine is pressurized, completely drain compressor casing and all suction drums and suction piping to remove any liquid, and then check for leaks.
6. Confirm that there are no leaks from the mechanical or dry gas seal area, now that you have suction pressure on the mechanical or dry gas seals. For dry gas seal systems, check the primary leakage rate or pressure to confirm that the system is working properly.
7. Confirm that the anti-surge valve is fully opened. This valve must be opened to give the compressor discharge gas a path back to the suction of the compressor and prevent surging. Once this compressor is up to speed then this valve will slowly close to force gas through the discharge valve. Most centrifugal compressors have a special control program or system which controls the anti-surge valve; therefore it is not something the operator has to manually open or close. However, the operator must understand what

position the valve needs to be in for a compressor start-up.

8. Adjust the controls to ensure there is a minimum driver load before starting. For a centrifugal compressor this means you are in a 100% bypass condition.
9. Ensure the driver is ready for start-up (e.g., prepare steam systems and superheaters and/or remove tagout devices from electrical switches).
10. Start the machine under a low or no-load condition and bring the train up to a minimum speed as allowed by the driver.
11. Closely monitor operation as the machine is idling. After about 1 to 2 hr and, when you are satisfied that there are no issues, gradually accelerate the train up to normal operating speed. During the acceleration process, avoid running the equipment in its critical or resonance speed ranges for any extended periods of time.
12. Once at full speed conditions, closely monitor operation as the compressor runs in and comes to normal operating conditions.
13. Make final adjustments to properly load the compressor train for the process requirements. If any problems are encountered during this start-up phase, either call in help for a second opinion or shut down until additional technical assistance is available.

14.4 Reciprocating Compressor Start-Ups

Every compressor is unique due to its construction and design conditions; therefore, every compressor train should have its own customized procedure to ensure that all necessary steps are followed during start-ups. Only approved procedures that

follow the manufacturer's recommendations, combined with operating experience on this compressor in this service should be used. Operating procedures provide a consistent basis for safely operating your equipment.

Remember that attention to detail is paramount when writing start-up procedures and that every compressor train should have provisions for common situation, such as:

1. Starting up a compressor after a major repair
2. Starting up a compressor after a minor repair, such as a seal or bearing change
3. Starting up a compressor with a start-up gas
4. A normal start-up after an unexpected shutdown.

For all these situations, operations must have a set of procedures that guarantees these shutdowns are uneventful and safe.

Most reciprocating compressors are mechanically designed to be started up and shut down under no load conditions. This means that the compressor must not be doing any work (or very little) at the start-up and shutdown operation. It is also very important to ensure that all lubrication points are receiving the proper lubrication prior and during start-up. The oil pump pressure and/or flow must be ensured before attempting a start-up.

There are many ways to start up a reciprocating compressor. Your specific procedure will depend on your process and the operating conditions. Your start-up procedure depends on:

1. How you purge, i.e., air free, the compressor
2. Whether you have 100% unloading capabilities
3. Whether have 100% spillback capabilities
4. If your valving and cylinder unloaders are manual or automatic

Before writing a procedure for your specific application, conduct a detailed hazard analysis so that you better understand the potential hazards involved in your normal, day-to-day operations.

Here's a starting procedure based on our generic compressor start-up procedure:

1. Assure that the machine is completely assembled and ready for start-up, and that the process is ready for the compressor to come on-line.
2. Most large reciprocating compressors have a jacking or barring gear to allow the compressor to be turned over to ensure that there is no liquid in any of the cylinders prior to starting. The compressor should be barred over at least twice before attempting to start a compressor.
3. Inform all affected personnel that the compressor is going to be started up.
4. If not already done, use an inert gas, such as carbon dioxide or nitrogen, to purge the compressor case or cylinders of air, if required to prevent air from entering the process.
5. Place ancillary lubricating and seal oil systems, housing vent lines, and other components into operation.
6. Once the machine is pressurized, completely drain all suction drums and suction piping to remove any liquid, and then check for leaks.
7. Adjust the controls to ensure there is a minimum driver load before starting. For a reciprocating compressor this means to either unload all cylinders or to go on 100% bypass. In some cases, you may choose to do both.
8. Ensure the driver is ready for start-up (e.g., prepare steam systems and superheaters and/or remove tagout devices from electrical switches).

9. Start the machine under a low or no-load condition and bring the train up to a minimum speed as allowed by the driver.
10. Closely monitor operation as the machine is idling. After about 1 to 2 hr and, when you are satisfied that there are no issues, gradually accelerate the train up to normal operating speed. During the acceleration process, avoid running the equipment in its critical or resonance speed ranges for any extended periods of time.
11. Once at full speed conditions, closely monitor operation as the compressor runs in and comes to normal operating conditions.
12. Make final adjustments to properly load the compressor train for the process requirements. For screw compressors, adjust the slide valve for efficient operation once you are satisfied that the machine is running smoothly. If any problems are encountered during this start-up phase, either call in help for a second opinion or shut down until additional technical assistance is available.

Keep in mind that every compressor is unique in its design and construction, and therefore should have its own customized procedure to ensure all necessary steps are followed during start-ups. Only approved procedures that follow the manufacturer's recommendations and experience with this compressor in this application should be used—attention to detail is paramount.

14.5 Screw Compressor Start-Ups

Every compressor is unique due to its construction and design conditions; therefore, every compressor train should have its own customized procedure to ensure that all necessary steps

are followed during start-ups. Only approved procedures that follow the manufacturer's recommendations, combined with operating experience on this compressor in this service should be used. Operating procedures provide a consistent basis for safely operating your equipment.

Remember that attention to detail is paramount when writing start-up procedures and that every compressor train should have provisions for common situation, such as:

5. Starting up a compressor after a major repair
6. Starting up a compressor after a minor repair, such as a seal or bearing change
7. Starting up a compressor with a start-up gas
8. A normal start-up after an unexpected shutdown.

For all these situations, operations must have a set of procedures that guarantees these shutdowns are uneventful and safe.

Remember that start-up procedures vary considerably from machine to machine. Here is a generic start-up procedure for screw compressors:

1. Assure that the machine is completely assembled and ready for start-up, and that the process is ready for the compressor to come on-line. Inform all affected personnel that the compressor is going to be started up.
2. If not already done, use an inert gas, such as carbon dioxide or nitrogen, to purge the compressor case or cylinders of air, if required to prevent air from entering the process.
3. Place ancillary lubricating and seal oil systems, housing vent lines, and other components into operation.

4. Once the machine is pressurized, completely drain the compressor's case and piping to remove any liquid, and then check for leaks.
5. Adjust the controls to ensure a minimum driver load at start-up. For screw compressors this means going on full bypass with the slide valve in the minimum load position.
6. Prepare the driver for start-up (e.g., prepare steam systems and superheaters and/or remove tagout devices from electrical switches).
7. Start the machine under a low or no-load condition and bring the train up to a minimum speed as allowed by the driver.
8. Closely monitor operation as the machine is idling. After about 1 to 2 hr and, when you are satisfied that there are no issues, gradually accelerate the train up to normal operating speed. During the acceleration process, avoid running the equipment in its critical or resonance speed ranges for any extended periods of time.
9. Once at full speed conditions, closely monitor operation as the machine runs in and comes to normal operating conditions.
10. Make final adjustments to properly load the compressor train for the process requirements. For screw compressors, adjust the slide valve for efficient operation once you are satisfied that the machine is running smoothly.

15

Compressor Trains: Drivers, Speed Modifiers, and Driven Machines

Process machinery is typically composed of a group of subelements that convert one type of energy into another until it is finally transferred into a useable form of fluid power within a process. Here is a simple flow chart showing how power flows through a machine train.

Energy (in) → Driver → Speed Modifier → Driven Machine
→ Process Fluid Power (out)

Machine train subelements are normally interconnected using flexible components called couplings. Figure 15.1 illustrates a simple machine train comprised of an electric motor (on the left) directly coupled to a reciprocating compressor (on the right).



Figure 15.1 An electric motor directly coupled to a reciprocating compressor.

Energy, such as electrical power, steam power, or fuel gas, is first converted into rotational output power. The speed of the driver output shaft may be increased or decreased by a speed modifier, i.e., gearbox or pulleys, depending on the requirement of process machine being driven. Finally, the output speed from the speed modifier powers the driven machine that produces fluid power in the process. Table 15.1 contains common designs for driven machines, drivers, speed modifiers, and combination machines.

15.1 Driven Process Machines

The purpose of a driven process machine is to deliver a given process fluid, at a given flow and pressure, to specific points in a process. Driven machines receive the power input from a driver or speed modifier and convert it into fluid power at the

Table 15.1 Common Types of Process Machinery Elements.

Driven machines	Drivers	Speed modifiers	Combination machines
Centrifugal compressors	Electric motors	Gearboxes	Turbo expanders
Reciprocating compressors	Steam turbines	Hydraulic speed modifiers	Turbochargers
Rotary compressors	Gas turbines	Sheaves and belts	
Centrifugal pumps	Reciprocating engines		
Reciprocating pumps			
Rotary pumps			
Fans			

process machine’s discharge flange. All driven process machines are composed of an input shaft, a casing to contain the process fluid, a suction nozzle for input flow, a discharge nozzle for output flow, bearings to support the rotor (or rotors), and one or two end seals to prevent process leakage into the atmosphere.

There are many different designs employed to convert rotary power into fluid power.

Process machines that move and compress gases are called compressors or fans and process machines that move liquids are called pumps. In this book, we are going to concentrate on centrifugal, reciprocating, and screw compressors.

15.1.1 Drivers

Before we begin discussing compressors, we will briefly cover drivers. A driver is any mechanical device that provides power to a process machine, such as a pump or

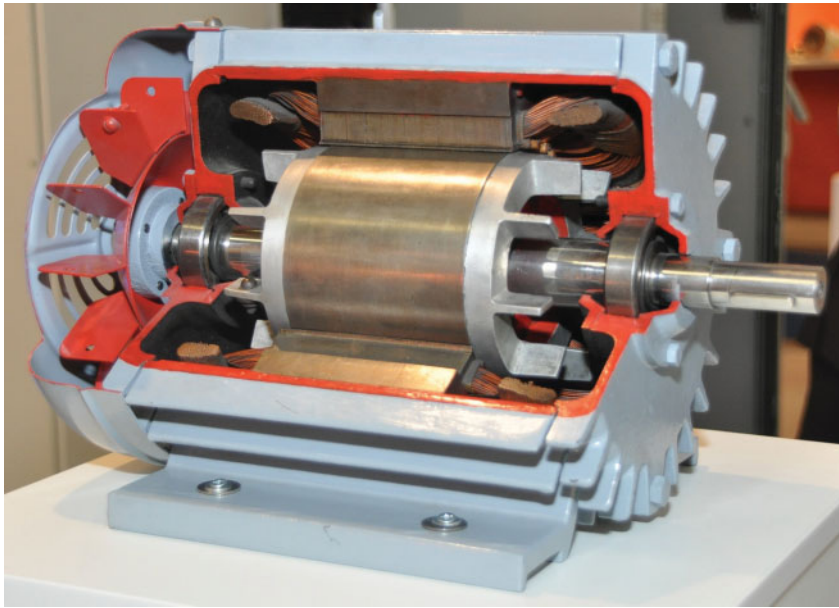


Figure 15.2 AC induction motor.

compressors, at a constant speed. Some drivers may require some type of speed control to maintain a constant speed under changing process loads. Here are some of the most common drivers:

15.1.1.1 AC Electric Motors

An electrical motor is an electromechanical device which converts electrical energy into a mechanical energy (see Figure 15.2). The two main types of AC motors are induction motors and synchronous motors. Induction motors work due to a small difference in speed between the stator rotating magnetic field and the rotor shaft speed called slip to induce rotor current in the rotor AC winding. For this reason, induction motors cannot produce torque near synchronous speed. In contrast, synchronous motors produce its rated torque at exactly synchronous speed.

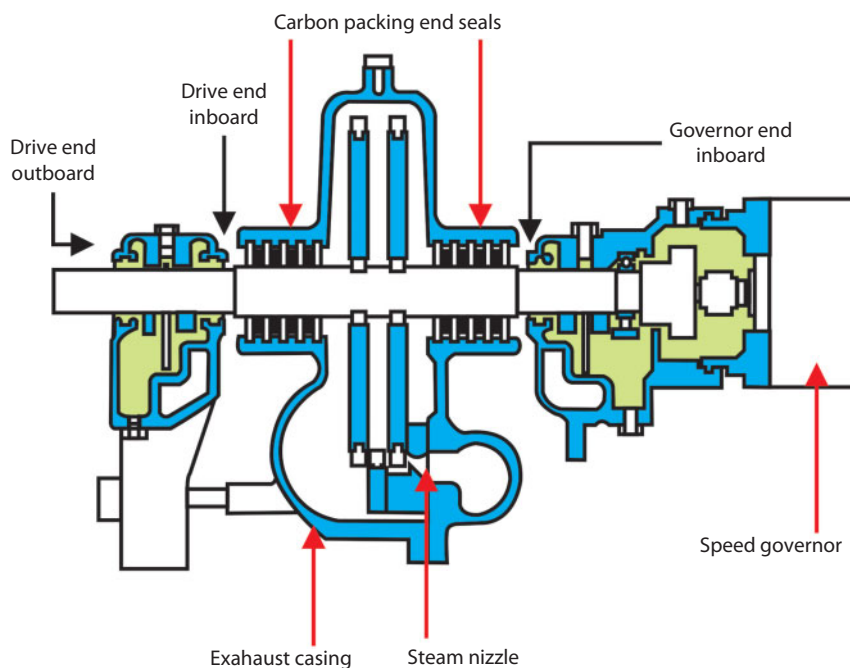


Figure 15.3 Cross-section of single-stage turbine and governor system.

The most common induction motors in industry run off three-phase electrical power. Three-phase induction motor as this type of motor does not require any starting device, or we can say they are self-starting induction motors.

15.1.2 Steam Turbines

A steam turbine is a mechanical device that extracts thermal energy from pressurized steam and transforms it into mechanical work (see Figure 15.3). Because the turbine generates rotary motion, it is well suited to driving process machinery, such as pumps and compressors. Steam turbines require some type of control system to maintain train speed under changing load conditions. In some applications a speed modifier, such as a gearbox, may be required to match the steam turbine to the speed of the driven process machine.

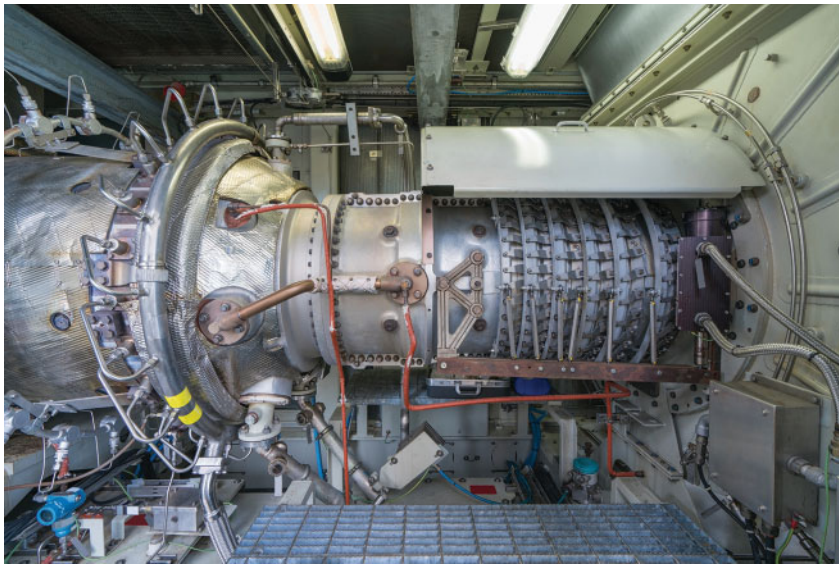


Figure 15.4 Industrial gas turbine: The air compressor is on the right and the combustion section and the power recovery turbine are on the left.

Steam turbine drivers may be chosen over other types of drivers if:

- There is excess, i.e., waste steam is available from the process that can be used to produce power
- Variable speed capabilities are required
- There is no electrical power available
- There is a need for immunity from power outages

15.2 Gas Turbines

A gas turbine is a type of internal combustion engine that has an upstream rotating compressor coupled to a downstream power recovery turbine with a combustion chamber in between (see Figure 15.4). Energy is added to the gas stream in the combustor, where fuel is mixed with air and ignited. In the high-pressure environment of the combustor, combustion of the fuel increases the temperature of the gas mixture. The combustion

products are then forced into the turbine section where the high velocity and mass flow is directed through a nozzle in front of the turbine's blades, creating torque and causing the turbine rotor to spin.

About two-thirds ($2/3$) of the horsepower recovered in the power turbine section is used to power the compressor. The excess power can be used to generate power or to drive a pump or compressor. The energy given up to the turbine comes from the reduction in the temperature and pressure of the exhaust gas.

Gas turbines require sophisticated controls to maintain speed under changing loads, while preventing internal temperature excursions that could result in permanent damage. In some applications a speed modifier, such as a gearbox, may be required to match the gas turbine to the speed of the driven process machine.

Gas turbine drivers may be chosen over other types of drivers if:

- There is an abundance of natural gas available at the site
- Variable speed capabilities are required
- There is no electrical power available at the site

15.2.1 Natural Gas Engines

Reciprocating natural gas engines are widely used to drive natural gas compressors used on gas transmission pipelines and in gas processing facilities (see Figure 15.5). These engines work on the principle of internal combustion where the ignition and combustion of the fuel occurs within the engine itself. The engine converts the combustion energy into mechanical work. The expanding combustion gases push the pistons, which in turn rotates the crankshaft. In many applications the crankshaft is directly connected to a reciprocating compressor through a coupling

Natural gas-fired reciprocating engine combustion designs are separated into three design classes: 2-cycle (stroke)



Figure 15.5 Reciprocating compressor driven by a natural gas engine.

lean-burn, 4-stroke lean-burn, and 4-stroke rich-burn. Combustion, also known as burning, is the basic chemical process of releasing energy from a fuel and air mixture. Unwanted emissions are produced during the combustion process that can be mitigated through engine design, controls, and exhaust catalytic converters.

Natural gas engines drivers may be chosen over other types of drivers if:

- There is an abundance of natural gas available at the site
- Variable speed capabilities are required
- There is no electrical power available at the site

15.2.2 Speed Modifiers

15.2.2.1 Gear Boxes

Gearboxes (see Figure 15.6) are used whenever 1) there is a difference between the speed of the driver and the driven piece

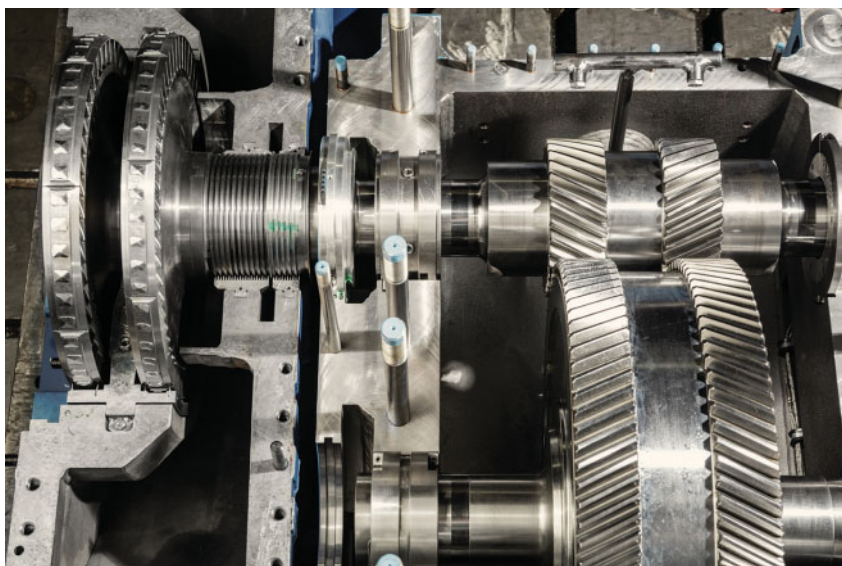


Figure 15.6 Gearbox.

of equipment, 2) the input and output shafts need to be in different directions, or 3) the input and output shafts operate in different axes of rotation. A speed increaser is a type of gearbox where the input shaft is turning slower than the output shaft. A speed reducer is a type of gearbox where the input shaft is turning faster than the output shaft.

From an operator's or casual observer's point of view, the most notable items when passing by a gearbox are the sound the gears are making and the feel of both temperature and vibration when touched. Current values need to be compared to the usual levels. Have they changed from what is accepted as normal, and if so, why? Always check foundation bolts by looking for oil being squeezed in and out between the base plate and the foot of the gearbox. Look at the shims as well. Do they look like they have been squirming out? This is a potential sign of looseness or high vibration at some time. Occasionally, check the foundation bolt or nut to ensure it is tight. This bolt could be broken off in the foundation (not easily detected), relaxing its grip on the rotating equipment. If, when trying to turn the

bolt, it moves, then it is likely the stud in the foundation that is broken.

15.3 Useful Gearbox Facts

The input and output shaft direction of rotation will depend on how many gear shafts are inside the gearbox. An even number of gear shafts equals opposite direction of rotation for input and output shafts; an odd number of gear shafts mean the same direction of rotation for input and output shafts.

15.4 Combination Machines

15.4.1 Turboexpanders

The term “Turboexpander” is used to define a combination expander/compressor machine constructed as a single unit (see Figure 15.7). It consists of two primary components; the radial inflow expansion turbine and a centrifugal (booster) compressor combined as an assembly. Both wheels are placed on a single shaft. The expansion turbine is considered the power unit and the compressor is considered the driven unit.

In a gas processing plant, the purpose of the turboexpander is to efficiently perform two distinctly different, but complementary, functions in a single machine. The primary function is to efficiently generate refrigeration in the process gas stream, which is done by the expansion turbine end extracting the potential heat energy from the gas stream, causing it to cool dramatically. The extracted energy is converted to mechanical energy that powers the booster compressor end of the turboexpander, which partially recompresses the residue gas stream. When designed properly, a turboexpander can yield very high mechanical and thermodynamic efficiencies at the “design point.”

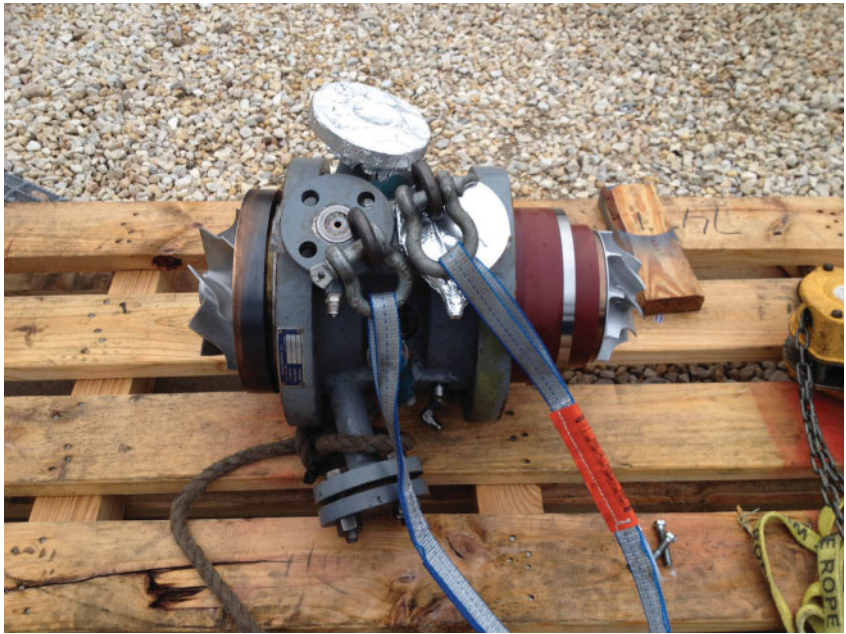


Figure 15.7 Turboexpander unit removed from service. The compressor end is on the left side and the expander is on the right side.

16

Compressor Components

The overall reliability of process machines depends in large part on the reliability of their components, such as bearings and seals. Each bearing and seal design come with its own set of challenges. If the proper bearings and seals are selected and maintained, you can expect long service lives from your machines. The following is a brief explanation of the various types of bearings and seals commonly found in process machinery.

16.1 Bearing Types

Figure 16.1 contains a list of bearing types, categorizing them in terms of their construction. Plain bearings fall into two categories: metallic and non-metallic, with each of these types having many different design options. Similarly, rolling element bearings fall into two main categories: unmounted and

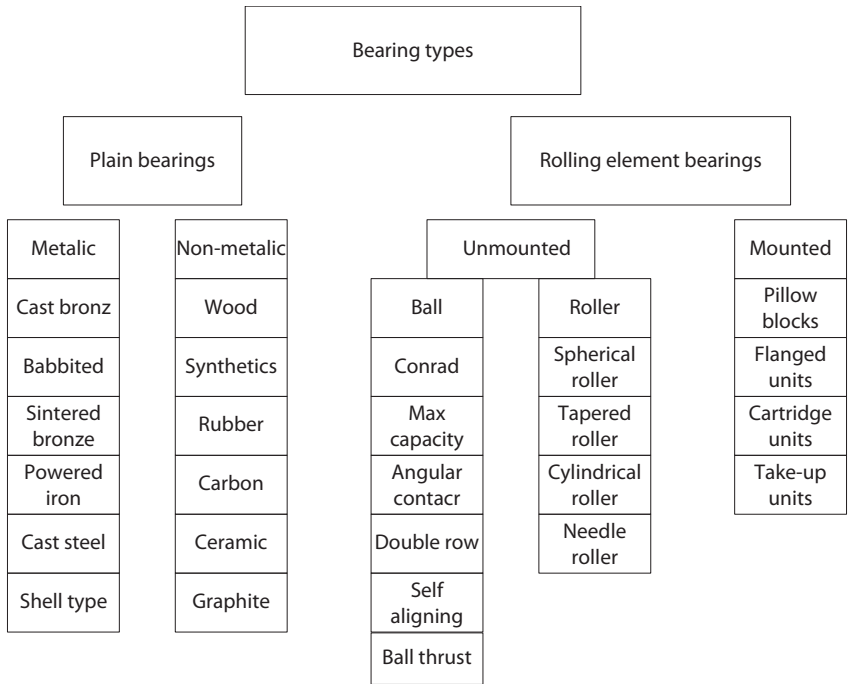


Figure 16.1 Types of bearings.

mounted. Unmounted rolling element bearings can further be broken down as either ball or roller types. Mounted rolling element bearings can be broken down into design options, such as pillow block bearings, flanged units, cartridge units, and take-up units.

The main takeaway from Figure 16.1 is that there are many types of bearing configurations available to machinery designers that utilize different materials and geometries. As an exercise, the reader can select a piece of equipment in a plant and determine the types of bearings it contains by talking to the people that maintain the equipment or by using the maintenance manual. It is beneficial for operators to understand what style of bearings a given machine uses in order to know how they must be maintained, and to be able to relate it to the information in the lubrication section in Chapter 17.

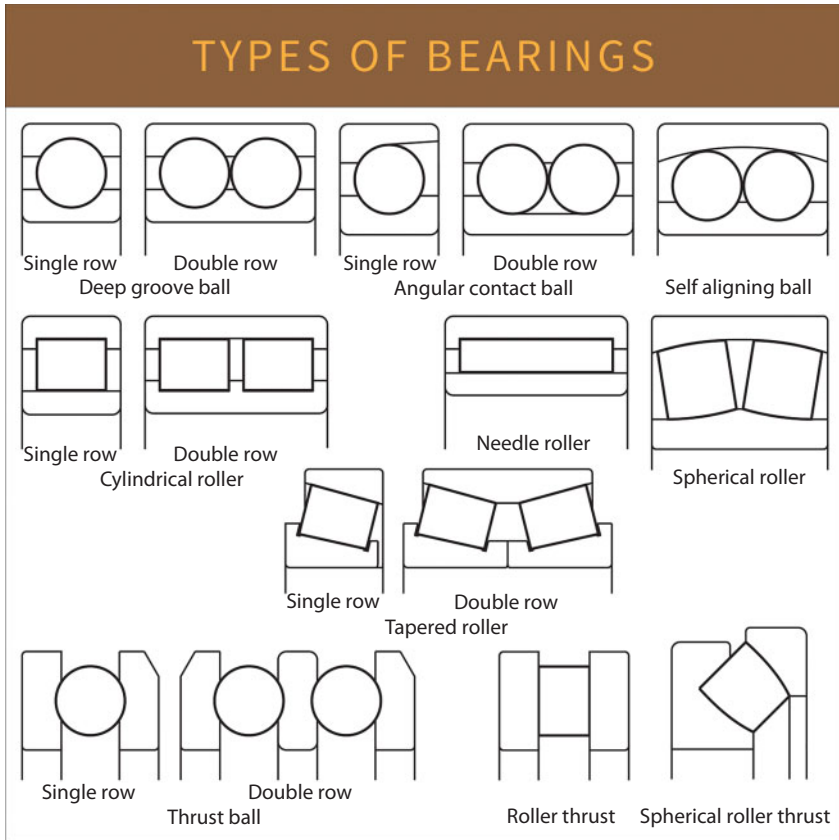


Figure 16.2 Different types of rolling element bearings.

16.2 Rolling Element Bearings

Rolling element bearings fall into a large category of rotor support bearings that include ball, roller, and tapered roller designs (see Figure 16.2). They can be lubricated by either grease or oil, depending on the manufacturer's recommendations. If a rolling element bearing is known to be running very hot, do not put a water hose on the outside of the housing. The water could get into the lubricant, causing a premature failure. The cooling water on the outer housing will shrink the outer housing and may cause the bearing to seize immediately, or at least have unintended consequences.

Bearings play crucial roles in machine reliability and performance. Bearings maintain the position of rotors, so that internal clearances are maintained for efficient operation, while preventing the contact between the rotating and stationary parts. A bearing failure will lead directly to a machine failure.

There are many different types of bearing designs. Plain and rolling element bearings are two of the most common in process machinery.

16.3 Plain Bearings

Plain bearings, also referred to as journal bearings or bushings, represent a general category of radial bearings that are the simplest in terms of design. They consist of a bearing material that is usually stationary and a rotating shaft. The rotation of a journal within a journal bearing creates a wedge-shaped oil film that generates a pressure region, which supports the journal load. (More about fluid film lubrication can be found in Chapter 17 of this book.)

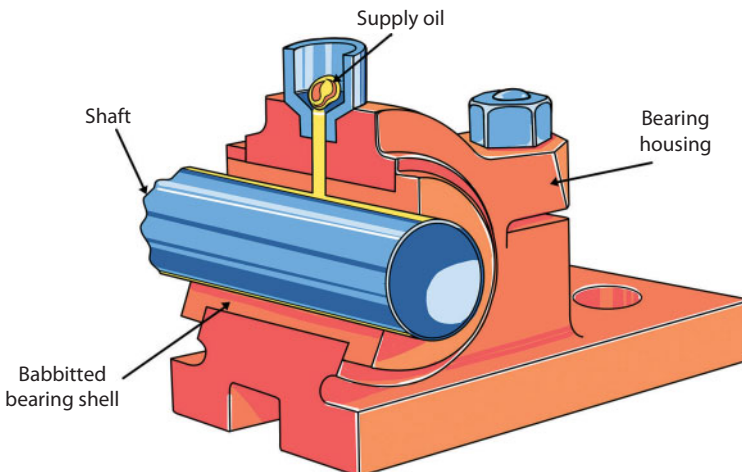


Figure 16.3 Cutaway of a plain (journal) bearing and shaft.

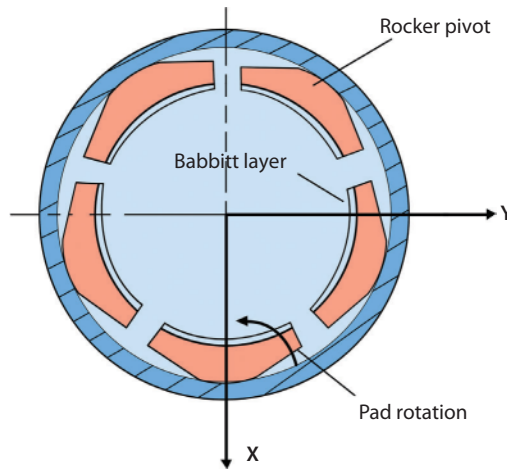


Figure 16.4 Tilting pad bearing.

16.4 Compressor Bearings

The most common radial bearings used in centrifugal compressors are the tilting pad type (see Figure 16.4), which are continuously lubricated with light turbine oil. Before tilting pad designs, sleeve type bearings were commonly used. The tilting pad bearing design provides rotor-dynamic characteristics that help assure smooth and reliable mechanical operation. Radial bearings are sized to be large enough to support the rotor weight, yet small enough to operate at sufficiently low peripheral speeds required to limit operating temperature to acceptable levels.

In early compressor designs, the cylindrical hydrodynamic journal bearings were the only design available. They have a cylindrical bore, typically with two axial grooves for lubrication. Journal bearings have a high load capacity, and the simple design is compact, bi-rotational, and easy to manufacture. However, as the design speeds of machines increased, it was found that journal bearings had limitations related to oil whirl. Oil whirl is a very undesirable condition that leads to high vibration amplitudes, forces, and cyclic stresses that are

imposed on the shaft, bearings and machine. Efforts to suppress and eliminate oil whirl have resulted in a variety of fixed geometry bearings modifications to the profile of the bearing bore. Variations are the lemon bore, pressure dam, lobed, and other fixed profile bearings. Eventually the tilting pad concept was developed and first applied to journal bearings approximately 75 years ago. A tilting pad journal bearing is comprised of multiple, segmented babbitted pads that can move freely around a shaft journal. Extensive tests and applications have proved the tilting pad journal bearing to be most effective in eliminating oil whirl.

16.5 Modeling Fluid Film Bearings

Today, there is a vast array of fluid film bearing designs employed in compressors. Every type has its advantages. For example, the plain cylindrical bearing is simple and relatively cheap compared to a tilting pad bearing, but it can be prone to bearing instabilities. The list of bearings in Figure 16.5 shows the various type of bearings in order of the most susceptible (plain cylindrical) to the least susceptible (tilting pad) design prone to bearing instabilities that are available to the compressor designer. Bearings are extremely important to the rotor-dynamist because they are often the best means of changing system stiffness and damping into the rotor-dynamic system to avoid critical speed problems.

To adequately model a compressor rotor/bearing system, the analyses must contain the following four essential elements: 1) the rotating shafts with distributed mass and elasticity, 2) rotating disks, 3) bearings, and 4) the most common synchronous excitation – mass imbalance. The hope is that the compressor model will:

- Determine the vibration response of the machine during steady state operation

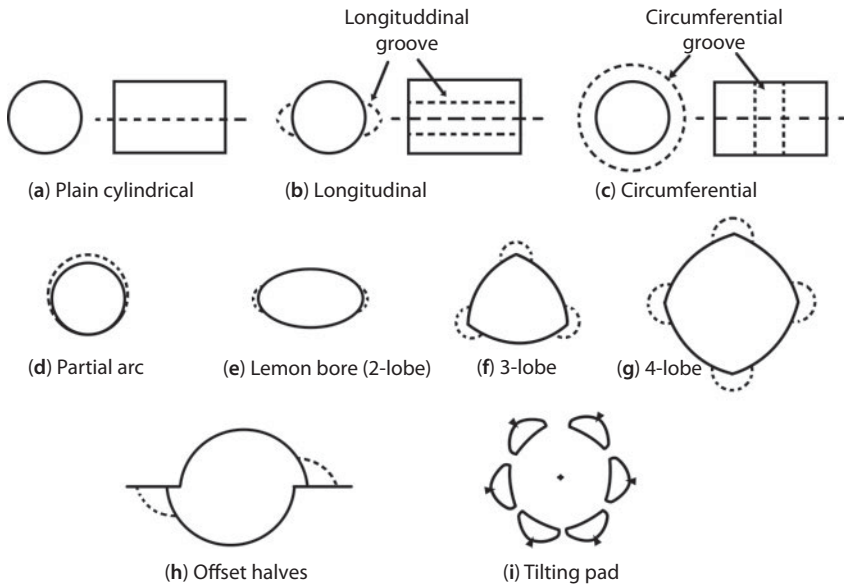


Figure 16.5 Here are some of the many fluid film bearings available to turbomachinery designers. The plain cylindrical bearing is the cheapest, most basic design and the tilting pad bearing is the most expensive and most complex design.

- Determine clearances at seals, blade tips, bearings during steady state operation
- Determine peak vibration response during start-up and coast downs
- Determine vibration response of machine if operating conditions change
- Check for the stability of the rotor-bearing system under full load conditions

The last item, stability check, is important for the machine to avoid unpredictable vibrations that grow in an uncontrollable manner over time. Stability problems are very complex and difficult to eliminate once they are encountered in the field.

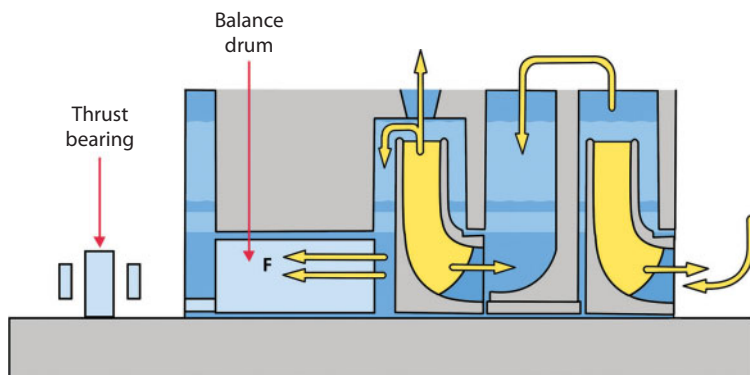


Figure 16.6 Balance piston or drum.

16.6 Thrust Loads

The pressure rise in each of the stages of a centrifugal compressor creates an axial thrust force that acts down the compressor shaft, in an axial direction. Depending on the overall pressure rise in the compressor, these thrust forces can be significant. An inline impeller configuration employs a thrust balance drum (balance piston) to generate a thrust force that opposes (“balance”) the sum of the impeller thrust forces. Typically, located at the discharge end of the compressor, the balance piston is a simple disc-shaped element installed on the compressor shaft that is equipped with a seal around its outer diameter. In the Figure 16.6 below, the pressure to the right of the balance piston is subjected to the compressor discharge pressure and the left side of the balance piston is subjected to compressor suction pressure, plus any pressure created by losses in the auxiliary thrust balance line. The resulting pressure differential across the balance piston creates an axial force toward the discharge end that opposes the impeller thrust forces. Proper selection of the balance piston diameter results in small net thrust force and allows that use of a reasonably small thrust bearing to absorb the residual thrust forces to maintain proper rotor axial positioning during operation.

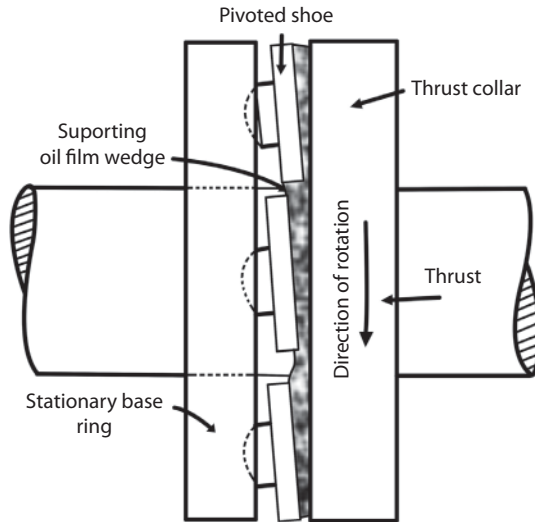


Figure 16.7 Kingsbury thrust bearing.

Like the radial bearing, the thrust bearing is usually a tilting-pad design lubricated with light turbine oil. Some thrust bearing designs employ a system of leveling blocks behind each tilting pad to ensure uniform load distribution. As with the radial bearings, magnetic thrust bearings also are available.

16.7 Kingsbury Thrust Bearing

Tilting pad thrust bearings (Figure 16.7), also known as Kingsbury thrust bearings, are designed to transfer high axial loads from a rotating shaft to a stationary outer casing with minimum power loss. The load in a Kingsbury bearing is transmitted through multiple wedge-shaped oil films formed between the shaft thrust-collar and a series of stationary pivoted pads or segments. This brilliant design results in an extremely robust and reliable bearing with an extremely low coefficient of friction. If designed and applied properly a Kingsbury thrust bearing can provide decades of reliable service.

Fluid thrust bearings contain a number of sector-shaped pads, arranged in a circle around the shaft, and that are free to pivot (see Figure 16.7). These pads create wedge-shaped regions of oil between the pads and a rotating disk, which support the applied thrust and eliminate metal-on-metal contact. Bearing lubrication for these types of bearings can be either fully flooded or oil spray directed to each individual thrust shoe.

16.8 Compressor Seals

All process gas compressors use some type of seal either to prevent the flow of high-pressure gas to the atmosphere or to improve its efficiency by preventing gas leakage between compressor stages. Seals for rotating shafts fall into the following general categories: labyrinth seals, wet seals, and dry gas seals. Sealing for reciprocating rods is referred to as packing.

16.8.1 Labyrinth Seals

A labyrinth seal has a series of teeth with a close clearance to the shaft (See Figure 16.8). As the high-pressure gas passes through the first tooth the pressure is decreased to approximately half its original pressure and velocity is increased. The gas then enters the chamber before the second tooth. A natural turbulence is created, which reduces the velocity allowing the gas to expand in the cavity between labyrinth teeth. This process is repeated for each tooth of the labyrinth. The result is a large reduction in gas pressure across the seal.

16.8.2 Oil Film Seal

An oil film seal is made up of two free floating rings that are machined with a very small clearance around the shaft (see Figure 16.9). Clean high-pressure oil is pushed between the rings and shaft, and out both directions. If the oil pressure is

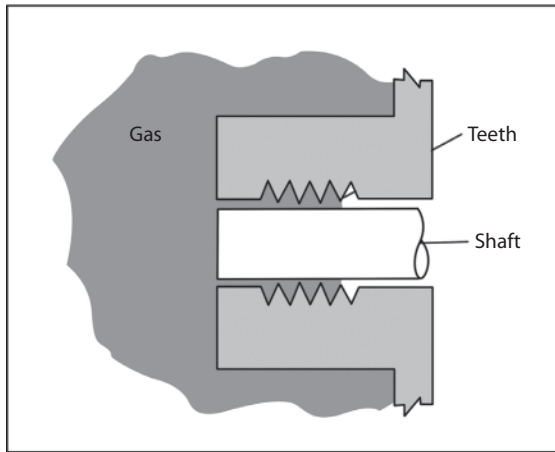


Figure 16.8 Labyrinth seal.

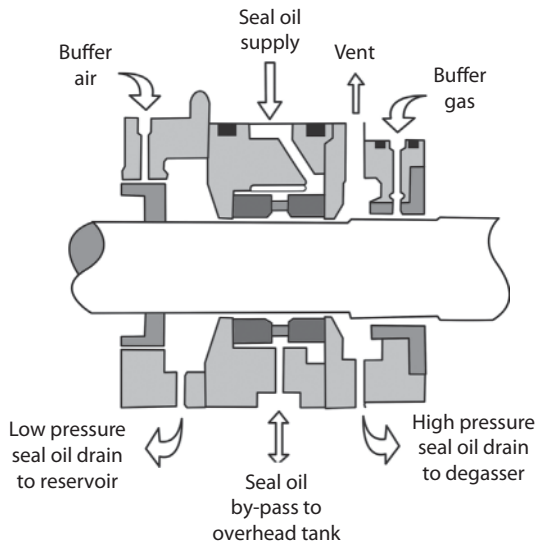


Figure 16.9 Oil bushing seal.

always greater than the gas pressure, the seal will not allow gas to escape.

Seal oil is supplied and flows between the seal rings (see Figure 16.10). Most of the oil will flow to the low-pressure side of the seal cartridge. The high-pressure side of the seal cartridge is where the gas and oil contact one another. The labyrinth seal

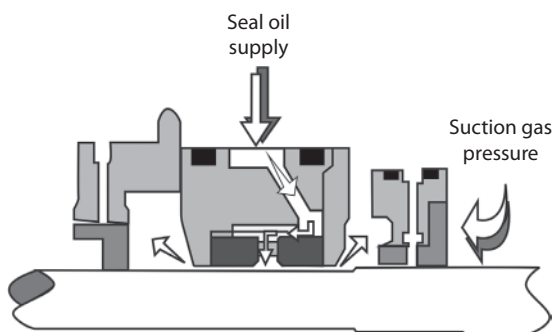


Figure 16.10 To prevent process gas from leaking, the seal oil pressure must always be maintained 5 to 10 psi (or more depending on the seal design) above the compressor suction pressure.

on the high-pressure side of the seal cartridge is designed to reduce the higher gas pressure to suction pressure. This reduced gas pressure is sealed from the system by the high-pressure oil passing through the gas side seal ring. Oil supply between the seal rings is maintained at a pressure greater than compressor suction pressure, at all times, to insure a positive seal and safe operation.

16.8.3 Face Contact Wet Seals

A compressor mechanical contact or oil face seal (Figure 16.11) uses a spring-loaded stationary carbon ring that is in sliding contact against a rotating ring manufactured from high-quality material and with a special surface finish. Face type seals work on the same principle as pressurized barrier mechanical seals in pumps but are different in that their operating speeds are much higher than seals found in pumps.

Wet seals require a liquid barrier, in most cases lubricating oil, to prevent the gas from leaking to the atmosphere. The seal oil pressure applied to the seal faces must be carefully controlled so that is always above the compressor suction pressure. If the seal oil differential is lost the seal will fail prematurely.

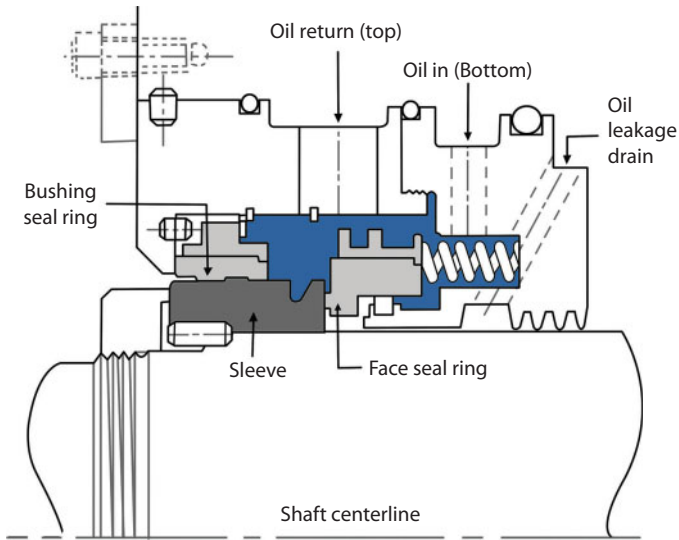


Figure 16.11 Compressor face seal.

16.9 Seal Oil System

The primary purpose of the seal oil system is to keep the compressed gas from entering the bearing assembly and escaping from the compressor to the atmosphere. The seal oil system may be a completely separate oil supply system; however, in most cases it is a part of the compressor (main) lube oil system. The seal oil system can be considered as having three major levels of operation: Seals, Oil Supply, and Oil Return.

16.10 Dry Gas Seals

Dry gas sealing systems are now a common part of most modern centrifugal process compressors (see Figure 16.12). A dry gas seal consists of a rotating, hard-face mating ring with either a machined or etched circumferential spiral groove pattern and a primary (stationary) ring in softer material. The primary ring is radially restrained but can move axially. Under static and depressurized condition, the springs behind the primary ring

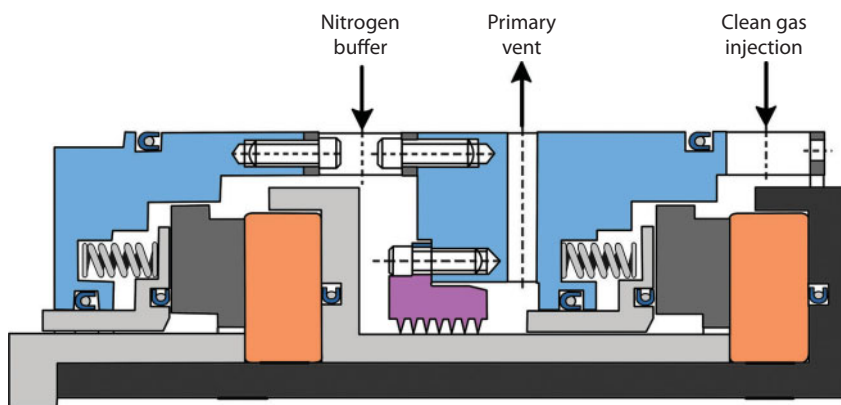


Figure 16.12 Dry gas compressor seal.

keep the seal faces closed and in contact. In the static and pressurized condition, the sealed gas travels across the faces at the tip of the groove. A sealing dam maintains uniform pressure distribution between the seal faces and helps to provide the hydrostatic lift that results in a low start-up torque and lower heat generation.

In the dynamic condition, the progressively shallow spiral grooves draw the sealed gas toward the center dam and create gas film pressure to separate the faces, so they become non-contacting during operation. Special geometry of the spiral grooves provides a uniform hydrostatic and hydrodynamic pressure distribution and maintains the proper seal face gap for gas film stiffness.

16.11 Seal Gas Quality and Control

Seal gas entering the primary seal area must be clean and dry; it should be 99.98% free of entrained liquid particles three microns and larger and be filtered to less than 10-micron solid particles. In addition, designers need to ensure that at least 36 °F (20 °F) dew point margin (superheat) is maintained throughout the dry gas seal system to prevent liquid from forming inside

the seal. The temperature of the seal gas must be measured at the point of seal gas entry to the seal, not at the source of seal gas supply.

The flow of seal gas to a primary seal is controlled by either a flow control method or a differential pressure control method. The main objective of both types of control methods is to positively sweep the seal gas across the process gas labyrinth to prevent a reverse flow of process gas into the dry gas seal.

A dry gas seal control system must have an adequate range and controllability to maintain a minimum 16 fps (4.9 m/s) gas velocity across the inner process labyrinth from the minimum labyrinth design clearance up to two times the maximum labyrinth design clearance.

16.12 Reciprocating Compressors – Packing

Packing provides the dynamic seal between the cylinder and the piston rod. It consists of a series of Teflon rings mounted in a packing case, which is bolted to the cylinder. The piston rod moves in a reciprocating motion through this case. The packing case is constructed of several pairs of rings (see Figure 16.13).

The gas pressure is higher on one side of each ring, which compresses the rings against the sealing area. Each pair of rings consists of one radial cut ring and one tangential cut ring. The radial cut ring is installed toward the cylinder (pressure) side. Gas flows around the front face of the radial cut ring and then around the outside of both rings. Since the ring outside diameter is greater than the ring inside diameter, a squeezing force is exerted on the rod, which seals the path between the rings and the rod. The radial cuts are positioned in the ring assembly so that they do not line up with the tangential cuts. Cylinder pressure will force the ring assembly against the packing case lip, thus preventing flow around the rings.

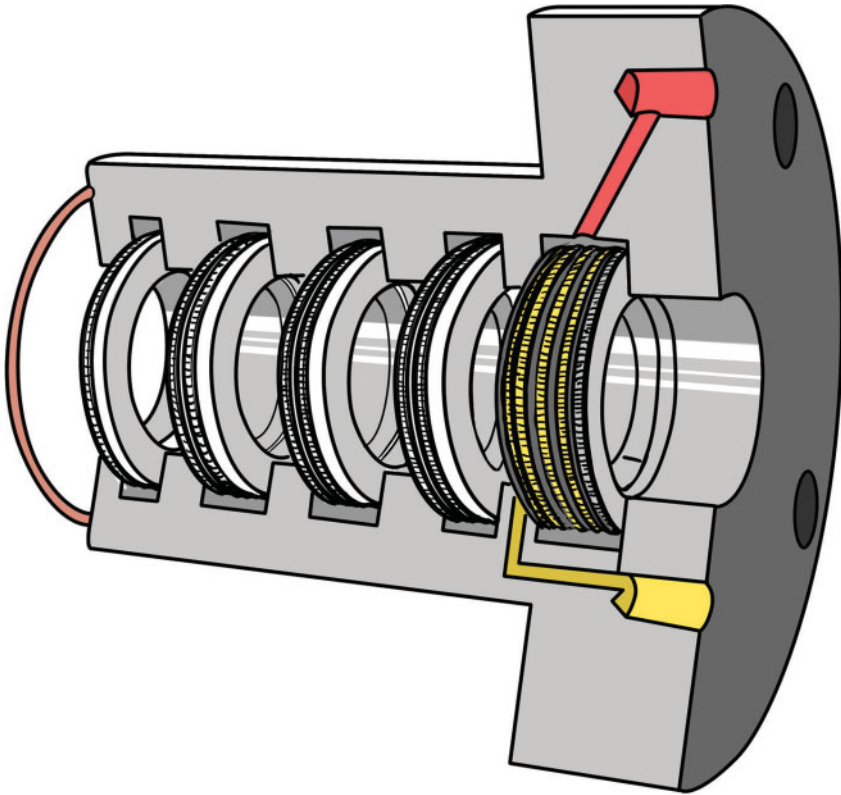


Figure 16.13 Typical reciprocating compressor packing box.

The amount of pressure differential one set of rings can withstand is limited. Therefore, several pairs must be installed to handle typical field gas compression applications.

Packing lubrication is needed to reduce friction and provide cooling. Lubricating oil, which must be finely filtered to prevent grit from entering the case, is generally injected in the second ring assembly. The pressure differential moves the oil along the shaft to lubricate the remaining packing rings.

17

The Importance of Lubrication



Lubrication is one of the most important aspects of any rotating equipment reliability program. Lubrication-related failures are the most preventable type of machinery failures. With sufficient attention to details, they should be rare events. It is

normally through the neglect of lubricants and lubrication systems that equipment fails prematurely. As a machinery operator your vigilance plays a central role in maximizing the useful life of a machine's lubrication system.

The lubricants and sealing methods available today have greatly improved the reliability of both pressurized, non-pressurized, sump and splash type lubrication systems. All machines require lubrication of their radial and axial bearings in order to maintain the rotor's position relative to the casing and to ensure reliable operation. Lubrication serves the following critical functions:

- Reducing wear by separating moving surfaces
- Reducing friction between the rotating and stationary components
- Absorbing shock
- Dampening noise
- Carrying heat generated by friction within the bearing
- Removing heat transmitted down the shaft from the process end of the machine
- Minimizing corrosion
- Keeping contaminants away from the bearing components
- Flushing contaminants away from bearings
- Acting as a sealing medium

The major lubricated components in a centrifugal compressor train include:

- Drivers
 - Electric motor
 - Steam turbine
 - Gas turbine
- Gearboxes
- Couplings (gear type)

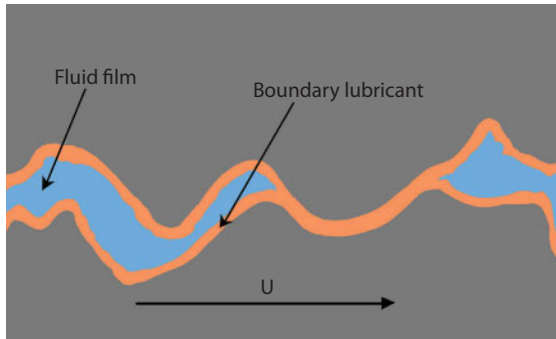


Figure 17.1 Boundary lubrication.

- Compressor
 - Radial journal bearings
 - Tilt pad radial bearings
 - Thrust bearings
 - Seals (oil bushing type seals)

17.1 Lubrication Regimes

There are four basic fluid film lubrication regimes encountered in machinery, listed here from thinnest to thickest:

1. **Boundary lubrication**, where surfaces are in contact with each other even though a lubricant is present (see Figure 17.1). This is generally an undesirable operating regime for a fluid film bearing since it leads to increasing friction, energy loss, wear, and material damage. If designed properly, most machines will experience boundary lubrication only during start-ups, shutdowns, and low speed operation. Special lubricants and additives have been developed to lessen the negative effects of this regime. When constant starts and stops create boundary lubrication the life of a machine is reduced.

2. **Mixed lubrication**, where surfaces are partially separated
3. **Elastohydrodynamic lubrication**, where two surfaces are separated by a very thin lubrication film. Elastohydrodynamic lubrication is employed in most rolling element bearings. Think of a balloon sitting on a table. As the balloon is pressed on the top the contact area on the table will increase, allowing the support of more load by reducing the unit loading. "Elasto" in the name means parts can deform elastically; "hydro" refers to a liquid such as oil, and "dynamic" means it is moving.
4. **Full fluid lubrication**, also called hydrodynamic lubrication, is where two surfaces are completely separated by a fluid film (see Figure 17.2). Full fluid lubrication is essential for the long-term reliability of high-speed, fluid film bearings. A full fluid film is formed when the bearing geometry, shaft speed, and oil viscosity combine to generate sufficient pressure to support the full bearing load. This is self-generating pressure that results in the complete separation of the shaft from the bearing (see Figure 17.3).

Most high-speed compressor bearings employ full fluid lubrication in their bearings.

The principle of fluid film bearing operation is based on Professor Osborne Reynolds's theoretical investigation of cylindrical journal bearings. He showed that oil, because of its adhesion to the journal and its resistance to flow (viscosity), is dragged by the rotation of the journal and forms a wedge-shaped film between the journal and journal bearing (Figure 17.3). This action sets up the pressure in the oil film, which thereby supports the load. This wedge-shaped film was shown by Reynolds to be the essential feature of effective journal lubrication. Reynolds also

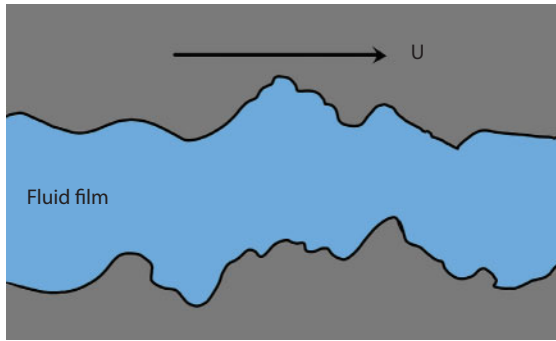


Figure 17.2 Full fluid lubrication.

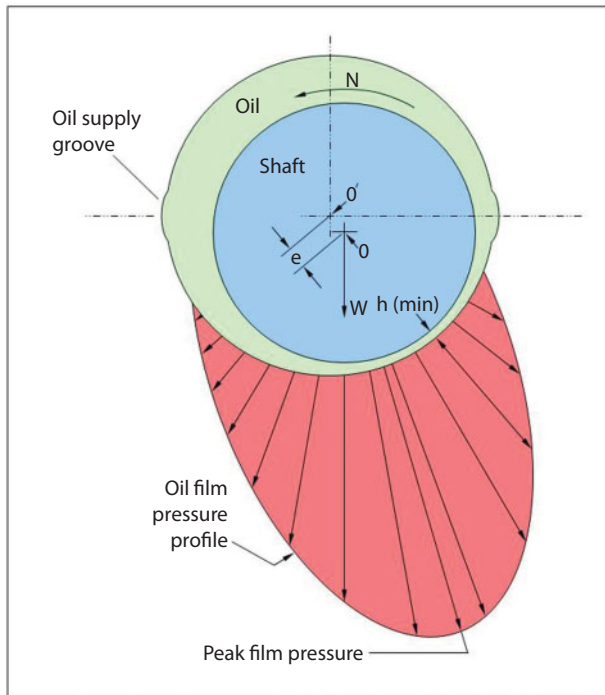


Figure 17.3 The rotation of a journal within a journal bearing creates a wedge-shaped oil film, $h(\min)$, that generates a high-pressure region, which supports the journal load.

showed that “if an extensive flat surface is rubbed over a slightly inclined surface, oil being present, there would be a pressure distribution with a maximum pressure occurring somewhere beyond the center in the direction of motion.”

17.2 Lubricating Oils

Machinery lubrication can be applied in a variety of ways depending on the equipment design, speed, and operating conditions. The machine's manufacturer normally determines the type and quantity of lubrication required. While numerous types of lubrication are available, most compressor manufacturers require the use of some form of oil to lubricate bearings and seals.

Lubricating oil, sometimes simply called lubricant/lube, is a class of oils used to reduce the friction, heat, and wear between mechanical components that move relative to one another. Oil is commonly used in critical process machinery operating at speeds greater than 1500 rpm. There are two basic categories of lubricating oil:

1. *Mineral* oils:
 - a. Are the most commonly used industrial lubricants due to their relatively low cost.
 - b. Are refined from naturally occurring crude oil.
 - c. Have moderate oxidation and viscosity stability, and cost \$12 per gallon or more.
 - d. Can be manufactured to have a varying viscosity, therefore making them useful in a wide range of applications.
2. *Synthetic* (man-made) oils:
 - a. Are manufactured, which is why they tend to be costlier than mineral oils.
 - b. Are used in the more demanding machine applications.
 - c. Cost about \$32 per gallon or more.
 - d. Tend to have very good oxidation and viscosity stability properties.

17.3 Compressor Lubricating Oil Systems

A lube oil system (Figure 17.4) is designed to supply oil to the compressor and driver bearings and to the gears and

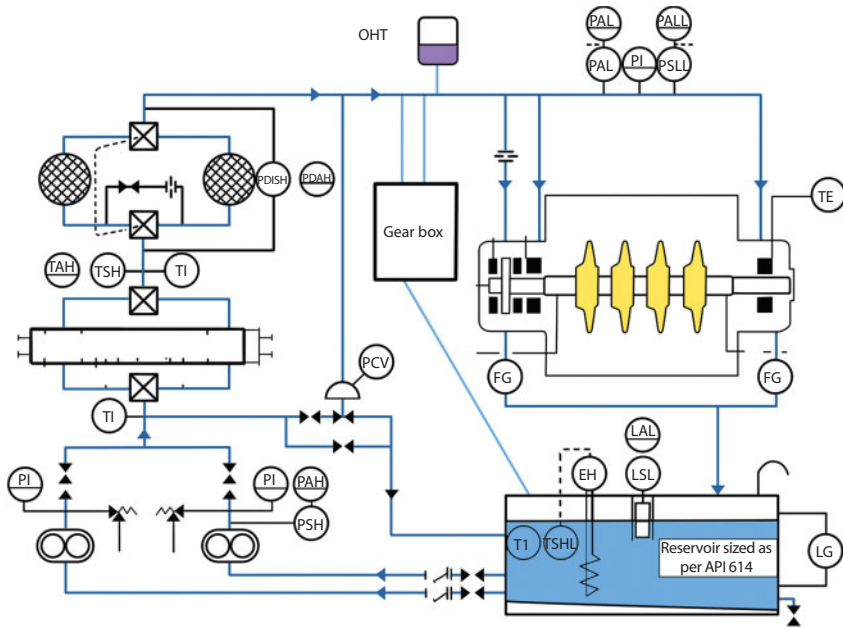


Figure 17.4 Typical compressor lube oil system.

couplings. The lube oil is drawn from the reservoir by the pumps and is fed under pressure through coolers and filters to the bearings. Upon leaving the bearings, the oil drains back to the reservoir.

The oil reservoir is typically designed to permit circulation of its entire fluid volume between eight to 12 times per hour. Oil reservoirs often have 1) thermal sensors for monitoring temperature levels during start-up and constant operations and 2) oil temperature controls that provide for preheating during cold start-up conditions and cooling to prevent overheating during peak operating cycles. Oil reservoirs can be either pressurized with nitrogen or vented.

When in operation, the compressor lubricant oil is normally circulated by the main oil pump with an auxiliary pump in standby mode. The two oil supply pumps generally have different types of drive or power sources to ensure reliability.

When both are driven electrically, they are connected to separate supply feeders to prevent dual power outages. Relief valves protect both pumps from the effects of excessively high pressures. Check-valves prevent reverse flow of oil through the idle pump.

Heat generated by friction in the bearings is transferred to the cooling medium, which is usually cooling water, in the oil coolers. Air-cooled oil coolers may be employed as an alternative to water-cooled oil coolers. A pressure-regulating valve is controlled by the pressure downstream of the filters and maintains constant oil pressure by regulating the quantity of bypassed oil.

A pressure switch activates the auxiliary oil pump. If the oil pressure falls below a preset limit, a second pressure switch shuts down the compressor train. Filters clean the lube oil before it reaches the lubrication points and a differential pressure gauge monitors the degree of fouling (flow restriction) of the filters.

The flow of oil to each bearing is regulated individually by orifices, particularly important for lubrication points requiring different pressures. Lube oil for the driver and other mechanical components is taken from branch lines. Temperatures and pressures are measured at all important locations in the system, including temperatures from oil sumps, return lines from bearings, gears and other mechanical components. The readings can be taken locally or recorded for trending purposes.

In addition to checks of the circulating system, additional checks should be made, such as "Are both the main and auxiliary oil pumps running?" If they are both running, ask yourself some additional questions: "Why are both pumps running? What is the system pressure? Is it normal?" Touch the discharge side of the relief valve on the discharge of the pump. Is it relieving? It should not be.

17.3.1 Lubrication Monitoring

Have the following recommendations regarding lubricant properties and oil levels in mind when inspecting and monitoring your lubrication systems:

- Check lubrication system pressures and levels daily.
- Immediately investigate any significant reservoir levels changes. A high level can be as important as a low level.
- An increase in system volume is cause for immediate attention. Likely scenarios for an unplanned volume increases are the ingestion of an unwanted liquid, such as a process liquid, coolant, etc., or the cross-contamination of lubricant by an unknowing but well-intentioned operator. The operator should always be aware of abnormal fluid levels and other peculiar conditions for the machines he monitors.
- To ensure your compressor lubricant possesses the right properties for reliable operation, oil samples must be taken and then checked periodically. Among the oil analysis tests that should be performed are an elemental analysis, a viscosity analysis, and a wear debris analysis. (Note: The original equipment manufacturer may recommend other lubrication tests. Based on their experiences.)
- Oil analysis for centrifugal and axial compressors should also include annual ferrography, filter-debris and varnish tests.
- Pay close attention to any changes in the lubricant's smell, color, and level. A dramatic change in color may indicate contamination with water (see Figure 17.5).



Figure 17.5 How oil appearance can change as it becomes contaminated. Remember to look for changes from newly installed oil.

- Look for foaming in oil return lines and level gauges (see “Oil Foaming” section below)
- If the color has changed significantly from your last inspection or from what you know to be normal, find out why.

1. Sample #1 shows ideal oil that is golden and fully transparent.
2. Sample #5 shows a sample that is discolored.
3. Sample #6 shows oil that is no longer transparent and has darkened.
4. Sample #7 is now completely dark due to oxidation.
5. The “Watery” sample (on the far right) is seriously contaminated with water.

17.4 Oil Foaming

Foaming (see Figure 17.6) in oil is mainly due to the accumulation of small air bubbles at the surface of the lubricant. It is caused by excessive agitation, inadequate levels of lubricating oil, air leaks/ingress, contamination, or cavitation.

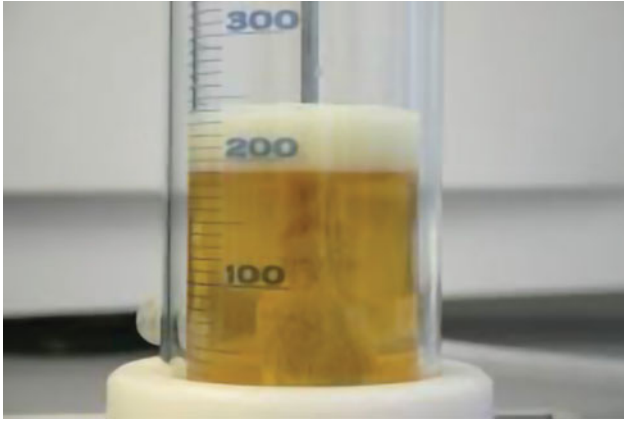


Figure 17.6 Oil Foaming.

17.4.1 Excessive Foam

- Lead to a misinterpretation of the oil level and subsequently result in machinery failure.
- Act as a thermal insulator making the oil temperature difficult to control.
- Be a major cause of overheating, loss of pump pressure, loss of power, cavitation, oxidation and failure of hydraulic systems.
- Have a direct impact on the lubrication of the engine/hydraulic systems by creating air buffer zones in the circuit, which degrade the lubricating properties of the oil.

To prevent or reduce the formation of foam, lubricants contain anti-foaming additives, mostly silicon-based additives.

There are many reasons why oil tends to foam. The most common reasons for foaming are:

- Contamination is a very frequent cause. Common contaminants consist of water, solid particles, grease or cross contamination of the oil with another fluid or addition of a wrong lubricant.

- Depleted anti-foaming additives (possibly due to the use of excessively fine filtration and electrostatic separation technologies).
- Mechanical issues causing excessive aeration of the fluid, leaky seals, etc.
- Contamination of the lubricant with grease.
- Overfilling of the sump with a splash- and bath-lubricated compartments.

18

Inspection Ideas for Operators and Field Personnel

“It is not enough to do your best; you must know what to do, and then do your best.” (W. Edwards Deming)

18.1 Equipment Field Inspections

Providing operators with simple, hands-on diagnostic tools and the training required to use them effectively affords them the ability to spot developing machinery issues before they result in unplanned shutdowns (see Figure 18.1). Regardless of whether operators are performing inspections with handheld computers or with paper and pencil, your plant will not be successful unless field personnel are doing thorough inspections on their equipment. The most successful processing facilities



Figure 18.1 Operators inspecting an electric motor in the field.

in the world employ the production operators to observe their plant equipment and report observed deficiencies back to maintenance.

To be effective, every field inspection program should include:

1. A structured, periodic inspection plan for critical equipment and systems
2. A means of capturing and recording of field observations

3. A timely analysis of the recorded observations to identify mechanical anomalies
4. A means of reporting issues to maintenance personnel so that critical issues are addressed

In the next few pages, I will cover some proven inspection methods that can be used as a basis for a machinery monitoring program.

18.1.1 Audible Inspections

Audible, or sound, inspections are done during routine process rounds. Listen to the equipment and notice differences that may have occurred since the last time you were near this piece of equipment. The listening must be done regularly so changes can be detected. It is necessary to know what “good” sounds like to be able to tell what is “bad”. The following examples of observations may be telling you something:

1. A loud hum on a motor may indicate a soft foot condition. (Soft foot is a condition where a motor foot is out of plane with the others, which leads to a warping of the motor frame.) If it was quiet yesterday and noisy today, you need to seek the reason for the change.
2. Squealing belts can indicate anything from an overload condition to just loose belts. A scraping sound or rhythmic sound can indicate a dragging or rubbing problem.
3. A clicking or regularly recurring sound from inside the gear box may indicate a broken tooth. If there is a sight glass on the oil system, see if metallic particles can be detected floating in the oil or try the paper test (described later in this book).
4. Tapping heard on a reciprocating compressor usually indicates a loose valve. Use a listening device

on the covers one after another to find the loose valve. This will ensure that only the loose valve cover needs to be opened to replace the gasket and the valve tightened before the valve seat is damaged. As a temporary “fix” the hold down device for the valve may be tightened to secure the valve but it could still be leaking.

5. A failing or poorly lubricated rolling element (or frictionless) type bearing can often be detected audibly.

18.1.2 Visual Inspections

Always be on the lookout for leaks, abnormal fluid levels, burned paint, and vibrating shafts or housings. Here are some specific situations to keep in mind when inspecting:

1. Check for any signs of discolored or burned paint. Paint burns at about 400 to 450 degrees F. If the burned paint is near lubricating oil, it is likely the oil is hot as well. This high temperature indication could mean the condition of the oil is no longer viable as a lubricant.
2. Look at motor fan guards for blockage and check the fins on TEFC motors to ensure cooling can take place.
3. Ensure pressure gauges are installed and working. It is most important to know what the normal pressure is for any piece of operating equipment. Sometimes gauges are marked with the acceptable range. If this is done on the gauge glass, make sure there is another mark to indicate the correct glass position, ensuring that it has not moved and is giving a false reading.



Figure 18.2 Over-lubricated bearing.

4. Look for signs of over-lubrication, which will cause leaks and shorten equipment life. It is also a potential environmental problem and fire hazard (see Figure 18.2).

If equipment is ring oiled, it is generally easy to look at the ring while the equipment is running. This is an excellent check, because if the ring stops turning for any reason, it has the same effect as a loss of oil flow in a forced lubrication system. Improper operation of an oil ring will likely lead to a bearing failure.

18.1.3 Tactile Inspections

Tactile inspections are easy to perform and can tell you a lot about how the machine is running. But they are to be carried

out only if you are sure you will not be burned by touching the equipment. First, swipe the equipment with a brushing motion of the fingertips. If it is not too hot, gradually slow the motion to determine if it will become uncomfortable to the touch. If you can leave your hand on the equipment the following states of operation may be examined:

1. Lightly touching a piece of equipment with fingertips can provide a subjective evaluation of how smoothly it is running. With practice, manual vibration checks can be a reasonably good method of detecting mechanical changes. A tingling in your fingers when touching the machine indicates a high frequency or fast occurring vibration. It is recommended that vibration checks be done regularly in order to note changes from yesterday's or last week's vibration level.
2. If any of the surfaces are hot, ask yourself the following questions: a) Does it seem hotter than the last time you touched it? b) Is the hot spot localized or is it generally hot? This is excellent data to share with engineering or maintenance.
3. If a pump has multiple filters, such as in an oil or hydraulic system, touching the outlet of the pump and then each filter downstream can help to determine which filter is in service. The filter with the same temperature as the pump discharge is the one in service.
4. A leaking pressure relief valve can be detected by touching the discharge side of the relief valve and the inlet to the relief valve. If there is leakage or if the valve is relieving, there should be a difference in the temperatures of the two lines. If the temperatures are the same, it is likely that the relief valve is working properly.

18.1.4 Smell

Smell is another means of detecting a change in machinery. Here are examples of how your sense of smell can be used as a monitoring tool:

- If belts are loose, it may be possible to detect this situation not only by the sound but by the smell of the hot rubber as it is removed from the side of the belts.
- Burned oil has a distinct scent that can indicate a potential problem.
- Paint that gets hot enough to discolor gives a distinctive odor. Each scent may indicate a particular problem with the equipment.
- A problem can also be detected if the product processed in the plant gives off a “normal” odor and a distinctly different odor when overheated or when a problem exists.

The advantages of the audible, visual, tactile, and smell-based inspections are that they are:

- Easy to use
- Easy to train someone to do
- Always available
- Inexpensive
- Can be done by anyone

The disadvantages are that they are:

- Subjective (Hot? Very hot? Burning up? What does that mean?)
- Difficult to communicate to work request or someone else

- Difficult to repeat (Different people perceive the same thing differently)
- Unable to be used at the earliest stage of a problem

18.2 Tools Available to Quantify What You Have Detected

18.2.1 Audible Inspection Methods

18.2.1.1 Ultrasonic Gun

An ultrasonic gun is a relatively inexpensive handheld device for listening to noises in the ultrasonic range. It is used for leak detection, steam trap maintenance, and listening to bearing noises.

18.2.1.2 Stethoscope

This is an inexpensive tool that can pick up every sound that the equipment is making, though that can also be counterproductive. The good news is that it picks up everything and the bad news is that it picks up everything. It must be used regularly to detect changes from the last inspection, or it is likely to be useless in determining if there is a problem or not.

18.2.1.3 Metal Rod

A steel or aluminum rod can be used to touch the piece of equipment where the suspected noise is located, with the other end touched to the ear to listen for unusual noises. The end touching the ear should be padded, and the other end should be kept away from rotating shafts. If a screwdriver is not handy, a valve wrench or even the edge of a hard hat can be used as a listening device. The goal is to collect the sound of what is happening inside rotating equipment, listening for potential problems.

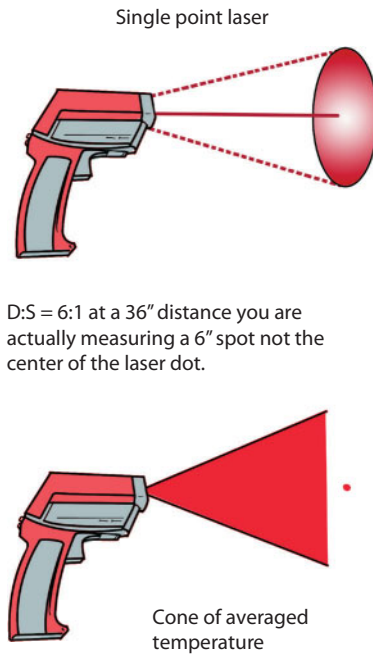


Figure 18.3 Infrared temperature gun.

18.3 Visual Inspection Methods

18.3.1 Infrared or IR Gun

Infrared guns are easy to use, non-contact temperature measuring devices. However, they have some limitations and are often misunderstood. One Important limitation of an IR gun is that the actual sensing area of the device is cone shaped. While the laser pointer indicates only the very center of the cone, the temperature being measured can be quite large depending on how far away the IR gun is from the item being measured. The temperature reading is an average temperature of the entire measurement cone. The effect of the ever-widening sensing cone is that the further away you are from the item of interest, the larger the surface area that will be averaged into the displayed temperature reading. (Refer to the upper images in Figure 18.3.)

Takeaway: The closer the IR gun is to the object being measured when the readings are taken the more accurate the temperature reading will be.

The first surface that the device encounters is the one that will be measured. If you are trying to measure the temperature of something behind a glass or plastic cover, the temperature of that cover is what will be measured. If you wish to obtain accurate surface temperatures, then the device must be used on flat, dark-colored objects. IR guns return very low readings on shiny surfaces. It is always most thorough to take a reading on a piece of paper that is at the temperature of the surroundings you wish to take readings. This will represent the “ambient” readings.

For the best temperature measurement results, paint a black spot on the point of interest and use your IR gun as close as possible to the black mark. This method is the only sure way to get useful, repeatable readings. Figure 18.3 indicates the measuring area increases the further from the IR gun you are. For best results hold it as close to the object as you can.

18.4 IR Camera

An IR or infrared camera is a device that forms an image using infrared radiation, similar to a common camera that forms an image using visible light (see Figure 18.4). The temperature of objects in the output screen may be depicted in a gray or color scale range, depending on the type of IR camera you are using. A scale indicating what the displayed gray or color scales mean is usually in the output screen as an aid to help the user determine the temperatures of the various points being surveyed. What may appear to not be a problem with the naked eye may be seen very differently with the infrared camera.

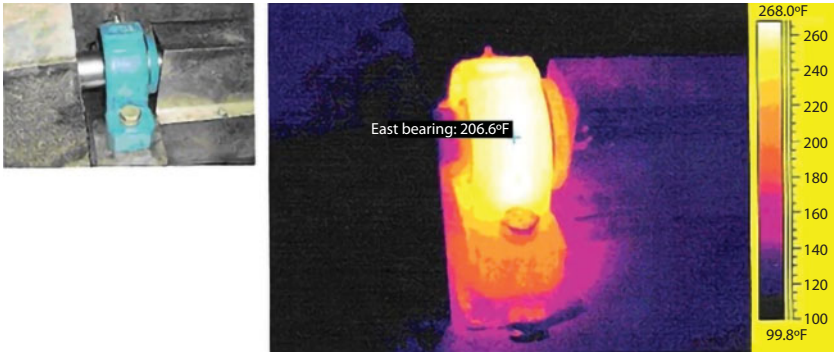


Figure 18.4 IR camera picture.



Figure 18.5 Pocket strobe light.

18.4.1 Strobe Light

This device has a high intensity light that can be controlled to a specific flash rate (see Figure 18.5). When an object is turning at a specific speed and the strobe flash rate is tuned to the same flash rate, the moving object appears to be stopped. This capability allows a good visual inspection to be done while a piece of equipment is turning at running speed. Issues like broken shim

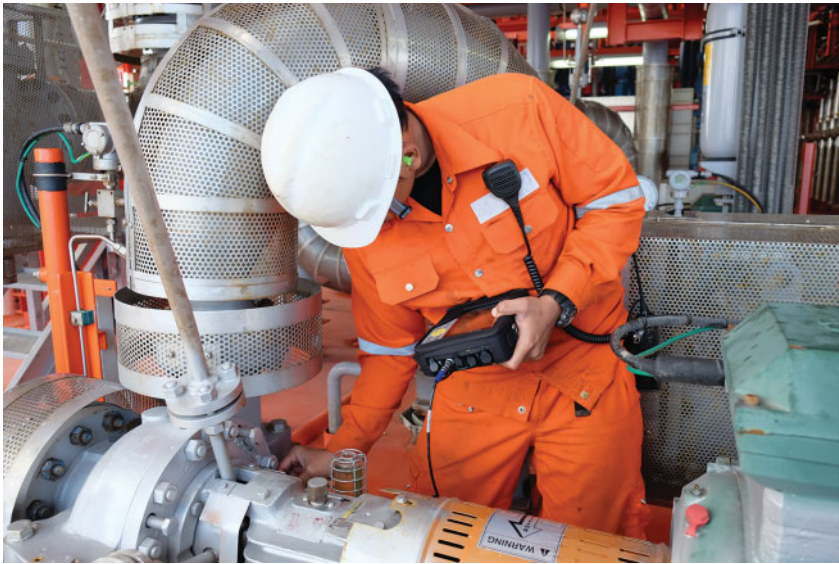


Figure 18.6 Technician using vibration meter with an accelerometer.

packs in couplings, missing keys, or broken fans on electric motors can all be inspected without having to stop the piece of equipment. With a small mismatch of flash rate the strobe will make the object viewed appear to rotate slowly so an inspection of the entire circumference can be inspected. If a spinning object has an unknown RPM, by adjusting the strobe through a range the speed can be determined.

18.5 Inspection Methods Using Vibration and Temperature Measurement Equipment

18.5.1 Vibration Meter with Accelerometer

A hand-held vibration meter (see Figure 18.6) with a mag-based accelerometer can be used to collect overall casing vibration readings as a means of gauging the mechanical condition of smaller machines, such as pumps and blowers. This additional monitoring tool can augment the five senses by providing

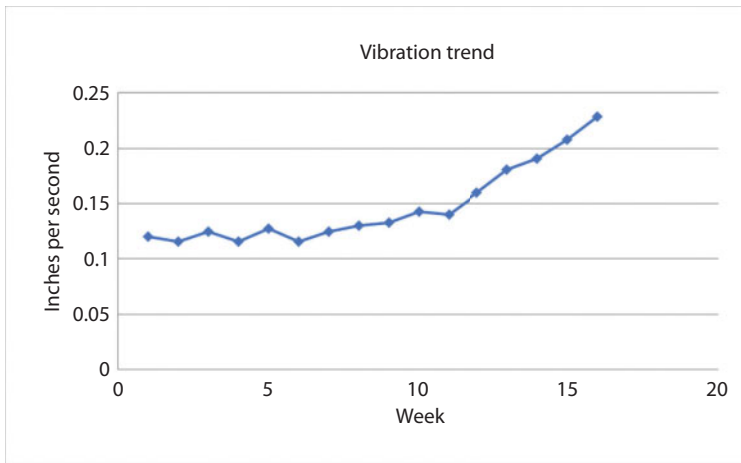


Figure 18.7 Vibration trend example.

information that is objective instead of subjective. Everyone that uses the same type of vibration equipment in the same place should get the same value; if 50 people touch the same piece of equipment with their hands in the same place, there are likely to be 50 different interpretations of how much vibration there is.

Figure 18.7 illustrates how a vibration trend can be used to identify a problem. In this example, vibration readings have been collected and plotted on a weekly basis. The magnitude of vibration is reported in units of velocity (inches per second). It is easy to see that in week 10, a detectable change in vibration occurred. Trend plots are useful because they provide visual representations of the measured parameter (e.g., vibration, temperature, pressure, etc.) over time, which can be helpful in troubleshooting and the decision-making processes. Suppose a step change in vibration occurred at the same time as a change in the process; this fact may point to a correlation between the two events that should be investigated. Always keep in mind that a gradually increasing trend plot may indicate either a deteriorating internal or external component and should be investigated once detected. Most measurements are in velocity

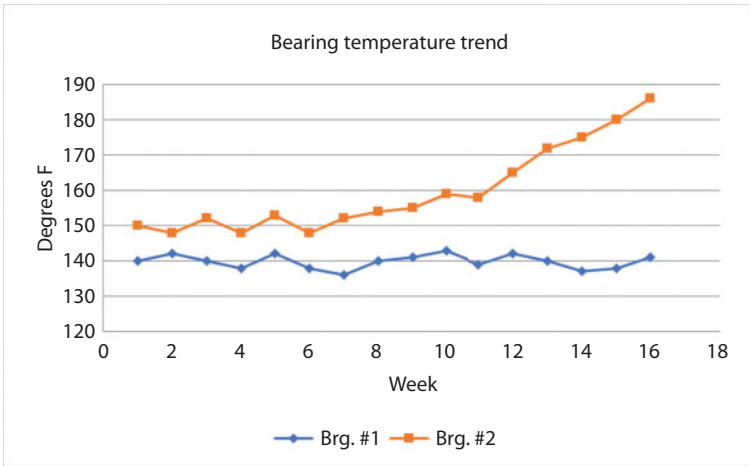


Figure 18.8 Temperature trend.

and as a general rule under .2 inches per second (IPS) is acceptable. The range of .3 IPS and higher is beginning to get rough. Generally, the lower the number the smoother the machine is operating. There must be a trend as below to know what is acceptable and when it is not.

18.5.2 Temperature Measurement Equipment

A comparison of subjective versus objective evaluations also holds for temperatures measured by touch versus a calibrated temperature device. If a quantitative analysis is required, a contact thermometer is an inexpensive and accurate tool to measure temperatures. It is simple to use and will give repeatable results no matter the color of the surface being measured as long as it is placed in the same place each measurement.

Figure 18.8 shows weekly temperature trends of two different bearing temperature measurements. It is easy to see that the two bearings seem to track together. However, when a bearing problem surfaces in bearing #2, its temperature begins to increase relative to bearing #1 after week 10. This simple visual trending method can help to identify developing machine issues.

These techniques visually explain the need to know what a given reading was yesterday, last week, last month, etc.

18.6 Generic Monitoring Guidelines

When dealing with a diverse machine population, it is difficult to provide field personnel with absolute operating limits for every machine type they may encounter. Here are some generic machinery monitoring guidelines that the author has found to be appropriate for most types of rotating machinery found in process facilities:

- Oil temperatures should generally be above 100 °F before starting any rotating equipment.
- Temperatures above 180 °F should be investigated further to determine if there is an identifiable cause, such as a cooler issue, failing bearing, etc.
- Bearing housing vibration levels below 0.2 IPS (inches per second) are considered generally acceptable.
- Bearing housing vibration readings above 0.3 IPS vibration should be considered rough and outside the normal range. Look at past vibration readings to determine if your reading is normal or elevated. If the reading is elevated, begin trending readings to see if an upward trend is developing.
- Bearing housing vibration levels above 0.5 IPS should be considered very high. Readings at this level typically indicate a real problem. You may want to take another set of readings or get a second opinion to ensure your readings are accurate. If the vibration level is determined to still be high, alert management of the situation. If a repair is initiated, make a note of the repair in your records

and confirm the problem has been resolved after follow-up readings are taken.

- When inspecting machines with proximity probe and bearing temperatures readout displays:
 - Look for any vibration or temperature levels that are running in an alert or danger condition. Alert and danger set points are usually provided by the original equipment manufacturer (OEM).
 - Any points in alarm should be investigated immediately. Determine if the alarm reading(s) is real or bogus.
 - Any points in a “Not OK” condition or non-functional should be reported and repaired as soon as possible.
 - Report confirmed alarm readings to management immediately. After maintenance has addressed the problem, follow up to ensure the machine has been returned to a safe running condition.

Remember that these are general guidelines. They may need to be modified based on the actual equipment type, criticality, and operating conditions. When available, use the OEM's recommendations to set acceptable machine operating limits. Anytime you are unsure of your findings, involve others, including the OEM, to help you decide on the appropriate course of action.

19

Addressing Reciprocating Compressor Piping Vibration Problems: Design Ideas, Field Audit Tips, and Proven Solutions

Anyone who has worked around reciprocating compressors has probably noticed piping around reciprocating compressors vibrating. Due to the presence of gas pulsations generated by reciprocating compressors and the mechanical nature of piping, piping vibration will always be present. However, with regular field inspections around reciprocating compressors, most piping vibration issues can be detected and addressed before they can lead to piping failures.



Figure 19.1 Operators must remain vigilant around reciprocating compressors and their associated piping. Poor designs and changing conditions can lead to excessive piping vibration.

Process piping performs a vital role in processing and fluid transport facilities by tirelessly transporting gases and liquids between different stages of processes or from point “A” to point “B.” However, piping is often forgotten due to its apparent passive nature; it only garners attention if failure occurs due to corrosion or fatigue. Corrosion is an electrochemical mechanism related to metal and fluid properties whereas fatigue is a mechanical process related to excessive cyclic motion that results in damaging stress levels. This section addresses the risk of piping failure due to vibration-related fatigue.

All piping associated with machinery vibrates due to the shaking forces that machines generate. Some forces are transferred to the piping via the machine flanges and mounts whereas others are fluid borne. Fluid-borne pressure pulsations generated by machines can result in destructive excitation forces under certain conditions. Here are some field situations that can lead to piping failures:

1. The frequency of any machine excitation, e.g., 1 x rpm and 2 x rpm, matches a piping natural frequency.
2. The unmagnified pressure pulsations, e.g., 1 x N x rpm and 2 x N x rpm—where N is the number of impeller vanes, number of plungers, etc., generated by a machine—are sufficient to create excessive piping shaking forces.
3. The pressure pulsations generated by a machine excite an acoustic resonance within the piping, which in turn generates large shaking forces at a piping elbow.
4. The combination of an acoustic resonance and piping resonance coincide.
5. The piping system is inadequately supported or restrained to resist normal machine generated excitation forces.

Once large piping shaking forces are present for whatever reason, piping may be prone to fail if the developed deflections, which can be correlated to stresses, are such that the endurance limit, a limiting stress, of the piping material of construction is exceeded.

Think of pressure pulsations as little hammers that constantly impact the piping at every elbow, tee, reduce, and valve. If fundamental (1x) or harmonics (2x, 3x, etc.) pulsation frequencies acting on piping coincide with any of the piping mechanical natural frequencies (MNF), a resonant condition will occur. Similar to a tuning fork, the piping will resonate when excited at any of the mechanical natural frequencies. High amplifications levels of 20 to 40 times are often seen when a resonance condition is present. As a means of avoiding dangerous resonance conditions, API 618 (5th edition) recommends that all MNF in the piping system and structure should be designed so that they are all less than 2.4 times the compressor running speed (see Figure 19.2).

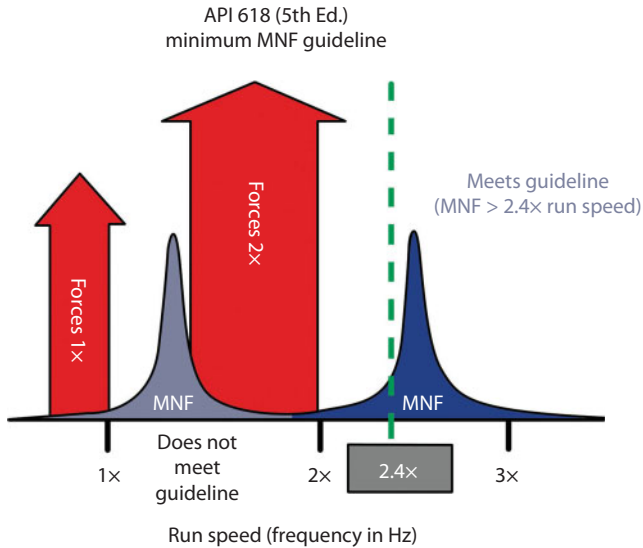


Figure 19.2 If fundamental (1x) or harmonics (2x, 3x, etc.) pulsation frequencies acting on piping coincide with any of the piping mechanical natural frequencies (MNF), a resonant condition will occur. Mechanical resonance conditions usually result in abnormal vibration levels that can result high piping stress. In extreme cases, piping vibration can lead metal fatigue and catastrophic failures.

In most cases, piping systems are properly designed; however, there may be times when vibration levels rise to levels where piping fatigue could occur. There are multiple reasons why a carefully designed piping system could become problematic: Perhaps a) process conditions are different from the analyzed values, b) required orifices were installed or mis-installed, or c) the field supports are either missing or mis-installed. It is not uncommon that piping modification such as orifices and pipe restraints need to be installed after the original compressor installation to correct piping design issues or installation errors.

19.1 Piping Restraints

Pipe restraints (commonly called pipe clamps) are used in piping applications where significant pressure pulsations or

shaking forces are likely to be present. There have been instances where pipe *guides*, which are loosely fitting restraints designed to prevent side to side piping movement while allowing for linear thermal expansion, have been erroneously installed in place of pipe clamps around reciprocating machines. A pipe clamp is a device used to control vibratory motion in all directions by tightly gripping the pipe. Figure 19.3a shows an uninstalled clamp with liner material, also referred to as gasketing, and Figure 19.3b illustrates how a pipe clamp should be installed. Notice in Figure 19.3b that there is an air gap between the clamp and the soleplate to ensure a tight fit around the pipe.

19.2 Pipe Clamping Systems

A pipe clamping system, which is made up of the pipe clamp affixed to a concrete pier or some type of rigid structural member must be rigid enough to resist the forces they are trying to control. Keep in mind that to be effective all pipe clamps must also be properly installed and maintained in the field. The following guidelines are aimed at insuring pipe clamps are effective for years after their installation.

19.3 Guidelines

1. Place piping supports at every elbow and concentrated mass. Locations are usually determined by the firm conducting the dynamic analysis.
2. Always use a pipe clamp approved by the engineering department.
3. When specifying piping clamps, always check with engineering before selecting the belting or liner material. The belting or liner material, which is placed between the pipe and the clamp, ensures a tight fit between the two surfaces.

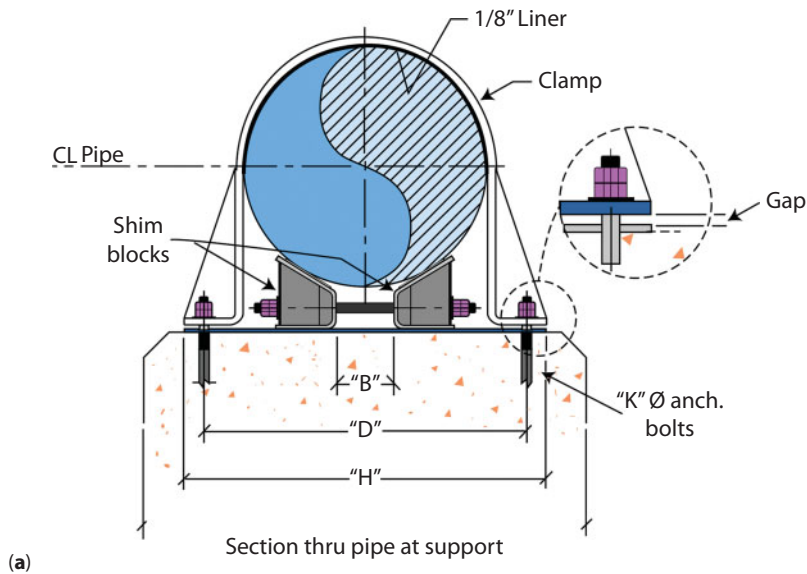


Figure 19.3 (a) Pipe clamp with liner material on the clamp ID and on pipe wedges. (b) Properly installed pipe clamp.

4. Anchor bolts should have a total L/D ratio (The ratio of the bolt length to the bolt diameter.) of 24 to 36. 1/3 of the anchor bolt should be free length, or 8 to 10 times the bolt diameter.

19.4 Piping Assessment Steps

Most plant personnel are not involved in the piping design process. They tend to get involved after the installation and start-up of the compressor and piping. This section is meant to assist those drawn into piping vibration problems that are encountered in the field after the compressor is under full load conditions and problems begin to arise. Here is a general procedure for evaluating vibrating field piping.

19.4.1 First, Perform the Following Pre-Field Analysis Steps

1. Ensure an acoustic and mechanical analysis has been conducted. If an acoustic and mechanical analysis has not been performed, you may want to initiate one.
2. Ensure recommended orifices and supports have been installed. Review all the recommendation drawings to understand what was recommended.
3. Ensure the operating conditions are correct
 - a. Speeds
 - b. Gas analysis

19.4.2 Next

1. Talk to anyone who works close to the problem areas. Ask if a) this is a new installation, b) if there have been any recent modifications to the piping system, and c) if the compressor operating conditions are normal or not.
2. Identify the problem points. If there are only a few problem locations, the field analysis should be quick and easy. However, if there are numerous points of concern, you will need to record numerous points

- simultaneously in order to understand the phase relationships between points.
3. Measure or have a qualified professional measure the vibration levels to determine if they are acceptable or not. (Refer to Chapter 20, "Collecting and Assessing Piping Vibration," for more information about assessing piping vibration levels.)
 4. If the vibration levels are excessive, identify any abnormal conditions, i.e., compressor speed, pressures, gas density, etc.
 5. There are typically several possible outcomes:
 - a. Change operating conditions if possible
 - b. Add bracing or supports
 - c. Bring in experts to evaluate the situation and redesign if necessary

19.4.3 Problem Locations

Be on the lookout for unsupported small-bore piping branches, such as those seen in Figure 19.4. Cantilevered branches and PSV branches tend to be less supported than major process piping. You can either support them off the main piping (see Figure 19.5), gusset them, or eliminate them.

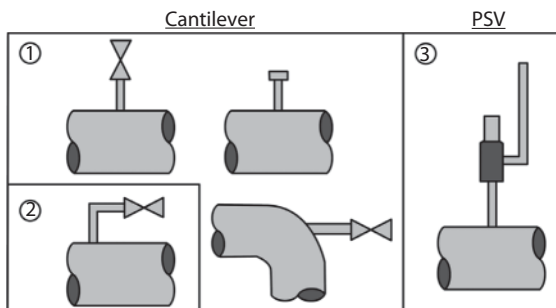


Figure 19.4 Some typical small-bore piping designs that can result in excessive vibration if not properly braced.

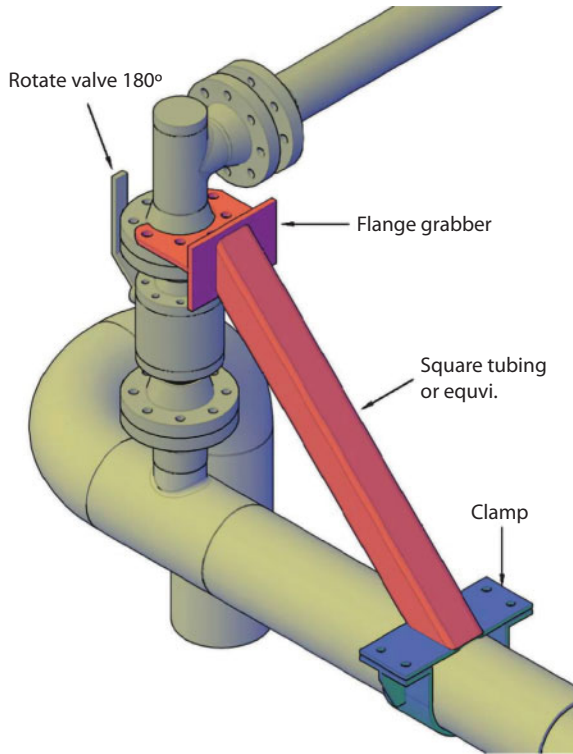


Figure 19.5 PSV support off a main piping.

19.5 Attaching Pipe Clamps to Structural Members

There will be times that due to the elevation of a pipe or space limitations a pipe clamp may need to be affixed to a structural member instead of to a concrete pier (see Figure 19.6). In these cases, ensure that the design provides adequate access to the bottom nuts and that there is a free bolt length of at least eight times the bolt diameter. Here are a few more notes concerning pipe clamps attached to structural members:

- The structural member that the clamp is attached to must be more than twice as rigid as the pipe it



Figure 19.6 Pipe clamp with integral spacers using cap screws.



Figure 19.7 A properly installed pipe clamp bolted to a concrete pier. Notice that there is a gap between the pipe clamp and the pedestal support to ensure there is adequate clamping force present.

is restraining. If it is not, the clamp will be totally ineffective at controlling piping vibration.

- Use bolt spacers to ensure the stretch length of a bolt is at least eight times the bolt diameter. A spacer is an extra strong tube or pipe that fits around a clamp bolt to increase its effective stretch length.



Figure 19.8 Correctly installed pipe clamps on concrete footing.



Figure 19.9 An incorrectly installed pipe clamp. Notice that with the clamp bottomed out there is an air gap between the pipe and the top of the clamp, i.e., there is no clamping force on the pipe.

- Cap screws are preferable.
- Use approved torque values to properly tighten the clamp bolts.



Figure 19.10 There is insufficient clamping force on this clamp due to the loss of wedge travel. (Notice that wedges are touching.).

19.5.1 Installation Examples

Figure 19.7 through 19.10 show some good and bad examples of pipe clamp installations that have been found in the field.

19.5.2 Here Are a Few More Pipe Clamp Tips

1. The surface of the clamp should be parallel with the surface of the concrete pier.
2. There should be a washer under every pipe clamp nut.
3. Foundations should have a minimum depth below grade and below the frost depth.
4. Concrete should have a minimum specified compressive strength.
5. Foundations should be constructed with footings.
6. If adjustable wedges are used, ensure they fit properly around the pipe. Below is an example of a pipe clamp with poorly fitted wedges.
7. The pipe must be in full contact with the sole plate liner.

8. All pipe clamps should be inspected periodically to ensure they are still effective at controlling vibration. Specifically, check that there are no broken hold-down bolts and that the clamp gasket materials are still intact and have not degraded.
9. Seating Pipe Clamps: New pipe clamps should always be checked for proper fit around pipe. The goal is to get 50% or more contact between the gasket material in the ID of the clamp and the pipe. If you are not sure of the fit, you can use a feeler gauge to ensure you have the proper contact. You shouldn't be able to get a feeler gauge between the clamp and the pipe if the fit is correct. If you are able to get a feeler gauge between the clamp liner and the pipe, you probably have poor contact. You can use a rubber mallet on top of the clamp to seat it against the pipe. Continue to impact the pipe clamp with the rubber mallet until you are sure it is fully seated. Once the clamp has been fully seated-in, recheck all clamp bolt torques and re-torque as required to achieve the recommended values.

20

Collecting and Assessing Piping Vibration

The purpose of this introduction to piping vibration analysis is to provide readers with a flavor of what is involved in the data collection process and how piping vibration data is evaluated.

If large piping shaking forces are present, for whatever reason, piping may be prone to fail if the experienced deflections are such that the endurance limit of the piping material of construction is exceeded. The evaluation guidelines we recommend are based on a proven methodology that has been used by vibration professional for decades. The assessment method defines marginal, correction, and danger piping vibration levels based on the amplitudes of the spectral frequency components detected (see Figure 20.1). If vibration levels are kept below the

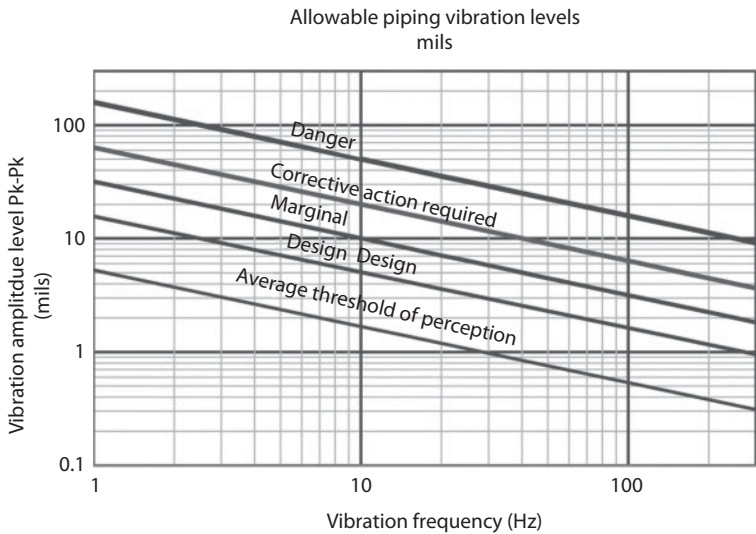


Figure 20.1 Piping assessment chart (Based on Engineering Dynamics Inc. (EDI) guidelines).

marginal vibration levels, critical stress levels should never be exceeded.

Piping vibration is normally acquired with a magnetically mounted accelerometer (see Figure 20.2 and 20.3 below) that is connected to a spectrum analyzer. The largest frequency component in the vibration spectrum should be compared with allowable piping vibration guidelines (see Figure 20.1). For example, you can see by inspection in Figure 20.4 that the plot of the 2x vibration component (labeled as “12.6 mil p-p @ 28.5 Hz” in Figure 20.4) is by far the largest vibration components in these spectra. The 2x component peaks at 28.5 Hz (1710 cpm) at a level of 12.6 mils, probably due to some type of mechanical resonance. The peak amplitude of the 2x component was found to be above the “correction action required” vibration level.



Figure 20.2 Mag-mounted tri-axial accelerometer on a pipe.

20.1 Piping Analysis Steps

Before beginning any evaluation effort, confirm that the field analysts have experience performing field piping assessments. Once approval is obtained for them to enter the processing area, the field analyst team typically follows these steps to complete their evaluation:

1. They perform a field piping survey to identify vibration problem locations.
2. They gather data at the points of interest using magnetically mounted accelerometers.
3. They generate and inspect spectra for all the vibration points to identify any significant spectral peaks.
4. They evaluate all the largest spectral peaks using the “Piping Assessment Chart” (see Figure 20.1).



Figure 20.3 Mag-mounted tri-axial accelerometer on a valve.

5. They report any of the piping vibration frequency components that fall above the “Corrective Action Required” line to operations.
6. After carefully reviewing all their results, they will provide a formal report with a summary of what was found along with recommendations based on good engineering practices.

20.2 Piping Vibration Examples

To better understand the piping assessment process, we will review a few examples. In each case, we will refer to Figure 20.1. to determine if the piping vibration levels are acceptable or not:

Example #1:

An 8” pipe is vibrating 20 mils pk-pk at 2000 cpm. Since 2000 cpm is 33.33 Hz, by inspection we see this vibration amplitude falls between the “correction” and “danger” levels.

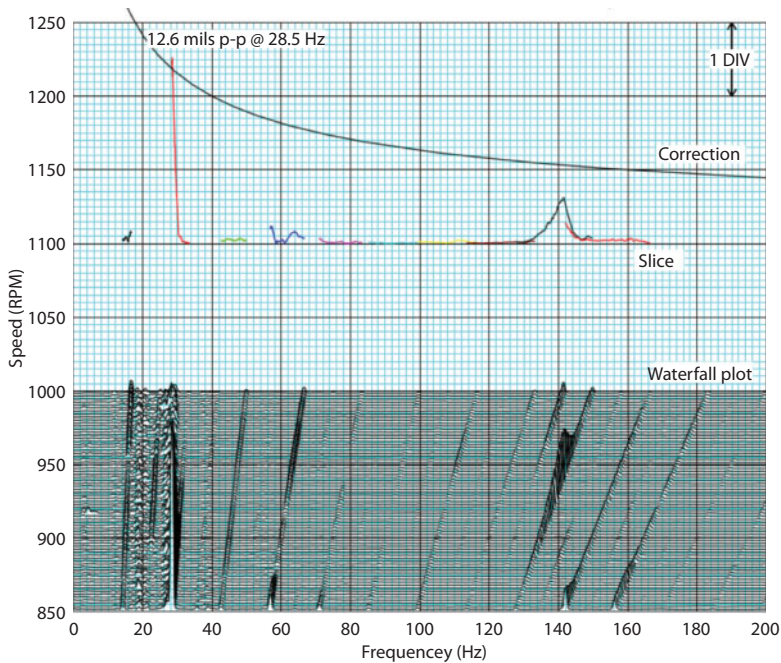


Figure 20.4 Waterfall Display of Piping Vibration.

Example #2:

A 6" pipe is moving 10 mils pk-pk at 600 cpm. Since 600 cpm is 10 Hz, by inspection we see this vibration amplitude falls at the lower boundary of the "marginal" level.

Example #3:

An accelerometer has been placed on a 12" compressor discharge line in the horizontal and vertical direction. The compressor runs at 900 rpm. The following spectral vibration data was acquired:

Horizontal data

Frequency (Hz)	Amplitude (in mils pk-pk)	Assessment based on EDI criteria above
15	9	Marginal
30	40	Greater than danger
45	5	Marginal
60	2	Design

Vertical data

Frequency (Hz)	Amplitude (in mils pk-pk)	Assessment based on EDI criteria above
15	3	Less than design
30	8	Greater than design
45	2.3	Design
60	1	Less than design

The 30 Hz component in the horizontal direction is the only spectral component of immediate concern. Additional testing and data acquisition will be required to determine if this high vibration is due to excessive compressor pulsations, lack of piping constraints, piping resonance, or some other phenomenon.

Appendix A: Practice Problems Related to Chapters 1 Through 4 Topics



This appendix contains some practice problems involving basic gas and compressor theory for those who want to better understand the theory presented in Chapters 1 through 4.

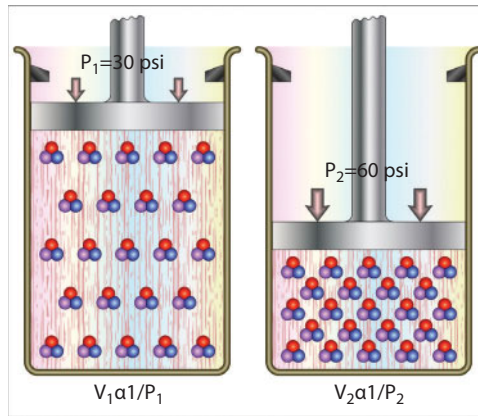


Figure A.1 Gas compressed with a piston.

Problem #1: Boyle's Law example

A volume with a movable piston has an initial volume of 6 ft^3 at a pressure of 30 psia (pounds per square inch absolute). What will the new volume of the trapped gas be when it is compressed to a pressure of 60 psia?

Answer:

First, we write out Boyle's Law:

$$P_1 V_1 = P_2 V_2 \quad (1.4)$$

Solving for P_2 we get:

$$V_2 = \frac{P_1 V_1}{P_2} = \frac{30 \text{ psi} \times 6 \text{ ft}^3}{60 \text{ psi}} = 3 \text{ ft}^3$$

Problem #2: Second Boyle's Law example

What pressure ratio is required to reduce the volume in a trapped volume by one third?

Answer:

First, we write out Boyle's Law:

$$P_1 V_1 = P_2 V_2 \quad (1.4)$$

Next, we solve for P_2/P_1 , which is called the pressure ratio:

$$\frac{P_2}{P_1} = \frac{V_1}{V_2}$$

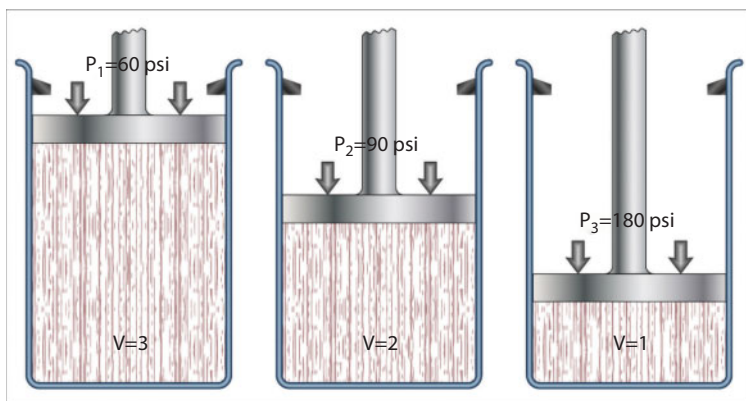


Figure A.2 How gas volume varies with pressure.

Since the volume is decrease by one-third of its original value, we know that $V_1/V_2=3$. Plugging the volume ratio into our modified Boyle's equation we get:

$$\frac{P_2}{P_1} = \frac{V_1}{V_2} = \frac{3 \text{ ft}^3}{1 \text{ ft}^3} = 3$$

Therefore, we can conclude that a pressure ratio of 3 will reduce a volume to one-third of its original volume. Note: Make sure you use absolute pressure values when using this equation.

Problem #3: Determining the molecular weight of a pure gas

What is the molecular weight of propane?

Answer:

By inspecting the molecular structure of propane in Figure A.3, we can say propane contains three carbon atoms in and eight hydrogen atoms. By tabulating this information in a table similar to Table A.1, we can easily determine propane's molecular weight.

Problem #4: Calculating the average molecular weight of a gas mixture.

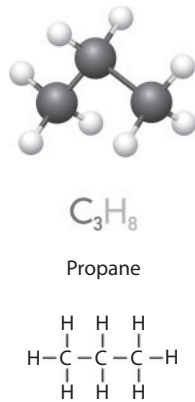


Figure A.3 The molecular structure of propane.

Table A.1 Composition of methane.

Element	Symbol	Atomic mass	Number of atoms in molecule	Mass percent
Hydrogen	H	1.00794	3	18.287%
Carbon	C	12.0107	8	81.72%

Molecular Weight of Methane = $8 \times 1.00794 + 3 \times 12.0107 = \underline{44.09}$

Assume you have a gas mixture composed of 80% hydrogen, 12% methane, 6% ethane, and 2% propane. Determine the average molecular weight of the mixture.

Answer:

The best way to solve gas composition problems is to enter the given data into a spreadsheet like the one seen in Table A.2. By summing all the proportional molecular weights (MW times the %volume of each given components), you will arrive at the average molecular weight of the mixture. In this example, the sum of the proportional MW values is 6.254. Remember that your answer should always fall somewhere between the lightest MW and the heaviest MW of the gases in the mixture.

Problem #5: Converting SCFM to ACFM

Table A.2

Gas	Chemical formula	Mole% or Vol %	Molecular weight	Proportional MW
Hydrogen	H ₂	80	2	1.6
Methane	CH ₄	12	16.4	1.968
Ethane	C ₂ H ₆	6	30.07	1.804
Propane	C ₃ H ₈	2	44.09	0.882
	Total %	100.0		6.254 is the MW of the total gas mixture

Let's assume you need a compressor to compress 20,000 scfm of gas. You have determined the suction pressure will be 50 psig at 100 °F and the compressibility factor for the gas will be 1.0. First, calculate the actual standard cubic feet per minute (ACFM) at the compressor's suction conditions using equation 2.2.

$$ACFM = SCFM \times \left(\frac{14.696}{P_{actual}} \right) \times \left(\frac{T_{actual}}{520} \right), \quad (2.2)$$

where P_{actual} is in psia and T_{actual} is in °R

Plugging all the given values into the equation we get:

$$ACFM = 20,000 \text{ scfm} \times \left(\frac{14.7 \text{ psi}}{(50 \text{ psi} + 14.7 \text{ psi})} \right) \times \left(\frac{(100 \text{ F} + 459.67)}{520} \right) \times 1.0$$

$$ACFM = 20,000 \text{ scfm} \times \left(\frac{14.7}{64.7} \right) \times \left(\frac{559.67}{520} \right) \times 1.0$$

$$ACFM = 4890.1 \text{ ft}^3$$

Based on the calculated flow (4890.1 ft³) and the given suction pressure (50 psig), we can see by studying Table A.3 that a rotary screw and a multistage centrifugal compressor design could fit this application. The next step would be to contact a

Table A.3 Compressor Coverage Table (English Units)⁽¹⁾.

Compressor type	Inlet flow range (ICFM)	Maximum discharge pressure (psig)
Rotary Screw (oil free)	0–35000	175
Rotary Screw (oil flooded)	0–35000	290
Reciprocating (single stage)	0–2000	250
Reciprocating (multistage)	0–3500	80000
Centrifugal (single stage)	90–2000	2000
Centrifugal (multistage)	900–20000	10000
Axial	80000–800000	90

rotary screw and multistage centrifugal compressor manufacturers to see if they offer a compressor model for your specific application.

Problem #6: Converting ACFM to SCFM

In this problem, we will assume that you have a reciprocating compressor compressing natural gas with a 300 psig suction pressure and suction temperature of 80 °F that is rated at 1000 acfm (actual cubic feet per minute). You are asked to convert the acfm flow to scfm flow. (Assume that you have a compressibility factor of 1.)

Answer:

The following acfm to scfm conversion formula can be found in Chapter 2:

$$SCFM = ACFM \times \left(\frac{P_{actual}}{14.696} \right) \times \left(\frac{520}{T_{actual}} \right) \quad (2.3)$$

Plugging in the values given you get:

$$SCFM = 1000 \text{ acfm} \times \left(\frac{(300 \text{ psi} + 14.7 \text{ psi})}{14.7 \text{ psi}} \right) \times \left(\frac{520}{(80 \text{ F} + 459.67)} \right)$$

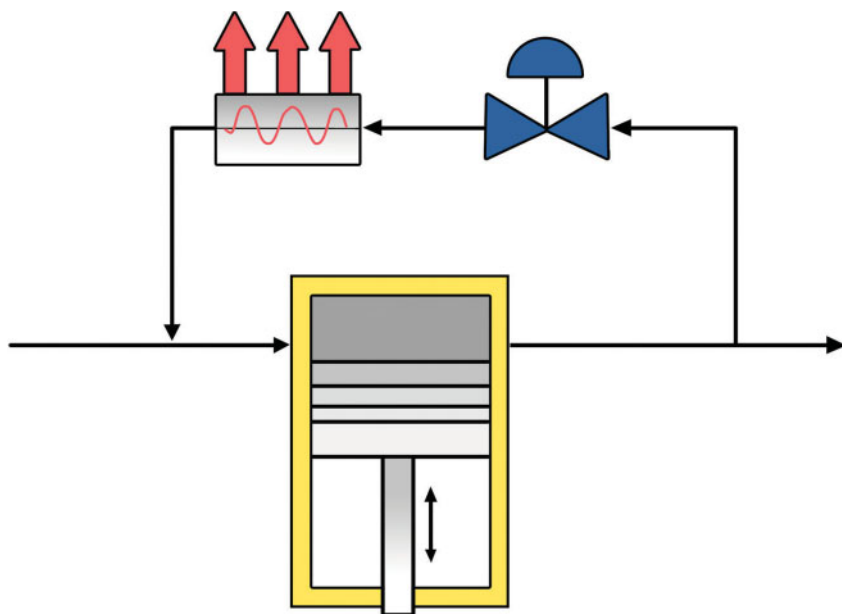


Figure A.4 Reciprocating compressor spillback line and cooler schematic.

$$SCFM = 1000 \text{ acfm} \times \left(\frac{314.7}{14.7} \right) \times \left(\frac{520}{539.67} \right) = 20627.8 \text{ scfm}$$

Problem #7: Calculating compression ratios

When performing theoretical discharge calculations, you must be able to determine the compression ratio of the compression stage you are analyzing. For example, let's say you have a reciprocating compressor stage with a suction pressure of 200 psig and a discharge pressure of 450 psig. What is the compression ratio?

Answer:

First, you must convert both pressures to absolute pressure values: 200 psig = 200 + 14.7 = 214.7 psia; 450 psig = 450 + 14.7 = 464.7 psia. Then simply divide the discharge pressure by the suction pressure to get the pressure ratio: 464.7 psia / 214.7 psia = 2.164.

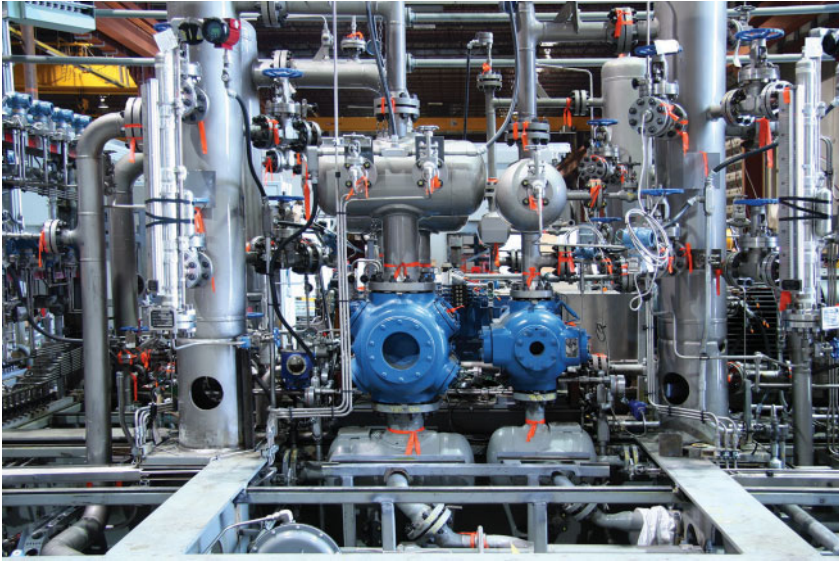


Figure A.5 Two stage reciprocating compressor installation.

Problem #8: Compressor Staging Problem

Let's imagine that you have a compressor with two stages of compression that can both develop a 2.5 compression ratio. If the suction pressure at the first stage of compression is 150 psig and there is a 10 psi pressure drop in the piping from the first stage compressor discharge and the second stage compressor suction, what is the expected second stage discharge pressure?

Answer:

Let's attach this problem in steps:

Step 1: Convert the suction pressure to absolute pressure:
 $150 \text{ psig} + 14.7 \text{ psi} = 164.7 \text{ psia}$

Step 2: Determine the first stage discharge pressure: $164.7 \text{ psia} \times 2.5 = 411.75 \text{ psia}$

Step 3: Determine the second stage suction pressure after the pressure drop in the piping: $411.75 \text{ psia} - 10 \text{ psi} = 401.745 \text{ psia}$

Step 4: Determine the second stage discharge pressure:
 $401.75 \times 2.5 = 1004.37 \text{ psia}$



Figure A.6 Electric motor driven reciprocating compressor.

Problem #9: Adiabatic compression example

In this example, we will assume that you are considering a reciprocating compressor requiring a compression ratio of 5 and a suction temperature of 100 °F. To make this problem challenging, let's say the compressor's rings and rider bands are only good up to 350 °F. If we know the ratio of specific heats (k) for this gas is 1.31, what would you expect the discharge temperature to be if the full compression ratio is developed across one compressor stage?

Do you think that another stage of compression makes more sense? If so, why?

Answer:

First we need the adiabatic compression formula to determine what the expected discharge temperature will be:

$$T_f = T_i \times \left(\frac{P_f}{P_i} \right)^{\left(\frac{k-1}{k} \right)} \quad (3.2)$$

Plugging in the information provided we get:

$$\begin{aligned} T_f &= T_i \times \left(\frac{P_f}{P_i} \right)^{\left(\frac{k-1}{k} \right)} \\ &= (100 \text{ } ^\circ\text{F} + 459.67) \times (5.0)^{\left(\frac{1.31-1}{1.31} \right)} = 819.04 \text{ } ^\circ\text{R} = 359.37 \text{ } ^\circ\text{F} \end{aligned}$$

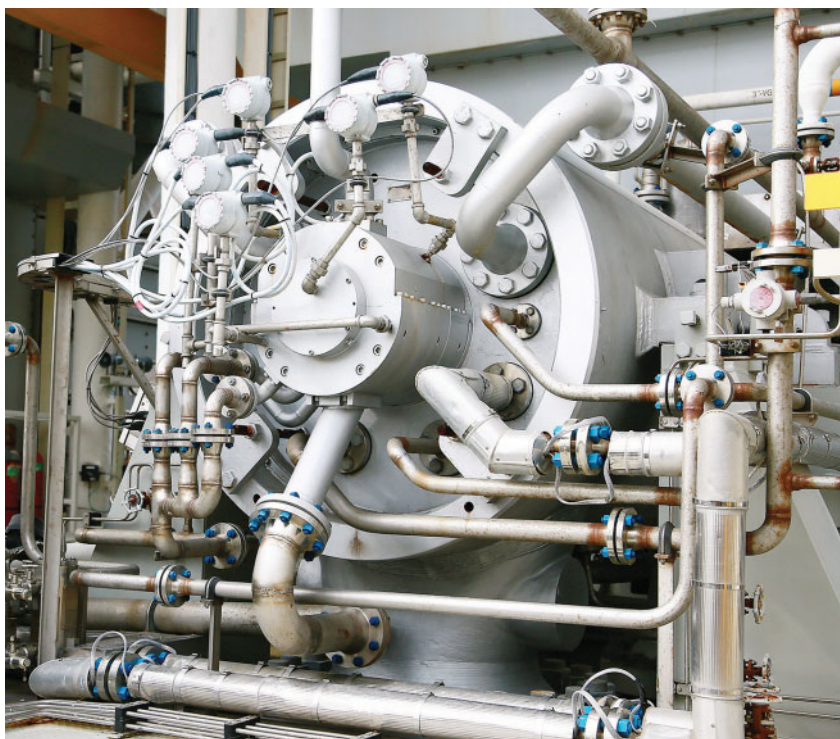


Figure A.7 Centrifugal barrel compressor.

The calculated theoretical discharge temperature is higher than the design limit of the rings and rider bands. In order to protect the compressor from upset conditions and compressor degradation, we should recommend that another stage of compression be specified.

Problem #10: Polytropic compression example

In this example, we are compressing pure methane at 100 °F ($k = 1.31$ @ 60 °F) from 200 psig to 325 psig. Let's say you have visited the compression site and have noticed that the compressor discharge temperature is running about 202 °F. How does this temperature compare to the theoretical discharge temperature, knowing that the manufacturer has documented the compressor's polytropic efficiency (ϵ_p) to be 82%?

Answer:

First, we need to calculate the compression ratio: P_2 (absolute)/ P_1 (absolute) = (325 psi + 14.7 psi) / (200 psi + 14.7 psi) = 1.5822. Next, we will need the polytropic theoretical discharge temperature formula:

$$T_f = T_i \times \left(\frac{P_f}{P_i} \right)^{\left(\frac{n-1}{n} \right)} \quad (3.4)$$

Where:

T_f is the final absolute temperature (T in degrees Rankine = T in degrees Fahrenheit + 459.67)

T_i is the initial temperature (T in degrees Rankine = T in degrees Fahrenheit + 459.67)

P_f is the final absolute pressure ($P_{sia} = P_{sig} + 14.7$)

P_i is the initial absolute pressure ($P_{sia} = P_{sig} + 14.7$)

n is the polytropic index

$$\frac{n-1}{n} = \frac{k-1}{k} \times \frac{1}{\epsilon_p} \quad (3.5)$$

Where:

n is the polytropic index

k is the ratio of specific heats

ϵ_p is the polytropic compressor efficiency.

Plugging the given values into equation 3.5 we get the polytropic index:

$$\frac{n-1}{n} = \frac{k-1}{k} \times \frac{1}{\epsilon_p} = \frac{1.31-1}{1.31} \times \frac{1}{0.82} = 0.2886$$

Plugging in the information provided we get:

$$\begin{aligned} T_f &= T_i \times \left(\frac{P_f}{P_i} \right)^{\left(\frac{n-1}{n} \right)} = (100^\circ\text{F} + 459.67) \times (1.5822)^{(0.2886)} \\ &= 638.91^\circ\text{R} = 179.24^\circ\text{F} \end{aligned}$$

We can see that the actual compressor discharge temperature is running 22.76 °F (202 °F - 179.24 °F) hotter than what is expected. This result seems to indicate there may be some

internal degradation of the compressor. Here are a few possible action steps before making any formal recommendations:

1. Double check all your input values.
2. Determine when the compressor was last overhauled. If it's been over 5 to 7 years, this might explain your findings.
3. Get a gas analysis to verify the actual composition of the gas being compressed.
4. If you discover that the gas is really a mixture, then determine a more accurate value for the ratio of specific heats (k) and recalculate the theoretical discharge temperature.

If there is still a discrepancy between the theoretical and actual discharge temperature, talk your results over with a machinery engineer or with the compressor manufacturer before making any recommendations to management.

Appendix B: Glossary of Compressor Technology Terms

Absolute Pressure – The existing gauge pressure plus atmospheric pressure measured from absolute zero.

ACFM – Actual cubic feet per minute, ACFM, which is used to define the flow of gases and vapors under pressure.

Adiabatic Compression – A compression process is one in which no heat is gained or lost by the system.

Aftercooler – A heat exchanger used for cooling air discharged from a compressor. Resulting condensate may be removed by a moisture separator following the aftercooler.

Air Dryer – A device that removes moisture from compressed air. Typically accomplished by cooling the air through a refrigerator or desiccant bed.

Area – The surface exposed to the flow of a fluid.

Atmospheric Pressure – The measured ambient pressure for a specific location and altitude in PSI (pounds per square inch).

Automatic Sequencer – A device which operates compressors in sequence according to a programmed schedule.

Avogadro's Law – A gas law that states that equal volumes of all ideal gases (at the same temperature and pressure) contain the same number of molecules.

Balance Drum – A device added to some centrifugal compressor rotors as a means of reducing the net thrust load generated by the internal pressure forces. Also called a balance drum.

Balance Line – A piping segment used to transport gas from the low-pressure end of the balance piston and the compressor's suction.

Balance Piston – *See balance drum*

Bar – A unit of pressure. One (1) bar = 14.5 PSI.

Boyle's Law – A gas law that states that equal pressure is inversely proportional to volume (when temperature is constant).

Capacity – The amount of air flow delivered under specific conditions, usually expressed in cubic feet per minute (CFM).

Centrifugal Compressor – A type of dynamic compressor that uses impellers to accelerate the gas to a high velocity and then decelerate the gas using diffusers to create pressure. Unlike displacement compressors that work at a constant flow, dynamic compressors work at a constant differential pressure. Their performance is affected by external conditions such as the gas's molecular weight and changes in the suction pressure and temperature.

Charles's Law – A gas law that states that volume is proportional to temperature (when pressure is constant). Remember that temperature must be measured in Kelvin.

Check Valve – A valve which permits flow in only one direction.

Coalescing Filter – A specific type of filter that not only works to remove dust and particulates from the air, but also oil from the air lines.

Compressibility Factor (Z) – Also known as the compression factor, is the ratio of the expected volume of a gas to the volume of an ideal gas at the same temperature and pressure.

Compression Efficiency – Ratio of the theoretical to the actual work required to compress air that takes into account slip leakage and frictional losses.

Compression Ratio – The ratio of the discharge pressure to the inlet pressure.

Constant Speed Control – A system in which the compressor is run continuously and matches air supply to air demand by varying compressor load.

Covalent Bond – A chemical bond that involves the sharing of electron pairs between atoms.

Crankshaft – An engine or reciprocating component that converts reciprocating linear motion into rotary motion (as in engines) or rotary motion into reciprocating linear piston motion (as in reciprocating compressors). (Also known as a Crank.)

Cubic Feet Per Minute (CFM) – The volume of gas or air that is able to pass through an opening in one minute.

Cut-In/Cut-Out Pressure – Respectively, the minimum and maximum discharge pressures at which the compressor will switch from unload to load operation (cut in) or from load to unload (cut out).

Cycle – The series of steps that a compressor with unloading performs; 1) fully loaded, 2) modulating (for compressors with modulating control), 3) unloaded, 4) idle.

Cycle Time – Amount of time for a compressor to complete one cycle.

Demand – Flow of air at specific conditions required at a point or by the overall facility.

Density – Mass (m) divided by the volume.

Desiccant – A material having a large proportion of surface pores, capable of attracting and removing water vapor from the air.

Dew Point – The temperature at which moisture in the air will begin to condense if the air is cooled at constant pressure. At this point the relative humidity is 100%.

Diatomic Gases – Gases composed of two atom molecules.

Diffuser – A section of a centrifugal compressor that converts velocity energy to pressure energy.

Discharge Pressure – The pressure produced at the discharge nozzle of a pump or compressor under specific conditions, measured in PSI (pounds per square inch).

Discharge Temperature – The temperature at the discharge flange of the compressor.

Drain Cock – Used to release accumulated moisture from the compressor tank. (Also see drain valve)

Drain Valve – A drain valve works to remove surplus liquid from the compressor. There are manual, float and electronic types of drain valves.

Driven machine – The process machine that converts mechanical power to fluid power.

Driver – Any element that provides a motive force for a machinery train.

Dryer – Any device or system used to remove moisture in a compressed air system.

Dry Gas Seal – A seal consisting of a rotating, hard-face mating ring with either a machined or etched circumferential spiral groove pattern and a primary (stationary) ring in softer material.

Filters – Devices for separating and removing particulate matter, moisture or entrained lubricant from air.

Filter Efficiency – A measure of how efficiently airborne particles are removed from an airstream as they pass through the filter.

Free Air – Air at atmospheric conditions. Be careful with this term because altitude, barometric pressure and temperature will vary. This term may not mean air at identical conditions.

Full-Load – Gas or air compressor operation at full speed with a fully open inlet and discharge delivering maximum air flow.

Gas – One of the three basic phases of matter. While air is a gas, in pneumatics the term gas normally is applied to gases other than air.

Gas Turbine – A type of internal combustion engine that has an upstream rotating compressor coupled to a downstream power recovery turbine with a combustion chamber in between.

Gauge Pressure – The pressure determined by most instruments and gauges, usually expressed in psig. Barometric pressure must be considered to obtain true or absolute pressure (PSIG).

Gay-Lussac's Law – A gas law that states that pressure is proportional to temperature (when volume is constant).

GPH – Gallons per hour.

GPM – Gallons per minute.

Guide Vanes – Fixed airfoils designed to direct air, gas, or water into the moving blades of a turbine or around bends in ducts with a minimum loss of energy.

Heat of Compression – The term used to describe the heating effect of compressing gas. If you compress a gas adiabatically, i.e., without allowing heat to enter or leave the volume being compressed, you are performing work on the gas, which in turn increases its internal energy and thus its temperature.

Horsepower, Brake – Horsepower delivered to the output shaft of a motor or engine, or the horsepower required at the compressor shaft to perform work.

Humidity, Relative – The relative humidity of a gas (or air) vapor mixture is the ratio of the partial pressure of the vapor to the vapor saturation pressure at the dry bulb temperature of the mixture.

ICFM – Inlet cubic feet per minute.

Ideal Gas – An ideal gas differs from a real gas in that the particles are assumed to be point masses, that is, particles that have a mass but occupy no volume.

Inlet Guides Vanes (IGVs) – Any set of vanes on the inlet of a compressor or expander designed to provide an efficient means of flow control. Guide vanes not only provide the inlet pressure drop but also impart a whirl motion to the gas as it

enters the compressor or expander impeller. Since this whirl motion is in the rotational direction of the impeller, it reduces the amount of work the impeller is required to do on the gas, resulting in energy savings at the design conditions.

Inlet Pressure – The actual pressure at the inlet flange of the compressor typically measure in PSIG.

Inlet Temperature – Temperature at the inlet flange of the compressor or inlet filter.

Intercooling – The removal of heat from air or gas between compressor stages.

Integrally Geared Compressor – A type of centrifugal compressor composed of multiple compressor rotors that are driven off a slower turning drive gear.

Isothermal Compression – Compression occurring at a constant temperature, which usually occurs inside lab devices that move slowly and are poorly insulated.

Journal Bearing – A babbitted sleeve radial bearing.

Kick-in Pressure (air compressors) – Factory set low pressure point of the pressure switch that starts the compressor to repressurize the tank to a higher pressure.

Kick-out Pressure (air compressors) – Factory set high pressure point of the pressure switch that stops the compressor from increasing the pressure in the tank above a certain level.

Kingsbury Thrust Bearing – Tilting pad thrust bearings are designed to transfer high axial loads from a rotating shaft to a stationary outer casing with minimum power loss.

Leak – An unintended loss of pumped product or compressed gas into the atmosphere.

Load/Unload Control (air compressors) – Control method that allows the compressor to run at full load or at no load while the driver remains at a constant speed.

Loading Sequence (reciprocating compressors) – A control algorithm designed to automatically run a compressor through a series of incremental load steps during a compressor start-up.

Loading is achieved by gradually loading valves, closing volume pockets, or closing spillback valves.

Machine Train – A group of process machine elements interconnected using flexible components called couplings on a common foundation.

Mineral Oil – An oil obtained by refining naturally occurring crude oil.

Modulating Control – System which adapts to varying demand by throttling the compressor inlet proportionally to the demand.

Moisture Separator – A device designed to collect and remove moisture from the air during the cooling process.

Molecular Mass – see *molecular weight*

Molecular Structure – The three-dimensional shape or configuration of the atoms comprising a molecule.

Molecular Weight – A measure of the sum of the atomic weight values of the atoms in a molecule.

Molecule – A group of atoms bonded together, representing the smallest fundamental unit of a chemical compound that can take part in a chemical reaction.

Monatomic Gases – Gases composed of single atoms.

Multistage Compressors – Compressors having two or more stages operating in series.

Oil free Compressor – Air compressor that has no oil inserted into the compression chamber for lubrication, cooling or sealing.

Packing (reciprocating compressors) – A dynamic seal between the cylinder and the piston rod. It consists of a series of Teflon rings mounted in a packing case, which is bolted to the cylinder.

Partial Pressure – The pressure exerted by one of the gases in a mixture if it occupied the same volume on its own.

Performance Map – A diagram created to depict the expected performance of a process compressor and assist users in understanding how their compressor will function under certain conditions.

Pipe Clamp – A physical restraint designed to limit piping vibration.

Piston Displacement – The volume swept by the piston; for multistage compressors, the piston displacement of the first stage is the overall piston displacement of the entire unit.

Polyatomic Gases – Gases made up of molecules with more than two atoms.

Polytropic Compression – Any thermodynamic process which can be represented by $pV^n = C = \text{constant}$, where n is the polytropic index and can take values from -infinity to infinity.

Pounds Per Square Inch (PSI) – A unit of measurement referring to the pressure applied on one square inch of an object's surface.

Pressure – Force per unit area, measured in pounds per square inch (PSI).

Pressure Dew Point – For a given pressure, the temperature at which water will begin to condense out of the air.

Pressure Drop – Loss of pressure in a compressed air or gas system or piping component due to friction or restriction.

Pressure Range – Difference between minimum and maximum pressures for an air compressor. Also called cut in-cut out or load-no load pressure range.

Pressure Switch – An instrument which senses a change in pressure and automatically adds or decreases air in the tank.

PSID (ΔP) – Pounds per square inch differential.

PSIG – Pounds per square inch gauge = PSIA minus atmospheric pressure (14.696).

Purging – The elimination of undesired gas or liquid from a system.

PV Analysis – An analysis method where the four parts of the reciprocating compressor cycle, compression, discharge, expansion and intake, are graphically plotted to produce a pressure vs. volume curve, known as a P-V diagram.

Ratio of Specific Heats – The specific heat at constant pressure divided by the specific heat at constant volume, it is sometimes referred to as the adiabatic index or the heat capacity ratio.

Receiver – A vessel or tank used for storage of gas under pressure. In a large compressed air system there may be primary and secondary receivers.

Reciprocating Compressor – Compressor in which the compressing element is a piston having a reciprocating motion in a cylinder.

Relative Humidity (RH) – The ratio of the partial pressure of a vapor to the vapor saturation pressure at the dry bulb temperature of a mixture.

Rotary Screw Compressor – A compressor which utilizes two intermeshing helical rotors to trap a volume of air, then compress it to a higher pressure. Rotary screw compressors can be run at lower temperatures for 24 hr per day, 365 days a year.

Safety Valve – A valve which limits fluid pressure by releasing some of the pressurized liquid or gas. Also called a pressure relief valve.

SCFM – Standard cubic feet per minute.

Slide Loading Valve – A slide mechanism located inside the compressor casing beneath the twin rotors that adjusts the inlet flow volume. The slide valve can be actuated with either a hydraulic or air cylinder. As the slide moves toward the discharge end of the compressor, a larger and larger area is opened allowing more and more gas to spill back gas back to suction.

Specific Gravity – The ratio of the specific weight of air or gas to that of dry air at the same pressure and temperature.

Specific Heat – The heat required to raise the temperature of the unit mass of a given substance by a given amount (usually one degree).

Speed – The speed of a compressor refers to the number of revolutions per minute (rpm) of the compressor drive shaft or rotor shaft.

Steam Turbine – A mechanical device that extracts thermal energy from pressurized steam and transforms it into mechanical work. Because the turbine generates rotary motion, it is well suited to driving process machinery, such as pumps and compressors.

Surge – The operating point where the centrifugal compressor peak head capability and minimum flow limits are reached. Surge can be a detrimental condition in compression systems because it causes the compressor to vibrate violently due to rapid internal flow reversals, potentially damaging internal compressor parts.

Surge Control – A control system designed to prevent surge from occurring.

Synthetic (man-made) Oil – A manufactured lubricating oil that possesses very good oxidation and viscosity stability properties.

Thermodynamics – The branch of physical science dealing with the relationships between heat and other forms of energy (such as mechanical, electrical, or chemical energy).

Thrust Bearing – A bearing that maintains the axial position of a rotor.

Tilting Pad Bearing – A radial journal bearing that is comprised of multiple, segmented babbitted pads that can move freely around a shaft journal.

Unloaded – (No load) A compressor operating condition when horsepower consumption is at the minimum possible level.

Unloaded Horsepower – The power that is consumed to overcome the frictional losses when operating in an unloaded condition.

Vacuum Pump – A compressor that operates with an intake pressure below atmospheric and discharge pressure.

Valve – A flow control device with internal passages that directs gas flow into alternate paths or prevents flow.

Valve Unloader – A pneumatic or manually activated device that disables a reciprocating compressor valve by physically holding it fully open.

Vane – An airfoil shaped stationary component within a flow passage to guide the flow of gas and extract energy, or around the periphery of a compressor rotor to guide the flow and impart energy.

Viscosity – A measure of the internal friction or the resistance of a fluid to flow. The standard unit of measure is poise or SSU.

Volume pocket – Any variable or fixed volume at the end of a reciprocating compressor cylinders that can be actuated to control flow and load.

Volumetric Efficiency – The ratio of the actual quantity of air delivered to the actual capacity of the compressor.

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